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The Effect of WARTIME FUEL Developments Upon POST-WAR AUTOMOBILES

by A. T. COLWELL

Vice-President, Thompson Products, Inc., and Thompson Aircraft Products Co.

IN exploring this subject, very few engineers seem to agree on any one point – which is an intimation of rapid progress. The automobile industry needs, and has, two types of engineering mind: the visionary "forward-looking" type, and the practical, "feet-on-the-ground" type. The visionary foresees developments which the practical does not quite see how to accomplish with the knowledge and materials available today – both are to a great degree correct in their opinions, the variable factor being the *timing*. Engineers are optimistic over developments to come, but are realistic as to the time element. But feature-story and human-interest writers have expanded the "vision" without

[This paper was presented at the SAE Meeting on Materials—War and Post-War, Cleveland, Ohio, Nov. 10, 1943.]

THE post-war automobile has received generous attention from the feature-story and human-interest writers, but little from the practical engineers who will build these cars.

To bring a more realistic viewpoint into the post-war car picture, Mr. Colwell interviewed 81 of the leading oil and automobile engineers of America. He has formed a weighted average of the combined opinions of all the engineers consulted. Briefly, these conclusions are:

1. Immediately after the war, 1942 models will be built. No new models will be on the market for at least 18 months.

2. Although oil companies hope that eventually only two grades of gasoline will satisfy all demands, it appears that immediately after the war, four grades will be marketed: aviation, of 100 and 100-plus octane; premium, of 85-87 octane; regular, of 75-77 octane; and third grade, of 70 octane.

3. Compression ratio will go up, although ratios above 8:1 are not foreseen for the immediate future.

4. Although cars will be lighter, the small European-type car is not foreseen. A smaller car with 4-cyl engine, however, is likely to become more popular to meet certain economic conditions.

5. Future car design will stress economy rather than performance.

6. Engine size and displacement will not change much with present-weight cars, although engines will be lighter within economical limitations.

7. Automatic transmission will receive great attention.

8. Supercharging will be used on heavy-duty vehicles, but not on automobiles in the immediate future, as long as the dominating factor is economy, not power.

9. Fuel injection will be used on some heavy-duty vehicles, but not on passenger-car engines.

10. Pressure cooling will be widely used.

11. Many engine parts, such as bearings, valves, piston rings, fuel systems, and the combustion chamber, will be improved in various ways, because of higher compression.

■ ■ ■

THE AUTHOR: A. T. COLWELL'S (M '30) career, from the time of his graduation from West Point until his election as vice-president and director of Thompson Aircraft Products Co., has been largely devoted to development work in engine and chassis parts, and he has gained an international reputation for accomplishments in improved heat-resistant valves and valve-seat inserts for high-duty automobile and aircraft engines. Mr. Colwell is a past-president of the SAE, and has been active on many technical and administrative committees of the Society.

conscience, to the general confusion of all. This is somewhat like the beloved Will Rogers' conversation with an admiral of the last war.

Will: "Admiral, I can tell you how to get rid of German submarines."

Admiral: "How?"

Will: "Boil the ocean. They can't exist in boiling water."

Admiral: "That's right; but how will we boil the ocean?"

Will: "I've given you the general idea – *you* work out the details."

■ Unrealistic Publicity

The post-war automobile has received generous attention in this respect, but little of it from the practical engineers who will build the cars. They are engrossed in war work and dropped practically all automobile experimental work when the war began. The public impression seems to be that because of war developments, tremendous changes will occur immediately. Points cited are the strides made in aviation fuel, plastics, synthetic rubber, light metals, and secret developments. The exaggerated trend of present publicity would indicate immediate post-war automobiles of perhaps one-half their present weight, achieving three times the number of miles per gallon, constructed largely of light-weight metals common to the aircraft industry, feeding 100-octane (or better) gasoline into small, high-speed, high-performance, high-compression, high-economy engines, supercharged and with fuel injection; the car in general resembling a combination of a crystal ball and a rolling solarium.

■ Leading Engineers Interviewed

To clarify the above picture, the leading oil and automobile engineers of America were interviewed. They represent the greatest pool of combined technical talent in the world in their respective industries. Opinions were varied and conflicting, as might be expected, but almost all agreed to predict the engineering trend quite accurately if someone could predict the economic and political conditions after the war. The conclusions expressed in this paper are a weighted average of the combined opinions, as all the individual opinions could not be expressed in a paper of this length.

Several factors are not clearly understood by the public, and in most cases this results in mentally transferring wartime aircraft developments to the automobile industry – particularly in engines and fuel.

1. The automobile engineer designs to a *budget* – the aircraft engineer considers *dependability* first, and cost second.

2. Aircraft engines today are no higher in compression ratio than automobile engines – the ratios vary approximately from 6:1 to 6.7:1. Their compression pressures are higher because they are supercharged for *power*, the volumetric efficiency ranging up to 120%; and they must have high-octane fuel for this type of operation, as supercharging increases octane requirement.

3. The aircraft engine operates at a greater load factor than the automobile engine, and nearly at constant speed, due to the variable-pitch, constant-speed propeller. Both these conditions tend toward economy, as opposed to low and fluctuating load factor in automobile driving.

4. Octane numbers for automobile gasolines are listed in two ways by the oil industry: ASTM motor method, and CFR research method. The motor method in general gives a lower octane number to the fuel, and is the rating familiar to the public. Briefly, the engine test conditions are as follows:

Motor Method	*Research Method*
900 rpm, mixture temperature, 300 F	600 rpm, mixture temperature, 125 F

Comparative Octane Ratings on Certain Pre-War Fuels:

	Motor Method	*Research Method*
Premium	80	87
Regular	74	80
Third Grade	65	67

A certain type fuel rating 87 octane, motor method, might be about 95 octane, research method. Thus the statement that 95-octane fuel, research method, will be marketed, does not register with the public as being *87 octane* on the scale they know. This can, and does, cause confusion.

■ Reasons for Variation in Ratings

There are good technical reasons for the different ratings. The motor method penalizes catalytically and thermally cracked fuels, which are high in aromatics. Oil specialists feel that a new and more satisfactory method of rating fuels will be adopted, and suggest that an average of the two present methods is better than either alone. In this paper all ratings will be *motor method,* which is the official method and the one known to the public.

All fuels of the same octane number do not perform the same. In general, gasolines of higher research value compared to motor method value have better antiknock qualities when judged by their performance in actual vehicles. There have been phenomenal developments in fuel manufacture, and each development is raising octane number, as well as producing greater yields from crude. Synthetic catalysts are an advance over natural catalysts. It is possible in the future that the type of fuel produced can be closely controlled, depending upon the type of catalyst used. Fuel is being produced which has potentially twice the power of pre-war fuel, and fuel will eventually be made with one pass, equal to that made by the present two and three passes.

■ New Superfuels

Quoting Mr. Ickes: "When I said that even 100-octane gasoline is beginning to be a little old-fashioned, I was probably thinking more particularly of the catalytic cracking plants that are now opening up in many parts of the country with startling regularity. They come close, in my opinion, to being the last nail in the coffin of the Axis. The new superfuel that they produce has quality factors which make it even better than the 100-octane of a year ago, and the use of it gives the American-made plane the advantage of greatly superior speed and maneuverability. Fueled with it, one of our bombers can carry a 25% greater bomb load. In other words, on a long-range offensive to the Axis capitals, 1000 American-made four-engine bombers with an aggregate bomb capacity of 8,000,000 lb would be able to transport 2,000,000 more lb of explosives on every visit than if fueled with yesterday's 100-octane

gasoline. A two-engine bomber with a 4000-lb bomb capacity can carry another 1000 lb. And that, as someone has observed, 'ain't feathers.' "

5. Aviation fuel is ***not*** obtained directly from the refinery. It is a tailor-made blend, differing with each producer, depending upon his base stock. Many of the ingredients are high in cost, and about $^1/_3$ more crude is needed to produce 100-octane fuel than to produce 80-octane fuel – but for war we are not concerned with cost.

The cost of this fuel is about 4¢ per gal above 90-octane fuel, about 5¢ per gal above 85-octane. This differential will probably be less after the war. 87 octane is a division point in cost, synthetics being generally required above it.

■ Blending Agents

The alkylates are used for good lean-mixture properties; the aromatics for good rich mixture; the isopentane, for high vapor pressure (20 psi Reid) for easy starting, and the lead for knock suppression. Lead is the cheapest way of increasing octane numbers.

Triptane, the paraffinic 2-2-3 trimethyl-butane, C_7H_{16}, is a more powerful knock suppressor by 50% than iso-octane. It has best properties on the lean end of the mixture, good on the rich; and if produced cheaply, should be an important blending agent, allowing the use of lower base stock or replacing some aromatics. It has been very expensive, but recent reports indicate it can be made for less than $1 per gal. One hundred per cent triptane as a fuel is not anticipated in the near future, but with 3 cc of lead it is a powerful fuel to be investigated in long-range research.

■ Manufacturing Problems

The blending agents are not used singly as a fuel, and many of them are so costly they will not be used for automobile fuel after the war. Fig. 1 and the following quotation from Dr. Graham Edgar, Ethyl Corp., indicate some of the complexities of gasoline manufacture:

"Today, we know that the least desirable hydrocarbons for any engine use are the normal paraffins because of their extreme tendency to knock. We know that the isoparaffins are much superior to the normal paraffins and, in many cases, are excellent fuels. We know that olefins, particularly iso-olefins, are excellent fuels for automotive use, especially under low-speed conditions. We know that naphthenes are moderately good automotive fuels, and that aromatics are excellent. We know that the distribution of the various types of hydrocarbon over the distillation range of motor gasoline affects profoundly the service performance of the gasoline. We know that for supercharged aircraft engines, many isoparaffins are excellent fuels for lean-mixture operation, and aromatics are effective under take-off conditions. And, as this knowledge has been attained, it has gradually become technically possible to alter the proportions of any one of these classes of hydrocarbons in gasoline. For example, the simple cracking reaction (Fig. 1) produces gasoline having a higher olefin content than straight-run gasoline. But the actual cracking reaction is never so simple.

"Under varying conditions of pressure and temperature, and particularly in the presence of catalysts, many other changes occur. The larger molecules of the heavy oils are broken into fragments of many sizes, including gases and carbon itself, and the fragments may undergo rearrangement in such ways that the types of hydrocarbon present in the finished gasoline may vary widely from the types present in the original crude oil. Even then, there is much that the refiner may do. The light gases may be built up into liquid hydrocarbons by such processes as polymerization and alkylation, the reactions being directed in such ways as to give maximum yields of specific hydrocarbon types. The crude gasoline (or naphtha) may again be subjected to temperature, pressure, and catalysts (reforming), with the result that further rearrangement of the hydrocarbon molecules may take place. Treatment with catalysts may cause conversion of normal paraffins to the more desirable isoparaffins (isomerization). It is even possible to fractionate out some of the normal paraffins, and by appropriate catalytic treatment convert them into aromatics. Accordingly, the petroleum refiner of today is approaching more and more the position of a synthetic organic chemist, and the gasoline of today is approaching more and more to the status of a blend of specific hydrocarbons, deliberately synthesized for various specific purposes." (See Appendix.)

Much of the loose use of values such as 150 octane number is apparently based on the ratio of

$$\frac{\text{power on fuel in question}}{\text{power on iso-octane (100 octane number)}}$$

It is likely that the difficulty of extending the octane number scale will be overcome in the near future by the adoption of a new scale which will have a value of 75 to 80 at 100 octane number, and which will thus leave 20 to 25 numbers available for expressing the value of fuels which have values in excess of 100 octane number.

It is an interesting point that the gasoline poured into the tank and the rubber on the wheels are different molecular combinations of exactly the same elements – carbon and hydrogen.

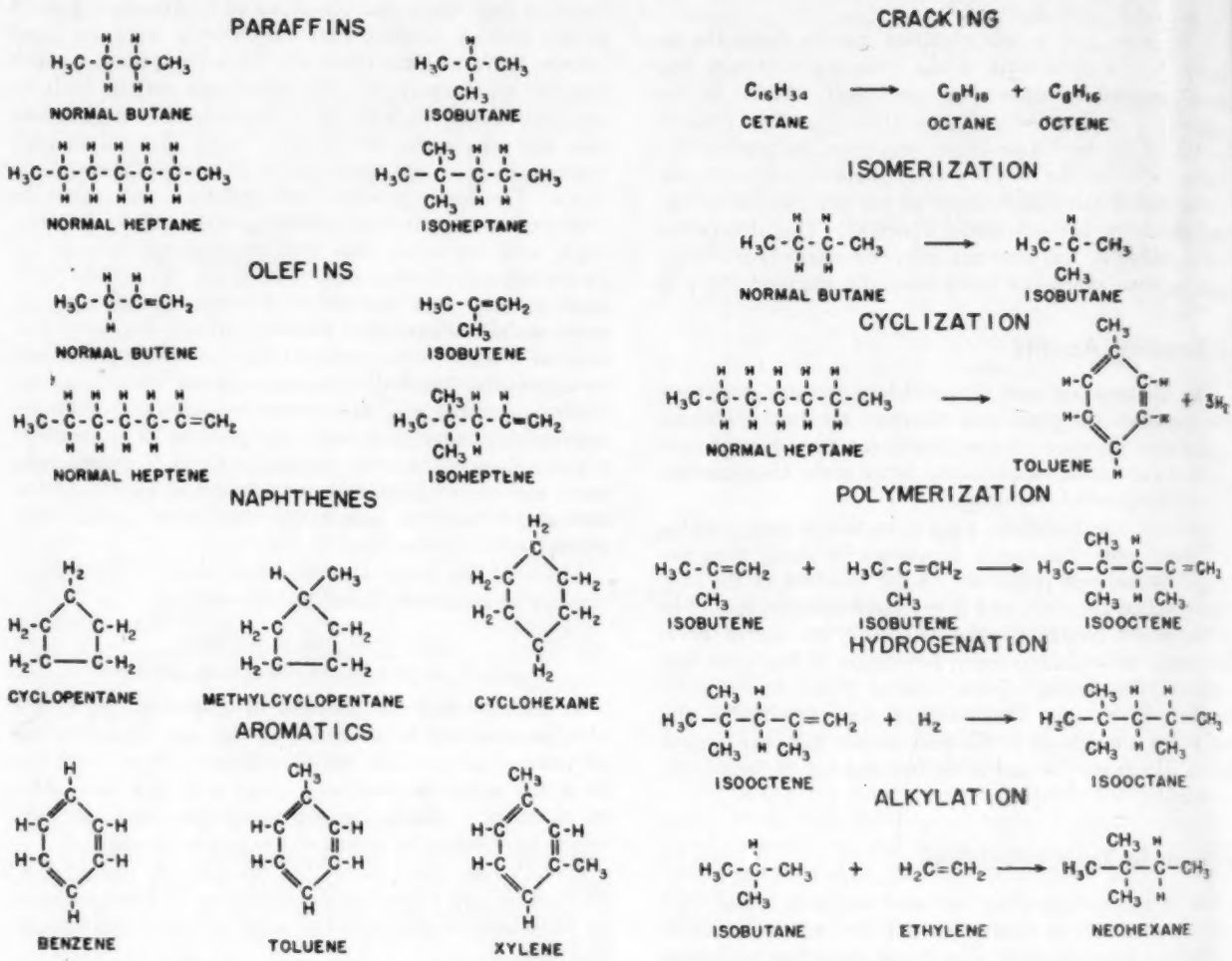
As one oil executive explained: "We can make the *fuel*, but what about the automobile engines?" *There is nothing to be gained in using high-octane fuel in an engine not designed for its use* – that is an economic waste of octane numbers. Greatest efficiency is obtained from fuel when it is on the verge of knocking.

■ Effects of Economic Conditions

It is true that we are in an era of great technical development, but it is misleading to apply these developments indiscriminately without weighing the factors involved. Many of these developments will be used in the future, but their use will be *evolutionary*, not revolutionary. The automobile engineer today is confused as to what trend his design should take, because economic and political conditions may well dictate the engineering trend. As a moving-picture chairman said to his Board of Directors, "For your information, I would like to ask some questions."

1. How has the American standard of living been affected by the war, and how will our post-war program affect it?

2. What will be the post-war demand for aviation gasoline? (The more aviation fuel made, the lower the octane of automobile fuel.)



Typical Gasoline Hydrocarbons.

Typical Refinery Reactions.

Courtesy, Graham Edgar

■ Fig. 1 – Typical gasoline hydrocarbons and refinery reactions

3. What will be the taxes on automobiles and gasoline?
4. What will the cost of producing an automobile be, compared with pre-war production cost?
5. What is the quantity of our oil reserves?
6. What will the dollar buy?
7. How many people will be employed?

These are imponderables, but they are very important. The post-war tax budget will look somewhat like this:

	Billion
Cost of state and county government	13
Federal government – not less than	7
Interest on war debt	7
To retire war debt annually	3
Navy and military	10
Total	40

Pre-war budget was about 20 billion, with a national income of 80 billion. We must pay for the war, which is 70% economic waste, indicating that individuals will have less. The national income must be raised. The dollar will probably buy less. Depending upon the magnitude of this inflation, many people in the fixed-income group may be driven from the automobile market unless cars are specifically designed for that group, but many others will buy cars who never before had them. Automobiles and fuel in 1940 contributed about 13% of all taxes, and automotive products paid 78% of all federal manufacturing excise taxes – they should not go disproportionately higher. Automobile price might change in the ratio of pre-war national income to post-war national income.

■ Post-War Use of High-Octane Fuels

Only a portion of our high-octane fuel capacity – about 10% of our total capacity – will be used by aviation after

the war. The excess capacity will be available for automobiles, *if it can be utilized*. Military requirements are unknown, but will probably exceed transport and private flying combined, the total being many times the pre-war consumption. Transport officials estimate that if all Pullman passengers, all express and all mail carried by railroads in 1939 were transported by air in modern planes, the gasoline consumed would not be over 50,000 bbl (of 42 gal each) per day. Pre-war non-military consumption in 1941 showed the domestic airlines used less than 6000 bbl per day, foreign and territorial about 2000, private and nonscheduled less than 1700, and engine builders less than 500. This totals about 10,000 bbl per day. It is probable that post-war capacity will be at least 10 times total pre-war consumption. It is likely that transoceanic planes will use somewhat better than 100-octane fuel, and domestic planes about 100-octane – this fuel may be alkylate, plus isopentane, plus lead. The present differential of 4¢ per gal between the cost of 100-octane and 90-octane fuel is greater than the difference in value to most commercial operators, as the *cruising* efficiency is only about 5% greater. This price differential will be reduced after the war, and 100-octane has other advantages in power output and maintenance costs. The development of a "safety fuel" to lower fire hazard would be welcomed.

■ Increase in Air Travel

Air travel will materially increase after the war – there were less than 400 transport planes operating in the U. S. before the war, and this will increase to perhaps between 7000 and 12,000 in the 10 years after the war. An appreciable percentage of the planes being made today are suitable for conversion to transports. Post-war private flying is unknown, the estimates running from 5000 planes per year to a total of 450,000 by 1950. 250,000 might be a fair estimate. The 5000-per-year estimate is based on planes of larger carrying capacity, whereas the 250,000 is a total of all planes, including the very small or class I types. The very small planes will probably use automobile premium fuel, whereas the larger type may use better fuel. The effect on fuel consumption will not be great until the numbers reach a high total. The helicopter is in its infancy, and there is great interest manifested in this device, both by the public and by prospective manufacturers. It will likely be perfected to create an industry of large proportions in the years to come – in the next 10 years, 1/3 of the private planes may be helicopters. The man 40 to 50 years old, who can afford a plane, has little business handling a plane of high landing speed. He should, however, be able to handle a simplified helicopter. They will probably use automobile fuel, because of the availability.

■ Problems for Management

Management will be faced with the problem of producing good cars cheaply enough to sell in volume. Wartime emergency rules, if carried over, would critically affect this problem, as would any drop in pre-war worker efficiency. The advances made in machine tools and tooling, manufacturing processes and materials will aid in lowering costs, but could hardly balance present general manufacturing costs. The companies producing the best cars at the lowest price will continue to do business – the best values will always have the biggest market.

■ Depletion of Oil Reserves

Our oil reserves are important from the standpoint of national defense and conservation, and present a highly controversial question. This controversy revolves about the reserves attainable at low cost by pumping from the ground, as against the total reserves regardless of production cost. Many authorities state that a shortage of low-cost oil *which can be pumped from the ground* seems likely over the next 20 years. This would advance petroleum and gasoline prices. We would therefore become an oil-importing nation or turn to other methods of fuel manufacture – manufacture of fuel from natural gas, the high-pressure hydrogenation of crude petroleum, recovery of oil from shale and sand, and the hydrogenation of coal. This war has taught America that materials *are* being expended. The all-time, all-world total amount of oil produced is less than *one cubic mile:* in Utah alone there is enough shale to produce at least two cubic miles of oil, *but at greatly increased cost*. We have ample supplies of natural gas, from which fuel can be made. The hydrogenation of crude petroleum is slightly more expensive than present methods, but yields about 110 bbl of gasoline from 100 bbl of crude, by the use of hydrogen, as against the present yield of 45 bbl by thermal cracking, or 65 to 70 bbl by catalytic cracking and polymerization.

Other authorities feel there is no shortage immediately ahead, that our method of recording reserves is misleading, and that petroleum reserves will increase with increased drilling.

Although opinions vary, *oil is an irreplaceable natural resource*, and good judgment should dictate that we do not waste octane numbers, or fuel in general, needlessly. Any economy attained has the effect of increasing our reserves.

If all the rubber used in this country were made from oil, less than 1% of our oil production would be used, and about another 1% for all plastics and chemicals. Therefore, the bulk of the petroleum is available for fuel.

■ Summary

Summing up the above intangibles, the overwhelming opinion is as follows:

1. The future trend of automobile design will be toward economy rather than performance, because of greater emphasis on petroleum conservation, increased general taxation, inflation, higher fuel costs, and probable higher automobile costs.

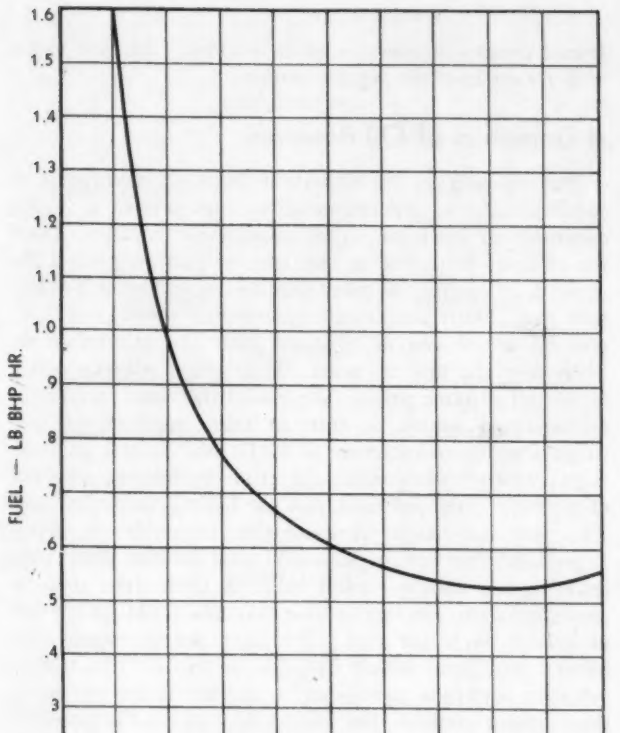
2. The very large automobile is on the wane – the buyers of higher-priced cars have been most affected by taxation.

3. Present comfort, safety, performance, and size of the medium-sized car should not be sacrificed any more than necessary.

4. The very small European-type car is not foreseen, unless absolutely forced by economic conditions, and would not be agreeably acceptable to the American public.

5. Automobiles must be produced with less man-hours.

A tremendous post-war automobile demand is foreseen. Average age of cars in use today is seven years. In former years cars have been scrapped at the rate of 83% of new



Courtesy, Max M. Roensch

■ Fig. 2 – Specific fuel consumption at 1600 rpm – typical passenger-car engine with all accessories

production. On this basis, production of 6,000,000 cars per year would be a net gain of only 1,620,000 and it would take several years to satisfy the backlog at that rate. *The automobile companies will supply the type of car that the public can afford to buy.* Under present conditions, the medium-sized car would be at least 15% over pre-war cost.

■ Improvement in Economy

With the premise of economy, there are several *major* methods of improving economy of operation. Due to the great amount of discussion about high-octane fuels, the general public may be surprised to find that the methods, listed in order of importance, are as follows:

1. *Transmissions and rear-axle ratio.*
2. *Rolling, wind resistance and car weight.*
3. *Engine size – number of cylinders.*
4. *Compression ratio.*

In approaching the problem, the engineer analyzes these factors:

1. If there are several methods of accomplishing a result, *do it the simplest way.*
2. Many times a thing *can* be done, *but does it pay?*
3. Saying a thing *can't* be done means that we do not know how to do it with the knowledge and materials of today – it may be simple tomorrow.

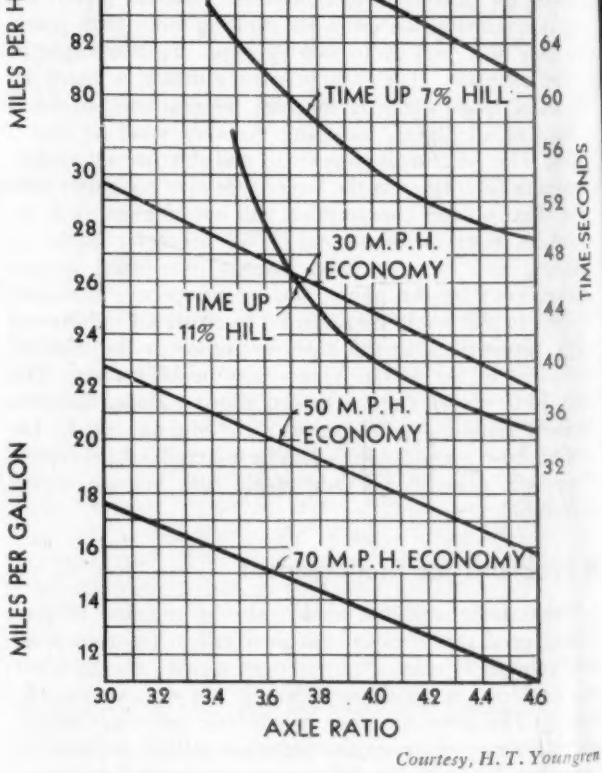
Regardless of speculation on wonder cars of the future, the 1942 automobile was a fine piece of equipment – it had comfort, performance, fair economy, safety, long life, dependability, and low maintenance cost. It is extremely difficult to design for brilliant performance and economy at the same time – the bald-headed man may have less hair to comb, but he has more face to wash. A glance at thermal and mechanical efficiencies indicates that there is room for *tremendous* improvement. For instance, the actual percentage of energy in the fuel delivered to the drivers of a steam locomotive is about 5%, which does not sound impressive. For automobiles, it is approximately as follows:

At speed, mph	%
20	5- 8
40	12-15
60	20

The ascending efficiency is due, of course, to higher load-factor (Fig. 2). Wind resistance, however, lowers mileage at high speeds – wind resistance horsepower goes up as the cube of the speed.

■ Economy versus Performance

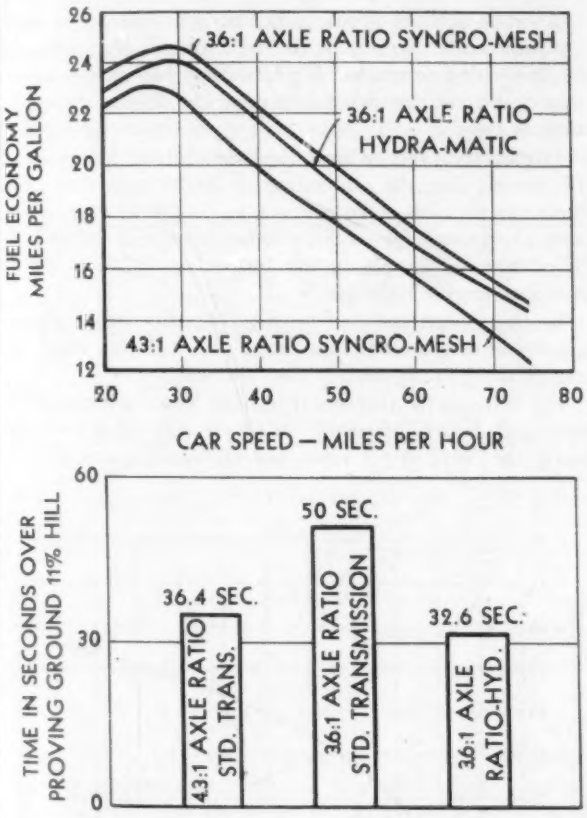
As miles per gallon become more important than performance, the biggest gain can be made by lowering rear-



Courtesy, H. T. Youngren

■ Fig. 3 – Effect of axle ratio on performance and economy

axle ratios (Fig. 3). This sacrifices performance in the low-speed range, which the American driver has heretofore demanded. The overdrive gives economy, but is costly. Automatic transmissions will play an important part in future car development. The hydra-matic and fluid drives are steps in this direction. An infinitely variable-speed transmission is theoretically desirable, but in the present state of the art it is not as satisfactory as the drives in use today. The automatic transmission has not been more economical in city driving, but has been greatly so in country driving, operating at higher load factor and less engine revolutions per mile. The automatic transmission may be compared in action to the variable-pitch propeller in marine and aviation use. A great portion of automobile driving is urban, resulting in low load factor and lack of economy. Passenger-car economy is in the range of 20 to 30 ton-mpg, while heavy-duty vehicles vary from 75 for cross-country buses to 125 for trucks, showing again the advantage of high load factor. Fig. 4 indicates the combination of performance and economy to be explored in future designs.



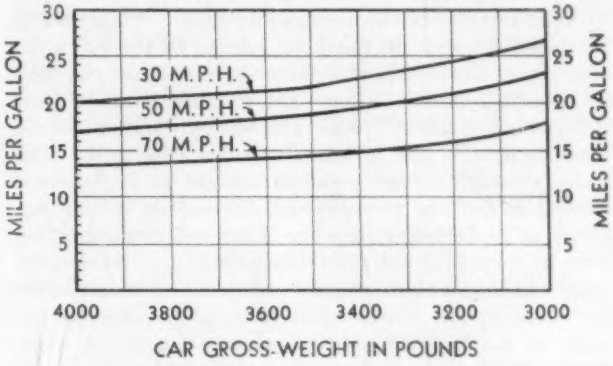
Courtesy, H. T. Youngren

▪ Fig. 4 – The combination of performance and economy to be explored in future designs is indicated in these graphs for a hydra-matic drive of 3.6:1 axle ratio and conventional transmissions of 3.6:1 and 4.3:1 axle ratios: (above) fuel economy plotted against car speed, and (below) hill climbing performance

Reducing car weight improves operating economy, particularly in acceleration and hill climbing (Fig. 5). There will eventually be express highways over America, and good roads to Alaska and South America, on which high driving speeds will probably be allowed. The engineer's task will be to maintain size and reduce weight, at the same time giving good riding qualities and roadability.

■ Use of Steel Will Continue

Steel will likely be the body material because of its physical properties, safety, and price. Aluminum sheet must sell at 6¢ per lb to compete with steel – the price will probably be three times this. Aluminum *can* be used, but does it pay? There are places where aluminum can be used to lighten the car – doors, hoods, and rear decks. Aluminum castings made from secondary metal can compete with iron castings. They can be used for transmission cases, differential housings, and possibly for cylinder heads, crankcases, intake manifolds, wheels, and many small castings and brackets. There is definite possibility that aluminum can be used for cylinder blocks with steel or cast-iron sleeves – this will necessitate the use of constant-clearance



Courtesy, H. T. Youngren

▪ Fig. 5 – Effect of car weight on gasoline economy maintaining the same performance

regulators for valves. There will be a definite trend to aluminum pistons in all cars, designed for improved cooling and controlled expansion. Great progress has been made in the manufacture of aluminum and in handling it. Aircraft manufacturers are machining aluminum at the speed of 7000 ft per min. It is possible that with very little increase in cost, medium car weight can be reduced 300 to 500 lb by the use of aluminum and can further be reduced by redesign of steel parts.

■ Smaller Engine Needed

A corollary of reduced car weight is the smaller and lighter engine. Should the 3000-lb car be reduced to slightly over 2000 lb, obviously a smaller engine can be used and still maintain the same accelerating and hill-climbing ability. Economy comes from raising the load factor. The 4-cyl engine will be found in some cars designed for portions of the market stressing economy and first cost. Changed economic conditions may increase interest in this type of engine.

The radial aircraft engine has been discussed for passenger cars. Present automobile engines cost about 80¢ per hp, as against $10 per hp in the large aircraft radial

engines, and $4 to $5 in the smaller types. Weight is a little over a pound per horsepower for large aircraft, 5 to 6 lb for automobiles. A great weight saving is possible, if installation difficulties are overcome, but again *will it pay?* This type of engine may have a field in expensive buses because of the great saving in weight.

■ Rear-Engine Car Not Foreseen

The rear-engine car appears to have no sponsor in the larger organizations at present. They see no valid reason why the car of the immediate future should have such a design. A transmission and weight-distribution problem is involved, as well as the question of ready public acceptance of such a radical change. New venturers in the automobile field may possibly consider more drastic design changes than those contemplated by the major companies, so far as is known.

Fuel injection does not offer a means to economy on automobile engines, particularly at low load factors. As engine size decreases, the design becomes more difficult. Further, the *cost* of the average automobile engine would be almost doubled. In the diesel engine, all the air in the combustion chamber is at a temperature to ignite the fuel; in the spark-ignition engine, the mixture must be kept rich enough to fire by spark. The economy realized in the diesel cycle at ¼ and ½ load is not realizable in the Otto cycle, although it has some advantages at higher load factors. It can and probably will be used on aircraft engines, as in Germany, but the American carburetor has been so successful that fuel injection is not being used today. In fact, aircraft engines are operating with specific fuel consumption nearly equal to diesel engines, on the order of 0.4 lb per bhp-hr. Fuel injection lowers volumetric efficiency, the cooling effect of the fuel in the intake manifold being lost, but in general, lower-octane fuel can be used with injection. It is likely to be used on trucks, and intensive research work will be done with it. For the immediate future, opinion is unanimous that it will not be used on spark-ignition automobile engines.

■ Limited Use of Supercharging

Supercharging, or positive induction, on aircraft engines is a romantic development. It is used on aircraft engines primarily for *power,* however, and not economy. Two-stage blowers are used further to increase power, the impeller tips traveling faster than a 0.45-caliber pistol bullet, and generating heat that must be removed by intercooling.

While supercharging has received wide usage in the aircraft field, very few attempts have been made to adapt superchargers to automotive equipment. The reason for this has been first, the lack of an efficient variable-speed blower, and second, the low fuel economy which results from supercharged operation. With the development of a satisfactory supercharger, however, the possibility of obtaining reasonable economy with a small engine which is still capable of high power output when required, is a definite one. Such an engine would be designed to operate normally aspirating at part throttle, but supercharged at full throttle. In this fashion the majority of operation would be unsupercharged near full throttle, resulting in low specific fuel consumption. When the need for high power should arise, the throttle would be depressed to the floor and the supercharger would cut in. This idea, far from being a new one, has been incorporated in various Mercedes-Benz models since about 1926.

In actual practice, supercharging has not resulted in fuel economies as good as those obtainable with larger, unsupercharged engines. Nor, in view of the preceding discussion on large and small engines (provided a satisfactory means for transmitting power is available to permit operation at or near full throttle), would any gain in economy be expected. The essential benefit lies in the use of a smaller engine operating at increased load factor; this, however, being complicated by the addition of the supercharger.

Relatively few data have been obtained on the octane requirement and performance characteristics of supercharged engines suitable for automotive use, thereby precluding a quantitative analysis of this problem. Considerable research work will be conducted on this subject in the post-war period, in all probability.

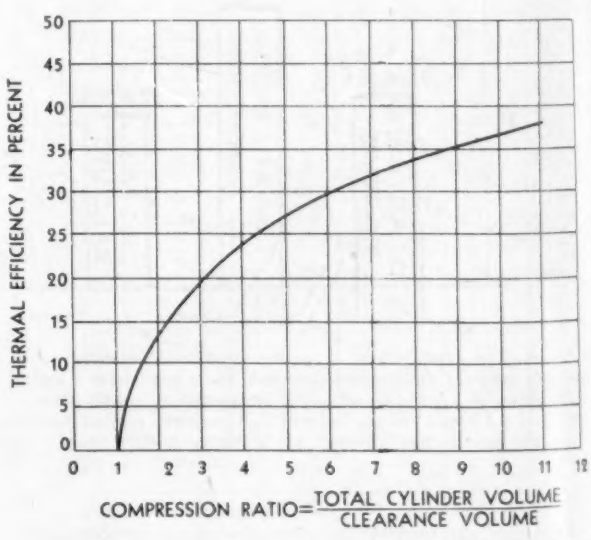
■ Superchargers for Trucks

Average opinion is that superchargers will not be used on automobile engines in the near future, but will very likely be used on trucks. *Higher-octane fuel may be necessary with their use,* dependent upon the degree of boost at various speeds.

Higher compression ratios are the route to higher cyclic efficiencies, and the utilization of high-octane fuel. But high compression ratios are not as simple as they sound, *with the knowledge we have today,* even with better fuel. They may be simple in the future, necessitating almost complete engine redesign.

Engine design and fuel must go together, as the phonograph-and-record combination. Fuel will be ahead of engine design immediately after the war.

Fig. 6 shows the theoretical relation between compression ratio and thermal efficiency. We have traveled a long way along the curve at 7:1 ratio, and the curve is one of defi-

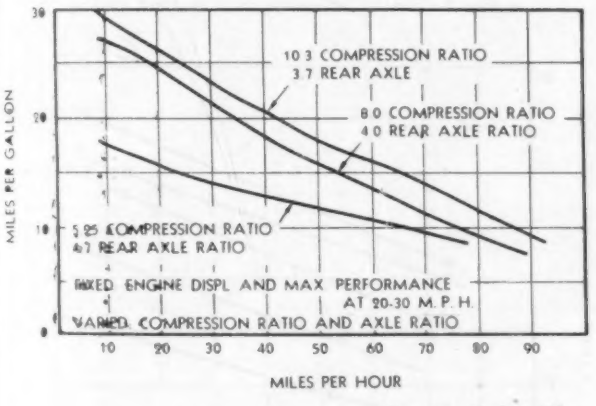


Courtesy, H. C. Dickinson

■ Fig. 6 – Theoretical relation between compression ratio and thermal efficiency

nitely diminishing returns. Mechanical difficulties increase rapidly with compression ratio, principally engine roughness and preignition.

Quoting Dr. Edgar: "In the past, the automotive engine has taken advantage of increasing fuel quality by increasing the compression ratio, which increases both power per cubic inch of displacement and power per gallon of fuel. It is possible that in the future this trend may be the main one which will continue. This trend cannot be followed indefinitely, but it can be pursued very much further than it has been. For example, several years ago, the General Motors Corp. (to which I am indebted for permission to use certain data), carried out an elaborate research project to determine the possibilities of increasing compression ratio under conditions in which the antiknock value of the fuel was no longer the limiting factor. An automobile equipped with a valve-in-head engine was selected for the investigation. The car was operated both on the dynamometer and on the road at a number of compression ratios and a number of gear ratios, and fuels were used which in each case were just capable of avoiding knock. The octane numbers of these fuels are a little uncertain since neither they nor the octane number requirements were determined by the conventional CFR method, but approximately 69-octane fuel was required for the standard 5.25 compression ratio, about 95 octane number for 8.0 compression ratio, and something better than 100 octane number for 10.3 compression ratio. Fig. 7 compares the miles per gallon obtained by increasing the compression



Courtesy, Graham Edgar

■ Fig. 7 – Effect of increase in compression ratio on miles per gallon

ratio when the gear ratio was adjusted until approximately constant low-speed performance was obtained (under these conditions the high-speed performance was better with the high compression ratios).

"The results are striking and show, for example, that at 40 mph the miles per gallon improved from 12.5 at 5.25 compression ratio to 18 at 8.0 and 21 at 10.3. The average increase in economy, between 10 and 60 mph, is about 45% in going from 5.25 to 8.0 compression ratio under these conditions of constant performance. If we take the average retail price of gasoline as 19¢ per gal, the driver of the 8.0 compression car could have paid 27.5¢ per gal at no increase in cents per mile, which would give the refiner a margin of 8.5¢ per gal above his regular cost of 5¢ per gal with which to attempt to produce the 95-octane gasoline required by the high compression car. Such an achievement would appear to be well within the eventual possibilities of the refinery technologist."

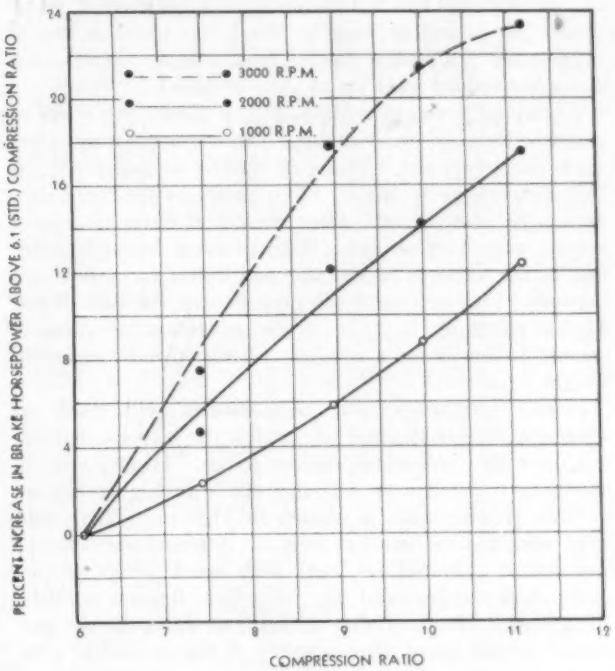
■ Problems of High Compression Ratios

In tests conducted by W. S. James, Studebaker Corp., on an L-head engine at 10:1 ratio, mechanical difficulties encountered were cracked pistons, and increased wear on both compression and oil rings. The power and economy gains in this particular test were disappointing.

When compression ratios go above 8:1, the L-head will likely be replaced by the overhead engine because of breathing ability. The L-head is the cheapest engine to build and is quieter, and will continue to be used below 8:1 compression ratio.

In another test, reported by John G. Wood, Chevrolet Motor Division, an engine was increased in compression ratio from 6.25:1 to 7.5:1 and ran well. The ratio was then increased to 9:1 for the use of 100-octane fuel, and the crankshaft broke. The tests indicated that an engine with high compression ratio could be made lighter in the upper structure, but the lower structure must be heavier. A smaller, higher-speed engine was tested against a larger engine of lower ratio to give the same power; there was no saving in *weight*.

Fig. 8 shows a recent test on a valve-in-head engine of 6.1:1 compression ratio, the engine being changed in steps to give the higher ratios. The chart shows the per cent increase in full-throttle brake horsepower obtained at 1000, 2000, and 3000 rpm with increasing compression ratio.



Courtesy, Ethyl Corp.

■ Fig. 8 – Per cent increase in full-throttle brake horsepower of a production engine with increases in compression ratio. Fuel at high ratios – iso-octane plus 3.0 cc tetraethyl lead per gal

The cost of present automobile motors is about 80¢ per hp. With high compression ratio it is estimated this cost would be over $1.00 per hp.

Smaller bores allow higher compression, because of better cooling. Roughness and preignition are the two main problems to be solved with high compression ratios. *The motor for best economy* is one of larger displacement running at lower speed, or nearer axle speed, rather than smaller, high-speed engines, with high friction and pumping losses. These losses are appreciable – when using the engine as a brake downhill, these are the losses which give the braking effect. After the war there will be a definite trend in aircraft engines toward lowered speed for long-range cruising – the military trend is now toward increased engine speed for *power* not *economy*.

Most automobile driving is done at low load factors with thermal efficiency on the order of 10%, and with specific fuel consumption high on the curve (Fig. 2).

The theoretical economy gain in moving from 7:1 compression ratio to 9.5:1 (requiring near 100-octane fuel) is about 12%. If the performance of the vehicle is maintained constant, an improvement in economy of 10 to 12% will be realized under road-load conditions.

Present Performance for Cars in Perfect Condition

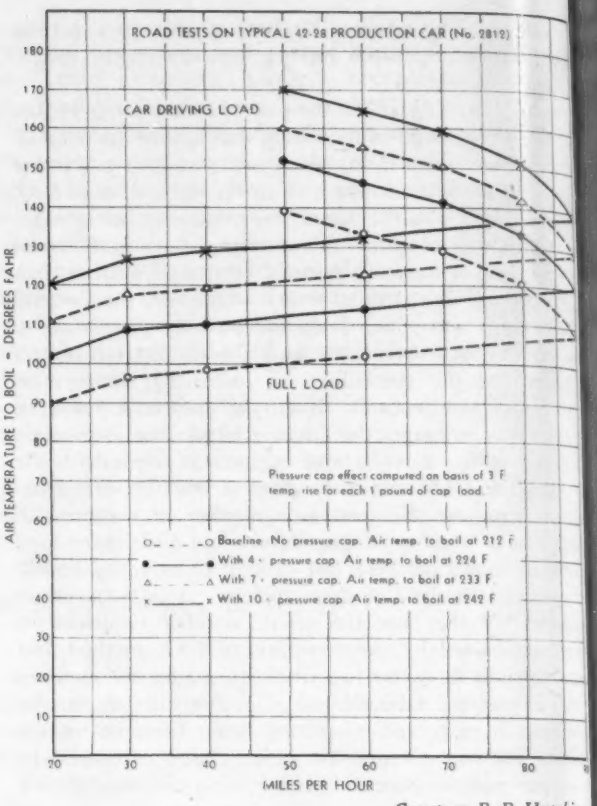
	Weight, lb	Rated Hp	Average Mpg	Top Speed, mph
Willys-Americar	2160	63	30	73
Ford, Plymouth, Chevrolet (Bulk of Market)	2950	91	18	80
Cadillac, Packard, Lincoln	3950	150	10.5	95

These figures tell the story. High roadability, that is, fast acceleration and hill climbing, is a question of horsepower per pound of weight. High top speed is one of horsepower per square foot of frontal area. Wind resistance horsepower goes up as cube of speed.

Regarding pressure cooling, Fig. 9 shows the effect of pressure caps on road cooling. One psi pressure gives approximately 3 deg increase in cooling or specifically, in "air temperature to boil." High pressures are more dangerous to "blowing off" when the cap is removed from a system containing alcohol. High pressure demands attention to the strength of radiator and heater cores and connections. Pontiac used a 4-lb pressure cap, the Rolls-Royce Merlin about 25 lb. One great advantage of pressure cooling is the use of a smaller radiator, thereby reducing weight.

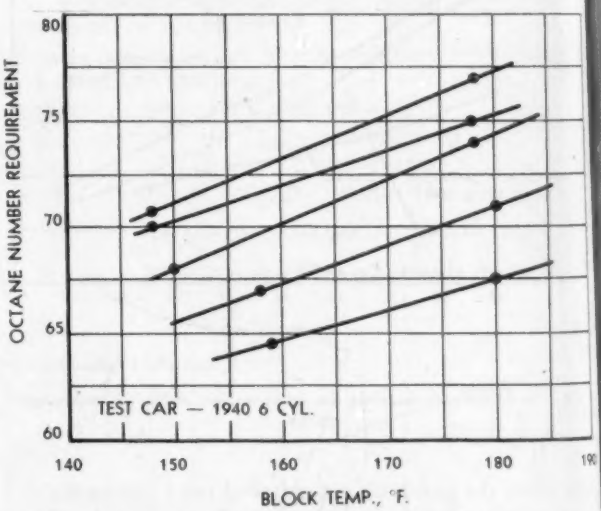
Higher engine-operating temperatures will result in somewhat higher thermal efficiencies, the increase depending upon the individual engine design. Higher engine-operating temperatures will also result in higher octane-number requirements, as shown in Fig. 10. These data were obtained on one car only at different atmospheric temperatures, humidities, and with spark plugs of different heat ranges, and are, therefore, limited in their application. However, they do serve to show, for the particular vehicle tested, that a 10-deg F rise in coolant temperature increases the fuel requirement about ½ to 2 octane numbers. Average opinion is that pressure cooling will be widely used. Research work indicates that water may be discarded as a coolant.

There are several minor ways of producing economy


ROAD TESTS ON TYPICAL 42-28 PRODUCTION CAR (No. 2812)
CAR DRIVING LOAD
FULL LOAD
AIR TEMPERATURE TO BOIL – DEGREES FAHR
MILES PER HOUR
Pressure cap effect computed on basis of 3 F temp. rise for each 1 pound of cap load.
Baseline. No pressure cap. Air temp. to boil at 212 F
With 4 - pressure cap. Air temp. to boil at 224 F
With 7 - pressure cap. Air temp. to boil at 233 F
With 10 - pressure cap. Air temp. to boil at 242 F

Courtesy, R. R. Hutchison

■ Fig. 9 – Effect of pressure caps on road cooling



NOTE: TESTS CONDUCTED AT DIFFERENT ATMOSPHERIC TEMPS., HUMIDITIES, AND WITH SPARK PLUGS OF DIFFERENT HEAT RANGES. EACH CURVE REPRESENTS TESTS AT A GIVEN ATM. TEMP., HUMIDITY, AND WITH THE SAME SPARK PLUGS.

Courtesy, Socony-Vacuum Oil Co., Inc.

■ Fig. 10 – Octane requirement versus block temperature

which will be explored further: induction manifold design, spark advance, valve timing, spark plugs, exhaust scavenging, mixture ratio, and fuel atomization.

Although there are many ways in which performance and economy can be improved, changes will be made only

gradually, and improvements will result through a process of evolution. Fig. 11 shows what can be accomplished in truck-engine improvement by patient development work.

Conclusions

1. Only one fact is certain: immediately after the war 1942 models will be built. By far-sighted policy, automobile companies have saved their dies and tools. Possibly the number of models will be curtailed. Production will start 4 to 6 months after receiving the go-ahead, and some production may be resumed before the European war ends. With Germany defeated, production may start quite heavily. It is likely that there will be large surpluses of material on hand which cannot all be used against Japan. The policy may be to move some Government equipment out of plants and turn the usable surplus material to rapid conversion.

2. No new models will be on the market for at least 18 months, and then *changes will be gradual, not radical.* By its acceptance, the public will actually design the cars, as it always has.

3. Oil companies would like to market two grades of gasoline after the war – premium and regular. Over 1,500,000 farm tractors must be supplied, and a third grade must be available for them and for other equipment demanding it. The hope is that eventually two grades will satisfy all demands. Over 20,000,000 older cars on the road will need slightly higher-octane fuel. Authoritative opinions of executives in the oil and automobile industries indicate that the immediate post-war octane numbers will be as follows:

Product	% of Total	Octane – Motor Method
Aviation	10	100 and 100 plus
Premium	20	85 – 87
Regular	60	75 – 77
Third Grade	10	70

The premium grade will cost about 2¢ extra. It will be made of cracked fuel, thermal or catalytic, straight run, and lead. The cost of aviation blending-agents will curb their use for automobiles (alkylate, 15¢ per gal, aromatics 30¢, others $1 or more). Lead will continue to be used as the cheapest way of increasing octane numbers. There will be less lead per gallon in some blends, more in others. Average lead for premium fuel will be 2 to 3 cc per gal, for regular, 0.5 to 1.5 cc per gal, and third grade 0 to 1 cc per gal. Long-range development may produce powerful fuel, on the order of triptane, which can be economically used.

4. Octane numbers will gradually increase in the future, as fast as engine development can use them. Any development to increase power without increasing detonation tendencies might affect the rate of increase of octane numbers.

5. Compression ratios will go upward. This will be forced by sales promotion, competitive conditions, and limited engineering gain. Even though the returns are diminishing, progress can and will be made. One important factor for higher compression is that at low load, where automobiles operate mostly, the economy gain is

315 CU. IN. ENGINE DEVELOPMENT

1936	CAMSHAFT	COMP. RATIO	CYL. HEAD	CARB.	VENTURI	MAX. HP	MAX. HP SPEED	MAX. TORQUE	MAX. TORQUE SPEED
Standard Engine	Standard	5.25	RE	SF-3	$1\frac{3}{16}$	78.4	2400	213	800
Cylinder Head Change	Standard	5.45(x)	RE	SF-3	$1\frac{3}{16}$	81.1	2400	216	800-1000
Cylinder Head Change	Standard	5.4	RM (x)	SF-3	$1\frac{3}{16}$	79.6	2400	214	800-1000
Camshaft change	X62661(x)	5.4	RM	SF-3	$1\frac{3}{16}$	83	2400-2600	215.5	1200
Cylinder Head change	X62661	5.65(x)	RM	SF-3	$1\frac{3}{16}$	86.3	2600	220	1200
Cylinder Head change	X62661	5.64	RE(x)	SF-3	$1\frac{3}{16}$	85.8	2600	220	1200
Cyl. Head & Camshaft change	Standard(x)	5.65	RM(x)	SF-3	$1\frac{3}{16}$	85	2400	222	1000-1200
Cylinder Head change	Standard	5.64	RE(x)	SF-3	$1\frac{3}{16}$	84	2600	220	800-1000
1937									
Standard Engine	Standard	5.78(x)	RM	SF-3	$1\frac{3}{16}$	87.2	2600	226	1000
Cylinder Head change	Standard	6.0(x)	RM	SF-3	$1\frac{3}{16}$	89.8	2600	231.5	1000
Masked Valves	Masked Valve Shaft	5.66	RM	SF-3	$1\frac{3}{16}$	90	2600	221.5	1200
Cylinder Head change	Masked Valve Shaft	5.85(x)	RM	SF-3	$1\frac{3}{16}$	91.3	2600	223	1200
Carburetor change	Masked Valve Shaft	5.85	RM	SF-3	$1\frac{1}{4}$(x)	94.2	2800	225	1200
1939									
Std. Engine except ratio	Standard	6.0	RM	SF-3	$1\frac{3}{16}$	85.5	2600	225	1000
Carburetor change	Standard	6.0	RM	SF-3	$1\frac{1}{4}$(x)	88.8	2600	229	1000
Camshaft change	X62661	6.0	RM	SF-3	$1\frac{1}{4}$	90.3	2600	230.5	1000
Valve Change*	X62661	6.0	RM	SF-3	$1\frac{1}{4}$	93.5	2600	226	1000
Standard Engine**	Standard	5.9	RM	SF-3	$1\frac{1}{4}$	91.5	2600	232	1000

*Intake valves increased ⅛" in dia. and have 30° seats.
**Built to latest high speed standard. No further development work after this date (1939).

	Standard Camshaft	X62661	Masked Valve
Intake open	Top center	Top center	6° before top
Intake close	35° after bot. center	45° after bot.	61° after bot.
Exhaust open	35° before bot. center	41° before bot.	37° before bot.
Exhaust close	Top center	4° after top	4° after top

■ Fig. 11 – Evolution of truck-engine improvement *Courtesy, B. B. Bachman*

greatest. In the present state of the art, ratios above 8:1 are not foreseen for the immediate future. Gasoline must probably be rated for preignition as well as knock, at higher ratios.

As ratios increase, straight-eight engines must adopt a more compact design to avoid excessive deflections in the engine structure. This may develop into types and shapes more difficult to manufacture and more costly per cubic inch of displacement, raising horsepower cost somewhat.

Designing for higher-octane domestic fuels which may not be available throughout the world would complicate the export situation.

6. The trend in future car design will stress economy rather than performance, but without a *radical* change in performance, medium size, and comfort, unless economic conditions force a lower-cost car to maintain volume production.

7. The medium-sized car will aim at 25-30 mpg, but may sacrifice some performance.

8. Cars will be lighter, light metals being used where cost permits, and parts engineered with the lightness trend.

9. The European-type, small car is not foreseen, but a smaller car with 4-cyl engine is likely to become more popular to meet certain economic conditions.

10. Some cars may be marketed in two models, one for performance, one for economy.

11. Engine size and displacement will not change much with present-weight cars. Engines will be lighter within economical limitations. Engine speeds will not be raised appreciably. Lighter cars will allow the use of smaller engines.

12. Automatic transmissions will receive great attention. Their use on any class of car will be governed by the question: "Does it pay?"

13. Supercharging will be used on heavy-duty vehicles but not on automobiles in the immediate future. New developments may alter this opinion, particularly if *power* and not *economy* should become the dominating factor.

14. Fuel injection will be used on some heavy-duty vehicles, but is not contemplated on passenger-car engines.

15. Pressure-cooling will be widely used.

16. The following engine-parts changes are predicted with higher compression:

(A) Crankshaft – stiffer.

(B) Bearings – improved, aircraft type; thin, soft-metal coatings plated on suitable backing material – will necessitate clean oil; possibly more bearings used, one between each cylinder.

(C) Valves – better material, better cooled; sodium-filled on heavy duty, perhaps on some automobile engines.

(D) Piston rings – big development needed; aircraft practice of chrome-plating likely.

(E) Pistons – aluminum, designed for better cooling, controlled expansion.

(F) Spark plugs – development needed to follow engine changes.

(G) Fuel system – keep lines cooler, perhaps aircraft system of pressure on line to prevent vapor lock.

(H) Combustion chamber – design to reduce tendency toward preignition and roughness.

(I) Higher compression pressures and temperatures will require improved lubricants with better characteristics, such as higher viscosity index, less sludging and oxidation, and more resistance to breaking down under increased pressures and temperatures.

Engineers are optimistic over the possible future developments – they feel that some time will be required to swing from war work to car design and progress. This progress will be gradual and evolutionary, rather than by radical changes in design – the automobile industry has attained its stature by this method of evolution. With an unlimited budget, most extraordinary cars could be produced, but engineers must strike a compromise on cost, performance, economy, comfort, size, long life and dependability, low maintenance, and simplicity. The American automobile industry has been the most progressive in the world – it will continue to be so.

Appendix
Glossary

Acetylenes – Unsaturated chain compounds having a triple bond in the molecule.

Aromatics – Ring compounds having basic 6 carbon atom ring with either 1 hydrogen per carbon atom or a chain substituted for a hydrogen normally joined to the ring. Chains may be saturated or unsaturated. Basic ring formula: C_6H_6.

n-Butane (*normal butane*) – Straight chain normal C_4 paraffin; saturated C_4. Formula: C_4H_{10}.

1-Butylene – Straight chain normal C_4 olefin; unsaturated C_4. Formula: C_4H_8.

2-Butylene – Normal C_4 olefin having double bond in middle of chain. Formula: C_4H_8.

Catalyst – A material which causes a reaction to occur, or determines the products produced. It is normally unaffected by the reaction.

Cyclics – Ring compounds having 2 hydrogens per carbon atom in the basic ring. Basic ring formula: C_nH_{2n}.

Diolefins – Unsaturated chain hydrocarbons having 2 double bonds per molecule.

Ethane – Saturated C_2 paraffin having formula: C_2H_6.

Ethylene – Simplest olefin; technical designation: C_2 unsaturate, or C_2 olefin. Formula: C_2H_4.

Isobutane – Simplest isomeric paraffin; C_4 isoparaffin. Formula: C_4H_{10}.

Isobutylene – Simplest isomeric C_4 olefin; C_4 iso-olefin. Formula: C_4H_8.

Isomer – A space configuration of chain hydrocarbons in which 1 or more methyl groups are offset from the normal straight chain.

Methane – Simplest paraffin hydrocarbon. Formula: CH_4.

Naphthenes – See Cyclics.

Olefins (*mono-olefins*) – Unsaturated chain hydrocarbons having 1 double bond in the compound. Basic formula: C_nH_{2n}.

Paraffins – Saturated chain hydrocarbons having basic formula: C_nH_{2n+2}.

Propane – C_3 paraffin; saturated C_3. Formula: C_3H_8.

Propylene – C_3 olefin; unsaturated C_3. Formula: C_3H_6.

Reforming – Cracking of naphthas to obtain more volatile products of higher octane number. May be thermal or catalytic. Differs from other cracking only in using lighter charge stock and corresponding temperature change.

Saturates – See Paraffins.

Unsaturates – See Olefins.

WAR TIRES—

Availability and Performance

by P. W. DREW
Goodyear Tire and Rubber Co.

A FEW months ago we were trying to make the best tires possible out of as little natural rubber and as much reclaimed rubber as practical. Of course, we were making only truck and bus tires and retreading material for truck and passenger tires. Since that time, the whole aspect of the problem has changed. Our present effort as tire designers and manufacturers is to make the best possible tires out of Government synthetic rubber, known as GRS rubber, which is the well-known butadiene-styrene copolymer. The Government Rubber Reserve Co. is pushing us to build both truck and passenger tires out of this material to the full extent of our manufacturing capacity.

A review of the natural and synthetic rubber situation is necessary to understand our present position. Furnishing the United States military and transportation rubber needs is a problem of great magnitude and probably was entered into with more uncertainty than any other major war program. The public presentation of the synthetic rubber program has been so full of contradictions that it has caused strong suspicions of bad failures. However, now that the program is nearing completion it is evident that no other major undertaking has been so quickly and efficiently handled. The Government and the rubber industries attacked this problem in the following steps:

1. The acquisition of a large enough stock of crude rubber to tide us over until sufficient supply of synthetic could be provided.
2. The selection of one or more synthetics which could be made in adequate quantities.
3. The provision of sufficient manufacturing capacities for the principal ingredients for synthetic rubber.

[This paper was presented at the SAE West Coast Transportation and Maintenance Meeting, San Francisco, Aug. 19, 1943.]

ALTHOUGH many tire manufacturers are claiming that synthetic tires are better than natural rubber ones, Mr. Drew believes this is too much to expect. He feels that production of synthetic tires 85 to 90% as good as those made of natural rubber would be a scientific achievement of the first rank.

The reason for the limitations in the use of present-day synthetic material is that the material has certain deficiencies. The chief faults of GRS Government synthetic are:

1. It develops too much heat when flexed.
2. It loses extensibility when hot.
3. It has low tear resistance when hot.

The development of excessive heat in flexing is due to a high hysteresis loss, which is not critical in articles having a low flex or that are thin-walled, and therefore have a low angle of flex, but in heavy articles such as truck tires, it is one of the biggest deficiencies. Present high-speed trucks and buses tax natural rubber to its full extent, and any additional heat build-up creates more heat in the tire, which greatly reduces the strength of the cotton carcass. One way to alleviate the problem is to substitute rayon or nylon cord for the usual cotton.

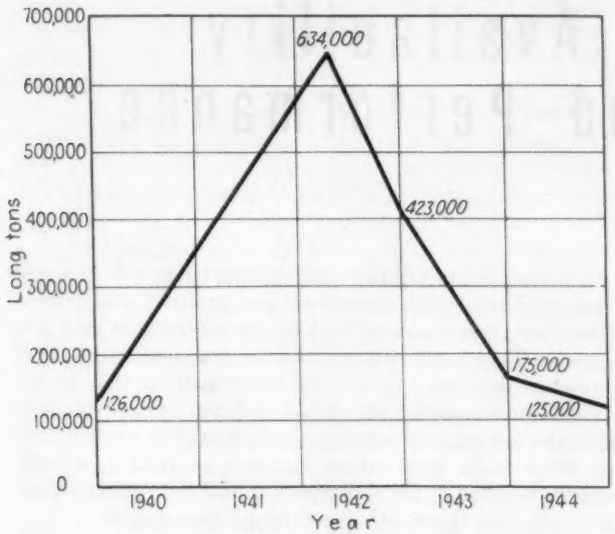
Low tear resistance of the material when hot is also a factor that is causing a lot of trouble in test runs. When the tires are hot, they tend to cut and crack easily – these cuts and cracks having a bad tendency to grow.

THE AUTHOR: P. W. DREW, rubber chemist for Goodyear Tire & Rubber Co. for 17 years, has been chief chemist for the California plant of the company for the last five years. He is also West Coast group chairman of the Rubber Director's Office – Tire Repair Materiel Consulting Committee. A graduate from Knox College, Ill., in chemical engineering, he was recently elected vice-chairman of the American Chemical Society.

4. Provision of sufficient manufacturing capacity for the synthetic itself.

How well these plans have been accomplished is shown in the following charts. These charts should convey the picture to you better than a whole list of quoted figures.

The first chart, Fig. 1, shows the natural rubber stock pile of 1940-1944 and the results of the intensive government program of buying rubber just before Pearl Harbor.



■ Fig. 1 – United States crude rubber stock pile during 1940-1944

The reserve was built up to 635,000 tons before the supply was cut off by the Japanese.

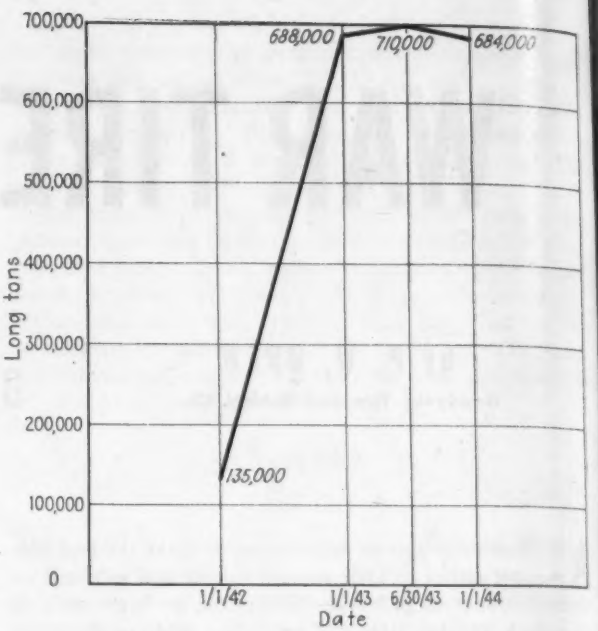
Fig. 2 shows scrap rubber stock pile in 1942-44, and indicates the large amount of scrap collected during the rubber scrap drive. We can now sit back and smile at some of the foolish things that were done during that drive. We in the rubber industry are still wondering what to do with some of the very low-grade scrap collected. However, it did one thing and this one thing probably made it worthwhile. It gave us a positive inventory of the amount of scrap there was for utilization until synthetic rubber was available.

Table 1 shows the reclaimed rubber situation based on industry capacity. This, compared with Fig. 2, shows the large backlog of scrap to be reclaimed. Many wonder why the reclaim capacity wasn't increased more than it has been, but it was correctly decided that it was more efficient to use the steel for equipment and the technical research

Table 1 – Reclaim Situation
(Long tons)

	Production	Consumption	End of Year Stock
1940....	225,000	190,000	34,700
1941....	276,000	254,000	44,300
1942....	285,000	254,000	42,500
1943....	340,000	330,000	42,000

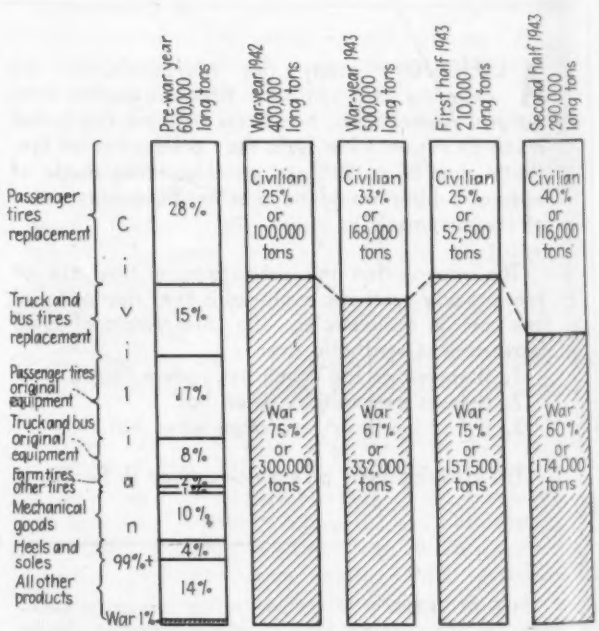
Industry capacity being increased to 30,000 tons monthly by fall of 1943.



■ Fig. 2 – Scrap rubber stock pile during 1942-1944

available to produce synthetic rubber rather than divert part of our energy to producing more low-quality reclaimed rubber.

Fig. 3 shows the consumption of crude rubber broken down into civilian and war requirements. This chart shows the enormous increase in military requirements of rubber. Military products, such as tanks, airplanes, and battleships, require enormous tonnages of rubber products, although none of this is in evidence in the finished war machine.



■ Fig. 3 – Consumption of crude rubber by product and use, in terms of per cent for each period – includes synthetics but not reclaim. Reclaim consumption: 1941: 254,000 long tons; 1942: 257,000 long tons; 1943: 330,000 long tons (estimated)

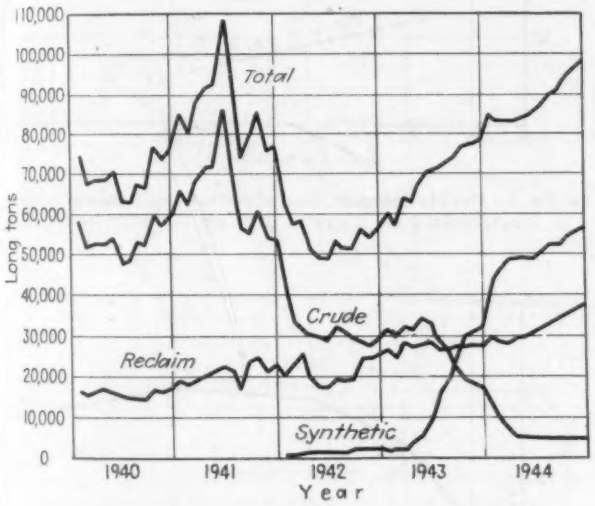
Table 2 shows production of synthetic rubber in 1943 and anticipated production for the first quarter of 1944.

Table 2 – Synthetic Production
(Long tons)

First quarter 1943	10,480
Second quarter 1943	30,640
Third quarter 1943	83,950
Fourth quarter 1943	150,150
Year total	275,220
First quarter 1944	195,750

The rubber industry feels justly proud of the enormous amount of rubber being produced. It should be pointed out that with 1000 American workmen, a synthetic rubber factory can produce as much rubber as 20,000 laborers in the Far East cultivating and milking rubber trees.

Fig. 4 shows the total consumption of rubber in the United States and its breakdown as natural rubber, reclaim, and synthetic to be produced in 1944, compared

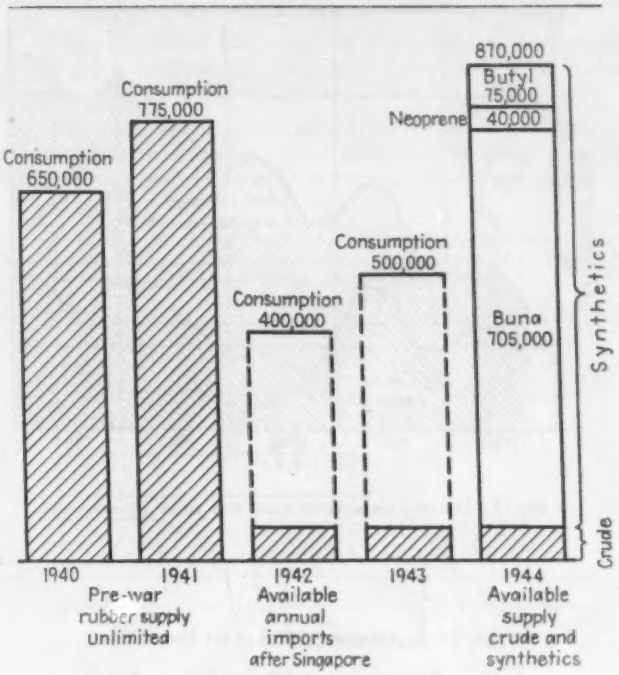


■ Fig. 4 – Total rubber consumption of United States

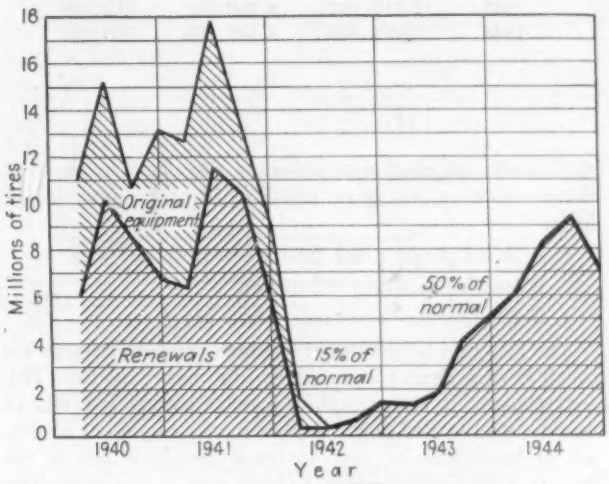
with crude rubber to be used. This crude rubber usage is the amount of natural rubber now being received. This includes the rubber from non-occupied rubber areas, the biggest producing area being the island of Ceylon, along with the rubber being received from the South American Amazon area and the guayule rubber being produced in Mexico and California.

Fig. 5 shows the increase in synthetic production from 1942-1944 and the breakdown of the 1944 production into buna, neoprene, and butyl.

Fig. 6 shows the past production and anticipated passenger tire production from 1940-1944. This chart indicates that although our hopes should not be too high, there should be a lot of nonmilitary tires available in 1944. Our program of tire conservation started early in 1942 was based on making the present tires last for two years. Therefore, it is now our problem to replace the tires that have



■ Fig. 5 – An American production and technical miracle in two years



■ Fig. 6 – Industry pneumatic passenger-car tire sales (quarterly)

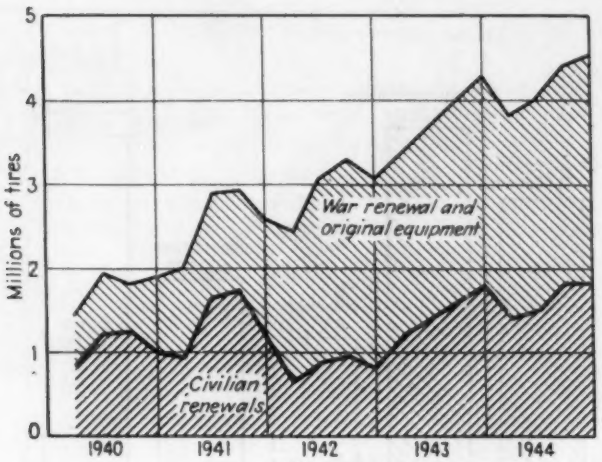
been carefully conserved to extend their life until synthetic tires are available.

Fig. 7 shows similar data for truck and bus tires.

Table 3 shows the new tires available in the years 1940-1944. Available passenger tires will reach pre-war levels while truck and bus tires will reach a new high volume.

As shown in Table 3, present plans for synthetic tires are 30 million in 1944. This means many civilian tires.

This brings us to the consideration of how good these synthetic tires are going to be. Let us look into the characteristics of the base materials, butadiene-styrene, or GRS synthetic rubber. The comparison of characteristics of natural rubber and synthetic rubber will give us a basis for the use of synthetic rubber in tires and other automotive uses. To look the facts in the face, we should list the



■ Fig. 7 – Industry pneumatic truck tire sales (quarterly)

Table 3 – Replacement Market for New Tires

Year	Passenger	Truck and Bus	Farm
1940...	31,040,000	4,305,000	260,000
1941...	34,180,000	5,550,000	385,000
1942...	2,633,000	3,300,000	210,000
1943...	12,000,000	6,000,000	610,000
1944...	30,000,000	6,500,000	700,000

chief faults of synthetic rubber or the places where synthetic rubber is not quite equal to natural rubber as follows:

1. It develops too much heat when flexed.
2. It loses extensibility when hot.
3. It has low tear resistance when hot.

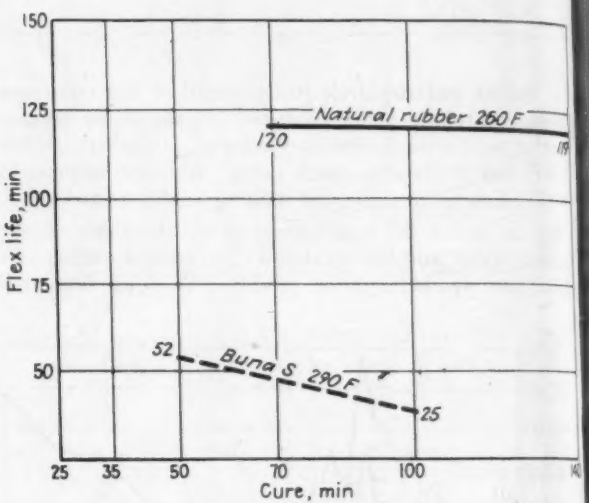
The fact that Government synthetic rubber develops too much heat in flexing is due to its high hysteresis loss. This characteristic is not critical in articles having low flex or that are thin-walled and therefore have a low angle of flex, but in heavy articles, such as truck tires, it is one of the biggest deficiencies we have to learn how to live with. The present high-speed trucks and buses tax natural rubber to its full extent, and any additional heat build-up creates more heat in the tire, which reduces the strength of the cotton carcass to an alarming degree. One way of combating this is to substitute rayon cord for cotton cord, and this has been done in rubber tires for so-called "murder runs." If we used it on all truck and bus tires it would only compensate for the additional heat build-up of synthetic rubber and, therefore, we would lose the ability to build a tire for very hot service conditions.

Low tear resistance when hot is also a factor that is causing us a lot of trouble in test runs. The tires cut and crack very easily when hot. These cracks or cuts also have a bad tendency to grow.

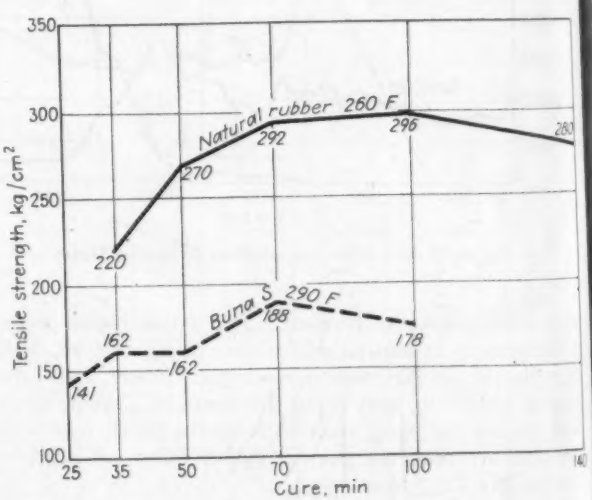
The next group of charts represents an attempt to show the important characteristics of GRS or Government synthetic rubber. It is hoped that these charts will present enough data so that an automotive engineer will be able to use them to estimate the service he may expect when changing from natural rubber to synthetic in any particular rubber usage.

Fig. 8 shows the flex-life comparison of synthetic and natural rubber. The minutes life in flex of buna S is only about one-half that of natural rubber. The results of this shorter flex life in terms of tire life are self-evident.

Fig. 9 shows the tensile characteristics of natural rubber and buna S. Here again the tensile strength is only 65% that of natural rubber. This, of course, is extremely important in tire manufacturing, but it appears that this characteristic can be overcome better than other shortcomings.



■ Fig. 8 – Flex-life characteristics of synthetic and natural rubber-black stocks, 0-116% stretch, 370 cpm (50 parts loading)



■ Fig. 9 – Tensile characteristics of synthetic and natural rubber-original properties (50 parts loading)

Fig. 10 shows the elongation characteristics of buna S compared with natural rubber. The elongation is important in that it affects the tear resistance of carbon-black stocks at room temperature, shown in Fig. 11. At normal cure or state of vulcanization, buna S has only approximately 18% tear resistance compared to natural rubber.

This is the greatest shortcoming of synthetic rubber as far as tire production and service are concerned.

Fig. 12 shows the tear resistance of carbon-black stocks at 246 F. This chart shows that buna S has only 5% the tear resistance of natural rubber at these temperatures. This shows clearly why synthetic rubber tires tear and crack easily at high temperatures.

Fig. 13 shows the effect of temperature on tensile strength in carbon-black stocks. Here again buna S is very deficient.

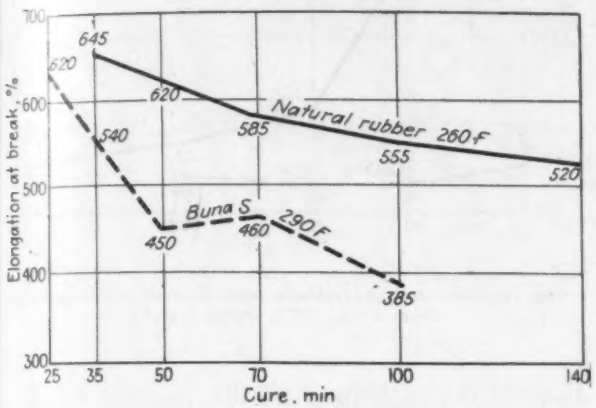
Fig. 14 shows the effect of temperature on the elongation of carbon-black stocks. The difference between buna S and natural rubber is not so marked in this characteristic.

Fig. 15 shows the effect of low temperature on synthetic and natural rubber. This is one field in which synthetic rubbers excel, which explains their desirability in airplane parts manufacture.

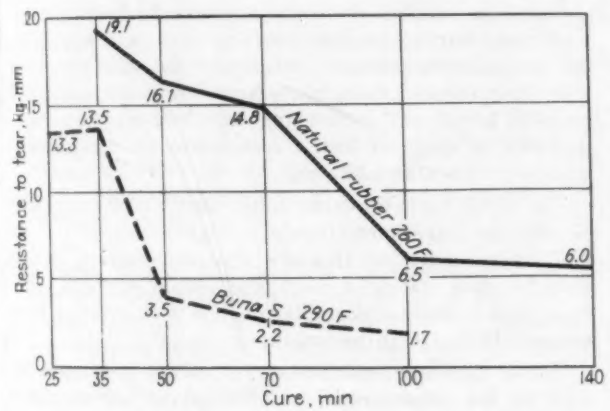
Fig. 16 shows the percentage of rebound of buna S compared with natural rubber at zero, room temperature, and 90 C. Rebound characteristics of the two materials are fairly even.

Fig. 17 shows the deflection characteristics of carbon-black stocks. Here again there is little choice between natural and synthetic rubber.

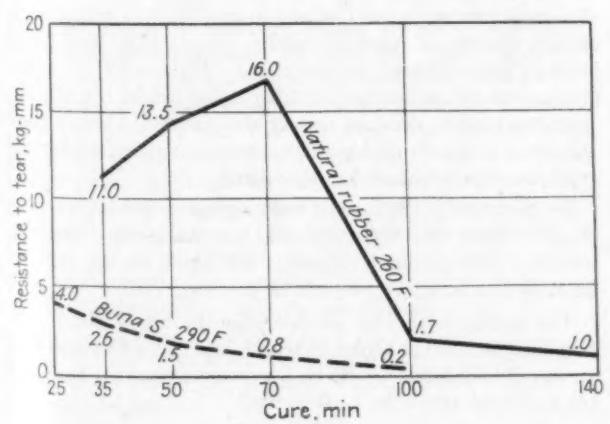
Fig. 18 shows the zero set or permanent set at 0 C of natural and synthetic, showing that set characteristics of buna S are much less than natural rubber.



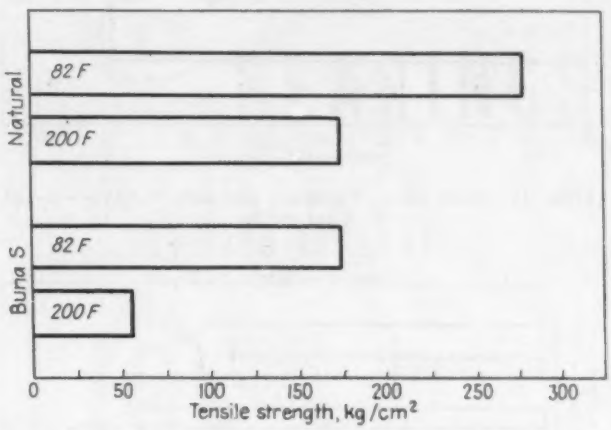
■ Fig. 10 – Elongation characteristics of synthetic and natural rubber – original properties (50 parts loading)



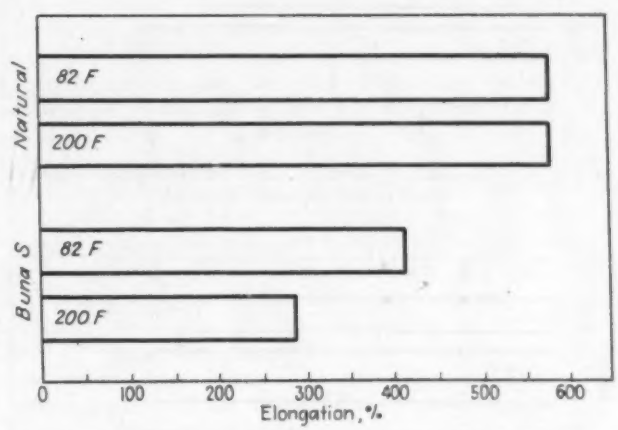
■ Fig. 11 – Tear resistance of synthetic and natural rubber at room temperature – black stocks (50 parts loading)



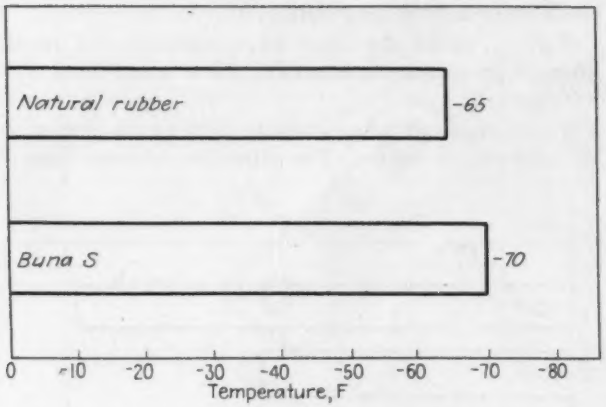
■ Fig. 12 – Tear resistance of synthetic and natural rubber at 246 F – black stocks (50 parts loading)



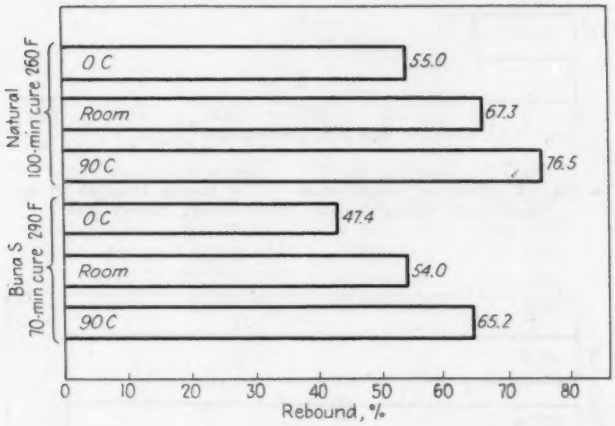
■ Fig. 13 – Effect of temperature on the tensile strength of synthetic and natural rubber – black stocks, 70-min cure



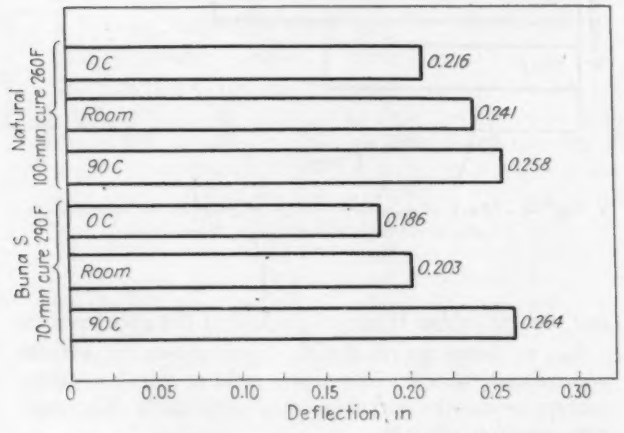
■ Fig. 14 – Effect of temperature on elongation of synthetic and natural rubber – black stocks, 70-min cure



■ Fig. 15 – Brittle point of synthetic and natural rubber – carbon black stocks

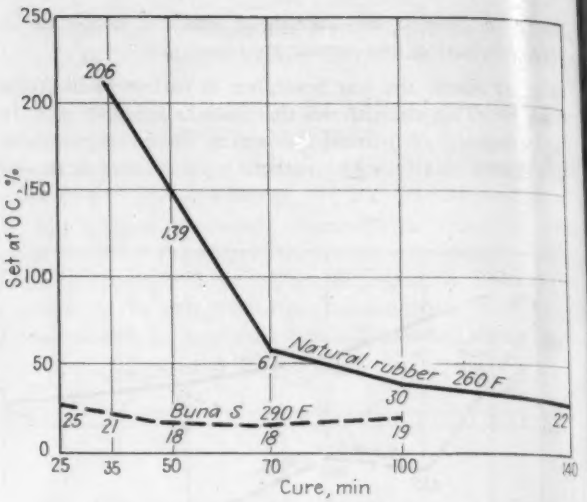


■ Fig. 16 – Rebound of synthetic and natural rubber – carbon black stocks (50 parts loading)



■ Fig. 17 – Deflection of synthetic and natural rubber – carbon black stocks (50 parts loading)

From all these data it can readily be seen that anyone designing a synthetic tire must take into account the many possible deficiencies of the material to be used. Nevertheless, tires have been designed and are being tested and produced which are showing such promising results that many tire manufacturers are claiming synthetic tires are better than natural rubber tires. On the basis of the data



■ Fig. 18 – Zero set of synthetic and natural rubber – carbon black stocks, 300% initial stretch

presented, this seems too much to expect, and with this background our best aim is to make a tire as near the quality of natural rubber as possible. If we can accomplish this aim within 85 to 90%, it will be a scientific achievement of first rank.

The heat build-up of synthetic rubber becomes a more and more detrimental factor as tire sizes increase. This can be somewhat compensated for by the substitution of rayon for cotton fabric, but even with this change the problem is still far from being solved. Large tires will probably be designed with a combination of natural and synthetic rubber for some time.

The use of nylon to replace cotton as tire cord has passed beyond the experimental stage. Fighter-plane tires are now being made with this new cord construction, which is better than cotton in every way – more strength, elasticity, and heat resistance. Rayon cord is only better than cotton fabric at high temperatures.

One of the large peacetime developments will, undoubtedly, be the improvement of tires by the use of nylon cords. This also may turn out to be the answer of how to produce large truck tires out of synthetic rubber.

I have presented the problem as it is today with the standard synthetic rubber that must be produced to obtain designed production out of the approximately 700,000,000 dollars worth of synthetic rubber plants that have been built or that will soon be completed. We can expect many improvements in synthetic rubber to be made, but we are going to have to produce tires of the standard Government synthetic product until present backed-up demand for all types of tires is somewhat dissipated.

By presenting the facts a rather gloomy picture of synthetic rubber has been painted, but we believe the tire industry chemists and engineers are equal to the task of making a satisfactory tire out of this material.

The author wishes to acknowledge the use of chart data prepared by the Goodyear Research and Sales Department, an unpublished talk by R. P. Dinsmore presented June 23, 1943, and an article by L. B. Sebrell.[1]

[1] See *Industrial and Engineering Chemistry*, Industrial Edition, Vol. 35, July, 1943, pp. 736-750: "The Second Mile," by L. B. Sebrell.

A METHOD of Predicting LIFE of Tractor BEARINGS

by JOHN BORLAND
The Timken Roller Bearing Co.

A METHOD of selecting satisfactory bearings for the transmissions and final drives of pneumatic-tired tractors that depends on a knowledge of average operating conditions is reported by John Borland.

The well-known method of rating bearings on a fatigue basis is, according to tests carried out by Mr. Borland, a reasonably accurate way of predicting bearing life when loading conditions are definitely established. However, the tables that have been compiled by the bearing manufacturers are satisfactory for determining bearing life only of bearings subjected to a constant load at a constant speed.

Since tractor transmissions are subjected to as many different loading conditions as there are speed changes provided in the tractor, tractor transmission bearings cannot be selected directly from these tables. They must be used in conjunction with the formula for determining a factor called weighted life. This formula takes into consideration the life factors of the bearing at the different loads and the per cent of time for which each factor applies.

To calculate bearing loads by the author's method, it is necessary to know the weight of the proposed tractor in operating condition, including the operator and water in the rear tires; the speed in each gear; and the tire size. The tangential force on the rear wheels in each speed can be obtained in terms of per cent of total tractor weight from a curve derived by Mr. Borland from data he collected. This tangential force on the rear wheels multiplied by the rolling radius will give the torque on the rear axle. From this torque, the bearing loads can be calculated.

Mr. Borland also discusses other factors that have an effect on bearing life in addition to the imposed loads.

THIS paper presents the results of an investigation made of farm tractor operating conditions. This work was initiated approximately two years ago to determine, if possible, what were the average operating conditions of a farm tractor equipped with pneumatic tires. These tires have less rolling resistance than steel wheels and consequently less power is required to produce the same drawbar pull. Indications are that the general-purpose tractor of the future may use pneumatic tires almost exclusively, and the study will be confined to that type of tractor. It was believed the results would be of value in arriving at bearing selections for transmissions and final drives which would be ample to handle the imposed loads and which, at the same time, would not be unduly large and uneconomical. The paper will discuss only the effect of average operating conditions on bearing selections, although the data obtained may be of value for other purposes.

This study is particularly timely because of the present need of conserving critical materials wherever possible. The use of larger bearings than are necessary is not only uneconomical from the dollar-and-cents point of view, but is also wasteful of the material used in the bearing itself and in its adjacent parts. With the coming of peace, economical bearing selections will become even more important in order to produce tractors which will give satisfactory performance at the lowest possible cost.

Before discussing the data required to make bearing selections in tractors, it would be well to review the basis

[This paper was presented at the SAE National Tractor Meeting, Milwaukee, Wis., Sept. 23, 1943.]

■ ■ ■

THE AUTHOR: JOHN BORLAND, assistant chief engineer of the Industrial Division of the Timken Roller Bearing Co. since 1938, has been engaged in the application of tapered roller bearings to tractors, farm machinery, machine tools, construction equipment, oil field machinery and aircraft. A native of Scotland, he studied mechanical engineering at Penn State, and joined Timken in 1929.

on which bearing ratings are established. The manufacturers of antifriction bearings publish tables giving the load ratings for the bearings they produce. These ratings, which are established on a basis of fatigue, are the result of extensive test work on large numbers of bearings and are generally expressed as so many pounds capacity for so many hours life at a certain rpm. The bearing catalogs give factors for converting the ratings to speeds other than the base rating speed and for converting the ratings for different hours of life. An increase in load or an increase in speed reduces bearing life; conversely a decrease in load or speed increases life.

■ Bearing Load Ratings

All bearings are not rated on the same life basis; there are on the market today bearings whose ratings are established on lives ranging from 500 to 5000 hr. The picture is still further complicated by the fact that "hours life" as given by one manufacturer does not necessarily mean the same thing as "hours life" given by another. For example, tapered roller bearings are rated on a basis of 3000-hr B-10 life. That is to say, out of a large group of bearings operating under specified loads and speeds, not over 10% of them will fail before the 3000-hr point is reached. Ninety per cent of the group of bearings will operate in excess of the 3000-hr figure and the life of the average bearing in the group will be approximately five times the 3000 hr, or 15,000 hr. Other types of bearings are rated not only on a B-10 basis, but in some cases on an average life. In these instances, the catalog rating as published is the rating at which the average bearing of the group will fail. Catalog ratings of different bearings, therefore, cannot be directly compared. The basis of rating must also be considered before making bearing selections.

■ "Spread" of Fatigue Failures

It is well known that in any study of fatigue failures in a large group of seemingly identical parts, there will be a considerable difference in the number of cycles operated between the piece which fails first and the piece which fails last. This fact is frequently overlooked, but it holds true not only for bearings but for gears, shafts, and other parts subject to repetition of stress. This characteristic of fatigue failures can be compared to that encountered in human life. From actuarial tables it is possible to predict how many men out of a large group will die within the next 10 years and how many will live to a ripe old age. However, it is impossible to tell just which of the men will die prematurely and which will still be living beyond their normal life expectancy. Most manufacturers of parts subject to fatigue are striving to reduce this spread or scatter between failures. However, it does exist and is a factor that must be taken into consideration by the engineer. The question as to whether bearing selections should be made on a B-10 life basis or on an average life basis must be decided by the individual designer, taking into consideration the characteristics of the machine on which he is working, the service to which it will be subjected, and the difficulty of replacing failed parts.

The actual procedure in calculating bearing loads and lives is a relatively simple matter when the power going through a mechanism is known. Let us consider a tractor transmission. Knowing the power and speed, the tangential forces and separating forces on the gears can be calculated. If spiral bevel or helical gears are involved, the thrust loads can also be computed. Knowing the gear loadings, moments are taken about the bearings and the radial load on each bearing established. When a bearing is subjected to both radial and thrust loads, it is necessary to determine an equivalent radial load. The equivalent radial load is a value which, were it applied to the bearing as a straight radial load, would result in the same fatigue conditions as the application of the combined radial and thrust loads. The rating of the bearing at its operating speed is then obtained from the catalog data and divided by either the imposed radial load or the equivalent radial load. The result is a life factor. By referring to the published bearing data, it is possible to determine the number of hours life to which this life factor corresponds. It should again be emphasized that the life factor on one make of bearing cannot be directly compared to the life factor on another make of bearing without taking into consideration the basis on which the life of the particular bearings under consideration are established.

■ Determination of Weighted Life

The foregoing illustrates the procedure in determining the expected bearing life when the bearing is subjected to a constant load at a constant speed. Such a condition rarely if ever applies to the bearings in a tractor transmission. Generally speaking, a transmission bearing will be subjected to as many different loading conditions as there are speed changes provided for in the tractor. This is due not only to the variations in speed but also to the differing locations of meshing gears. Under such conditions, it is necessary to determine weighted life, taking into consideration the different loadings under which the bearing will operate. The method is to check the loading under each condition to determine the life factor. Taking into consideration the various life factors and the percentage of time for which each applies, a weighted life may be calculated.

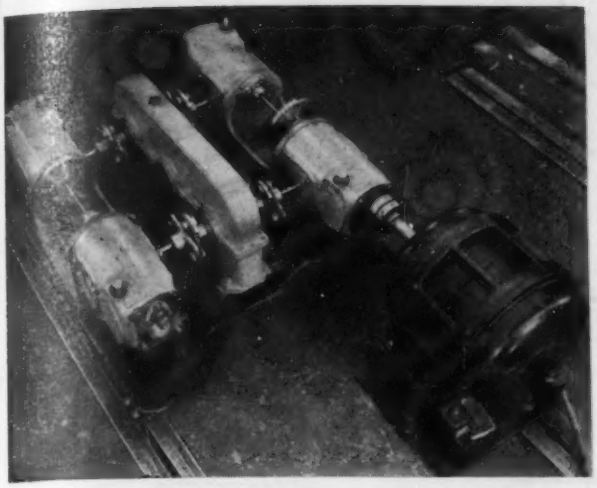
For tapered roller bearings, the weighted life is given by the formula:

$$\text{B-10 Life} = \frac{3000}{\frac{T_1}{L.\,F._1{}^{3.32}} + \frac{T_2}{L.\,F._2{}^{3.32}} + \cdots}$$

In the above formula, T_1 is the percentage of time for which Life Factor$_1$ applies, T_2 is the percentage of time for which Life Factor$_2$ applies, and so on. The 3000 constant in the formula is derived from the 3000-hr B-10 rating of tapered roller bearings. The 3.32 exponent is used since tests indicate that the life of these bearings varies inversely as the 3.32 power of the load. It should be emphasized that the life obtained from this formula is B-10 life; the life at which not over 10% failures can be expected. This formula applies only to tapered roller bearings but similar formulas for other types of bearings can no doubt be obtained from their manufacturers.

■ Test Procedure

As a demonstration of how bearing life can be predicted when the operating conditions are known, a 4-square test



■ **Fig. 1 – Four-square transmission test**

This test is used to predict bearing life when operating conditions are known. Four standard tractor transmissions are mounted so that gear loads can be applied to all of the transmissions in the same manner as though they were operating in forward speed

set-up was made as shown in Fig. 1. Four standard tractor transmissions are mounted in such a manner that gear loads can be applied. The arrangement is such that all four of them are loaded in the same manner as though the transmissions were operating in forward. The set-up is shown diagrammatically in Fig. 2. The couplings between the input shafts on these transmissions are "wound up" against the mesh of the gears in the spur-gear box. This imposes loads on the gears and in turn on the bearings similar in nature to those obtained in actual field service. These units are loaded to a known torque and from this, the expected bearing life can be calculated. The torque used is approximately 75% of the rating of the motor normally used with these units. Arrangements have been made to test four different sizes of bearings at the position immediately ahead of the bevel pinion on these transmissions. These four sizes of bearings are all considerably smaller than those used in this unit in actual service and are mounted in the transmission cases by the use of suitable adapters. The smaller sizes of bearings were used in order that bearing failures could be obtained in reasonable periods of time.

■ Tests Corroborate Theory

The test procedure is first to calculate the expected B-10 life on the bearing being tested from the known torque and speed. The transmissions are torn down periodically for inspection and a careful record is kept of the bearing failures and of the hours operated. To date, 50 bearings of two different sizes have been tested and 26 failures, or 52%, have occurred at an average of 5.73 times the B-10 life calculated. The ideal, according to the theory, would of course be 50% failures at five times the B-10 life. In this test, a bearing is considered to have failed when it has a total spalled area on the rolling surfaces of 0.1 sq in. As would be expected, all of the failures have been by spalling of the rolling surfaces, which is characteristic of bearing fatigue. A test of 50 bearings is not conclusive evidence. The test is, therefore, being extended to secure further data. However, the test of the 50 bearings does indicate that when the loading conditions on a bearing are known, it is possible to make a reasonably accurate prediction of the expected bearing life.

■ Other Factors Affecting Bearing Life

It is recognized that there are factors such as bearing mountings, alignment, deflections, lubrication, and cleanliness that enter into the life of bearings in addition to the imposed loads. Several of these factors can be minimized or eliminated by proper design, while others are beyond the control of the manufacturer. With particular reference to tractor transmissions, let us consider some of these points.

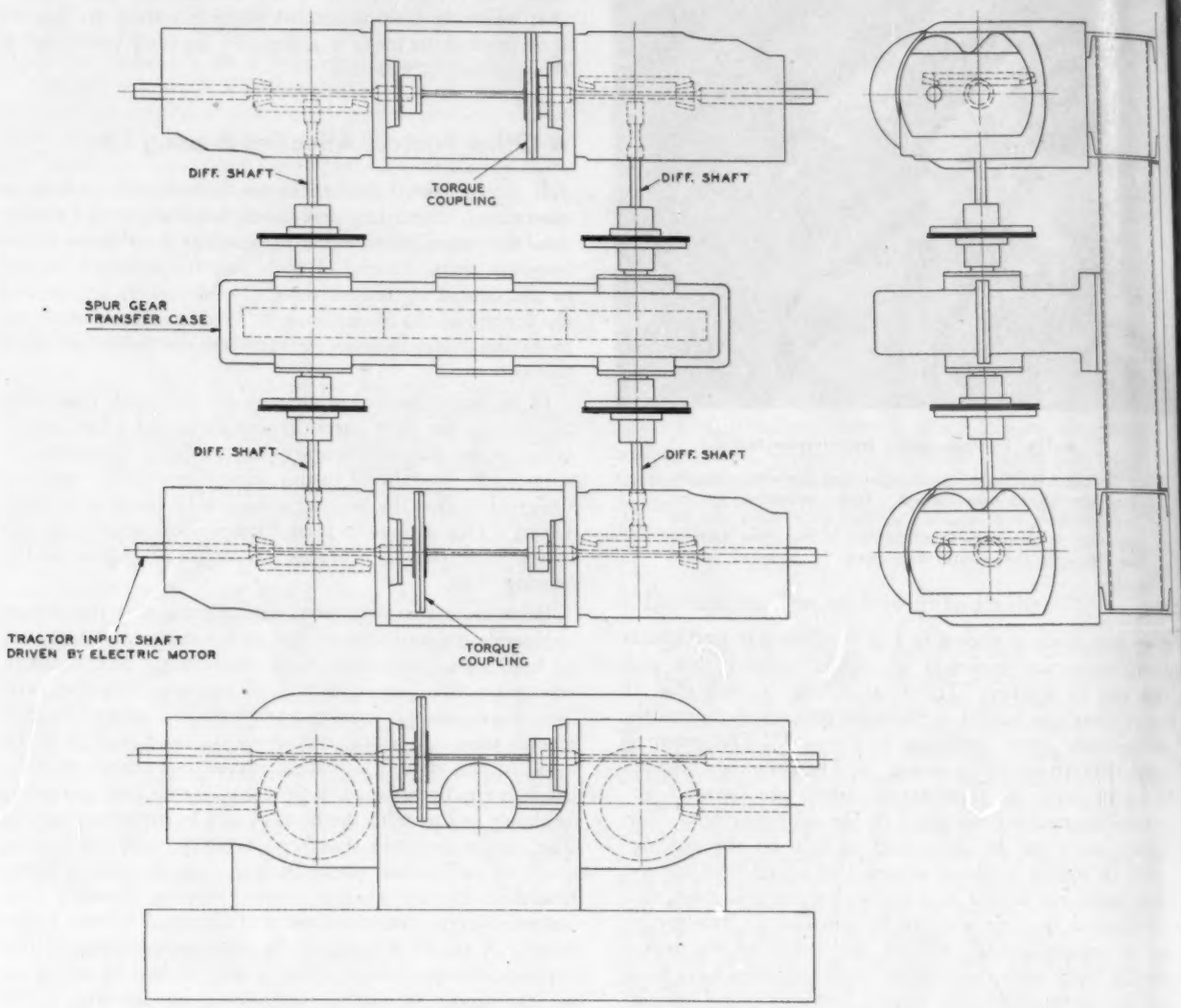
In so far as bearing mounting is concerned, practically all tractors are very carefully designed and great care is taken to see that the bearings are properly mounted and supported. In addition to this, most transmission cases are designed so that the bearing seats can be bored at a single setting. This is done to permit economical machining, but at the same time it provides for correct alignment of the bearing seats.

With regard to deflections under load, it is the tractor engineer's responsibility to design his shafts and housings to hold these deflections to a minimum. The industry recognizes this responsibility. Numerous deflection tests have been run on tractor transmissions during the past several years. A typical test set-up is shown in Fig. 3. In a test of this nature, a bracket supporting numerous indicators is rigidly fastened to the transmission case at a point as closely in line with the bevel pinion bearings as possible. The unit is then run slowly under 25%, 50%, 75%, and 100% of full motor torque in low gear; the power being furnished by an electric motor driving through two automotive-type transmissions and absorbed by the brake shown. A record is made of the movements shown by the various indicators. Gear contacts are checked by red leading the teeth. In the transmission shown in Fig. 3, 28 different indicators were used to measure deflections at all critical points. Different transmission designs require more or fewer indicators, depending on the number of points to be checked.

Deflections must be held to a minimum to obtain calculated bearing life and proper gear contacts. These tests show that, generally speaking, tractor transmissions show rather low deflections under load compared to similar units used in other industries. It is also interesting to note that none of the bearings in the 4-square test have shown any evidence of having operated in a deflected or misaligned state. The writer's opinion, based on these deflection tests and on actual field service, is that deflections in tractor transmissions do not enter greatly into bearing life.

Lubrication is the third factor affecting bearing performance. In tractor transmissions it is a negligible factor in bearing life since practically all units are designed to supply the bearings with ample oil. Suitable troughs are usually provided to catch the lubricant thrown by the gears, convey it directly to the bearings, and thereby provide almost perfect lubrication.

A fourth factor, one over which the tractor manufacturer has little control, is operating cleanliness. It is im-



■ Fig. 2 – Layout of 4-square transmission test

possible to predict bearing life if the operator permits dirt and grit to enter the transmission case, to lap the gears and bearings. However, a tractor transmission is just about as well protected from dirt and grit as is any piece of mobile machinery. All openings into the case are provided with suitable oil seals and every precaution is taken in the design to prevent the entrance of foreign material. In addition, all the openings are located high enough above the ground level so that they are not subjected to the dirty conditions found, for example, around front wheels. Increasing the size of a bearing does not necessarily ensure longer life of the bearing if dirt is present. Therefore, in selecting bearings for tractor transmissions based on loading considerations, it must be assumed that the operator will keep the lubricant in the transmission case reasonably clean. If he does not do so, both bearing and gear trouble can be anticipated.

Methods of calculating bearing loads in tractor transmissions used in the past have resulted in bearing selections that have given satisfactory field service. None of these methods, however, have given a true picture of the actual field loads encountered on pneumatic-tired tractors. This is not intended as a criticism of either the tractor or bearing engineers since until two or three years ago, data were lacking for the development of any better basis for bearing selection. These methods will be reviewed to show why the results obtained did not reflect actual field conditions.

■ Review of Older Methods

One of these methods based the bearing loadings either on the rated torque of the motor or on some arbitrarily assumed percentage, usually 80% or 85%, of this value. The calculation was made for second gear since it was assumed the tractor did most of its heavy work at this

speed. In many cases, bearings calculated by this method showed theoretical lives which were extremely low. Nevertheless, these bearings in field service performed very satisfactorily. It is obvious, therefore, that the calculated loads were higher than those usually encountered in farm work. It is also well known that there are few tractors on the market today that are heavy enough to use full motor torque in the lower speeds without adding considerable weight. Therefore, while bearing loadings may be at times of the same magnitude as those caused by full motor torque, this is certainly not an average condition and should not be used as a basis for bearing selection.

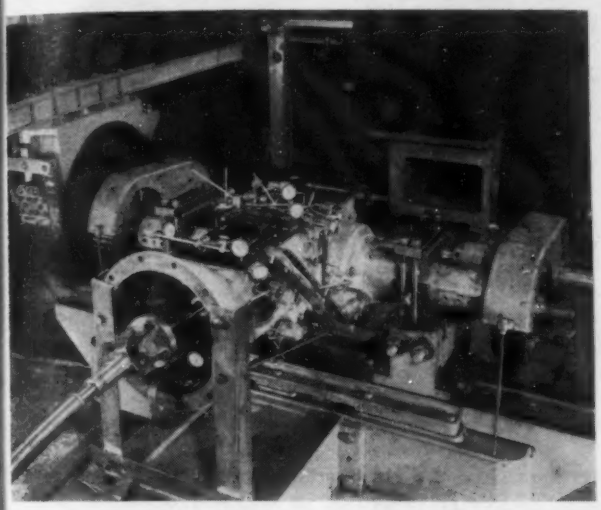


Fig. 3 – Typical set-up for a deflection test of a tractor transmission

Numerous indicators are mounted on a bracket that is rigidly fastened to the transmission case at a point as closely in line with the bevel pinion bearings as possible. The unit is run at varying percentages of full motor torque in low gear, deflection at the critical points being shown by the indicators

A second basis of calculation assumes that the tangential force on the rear wheels of the tractor is equivalent to 66% of the tractor weight. This is, in effect, assuming that the traction coefficient between the tires and the ground is approximately 0.66. Knowing the tangential force on the rear wheels, the torque on the rear axle can be computed and the bearing loads in the transmission figured through from this value. Again, bearing lives so calculated were low and premature bearing failures should have resulted. Despite the calculations, bearings thusly selected have given satisfactory field service. Again the inference is that the loading figured was on the high side. The SAE Cooperative Tractor Tire Testing Committee in 1937 reported that while a traction coefficient of 0.66 can be developed, it is certainly not a normal condition. This report further points out that such a coefficient can be reached only on concrete or dry clay and that the values on other types of soil are considerably lower. Such a calculation, therefore, does not represent a normal condition.

A third method, the one which has been in general use, bases the bearing selections for a new tractor on the performance of bearings in service on current production tractors. If a bearing in service figured on full motor torque shows a certain number of hours life, a bearing for a new tractor figured on full motor torque and showing the same number of hours life should be satisfactory. Selected bearings in this manner will assure designs that will give satisfactory service, but there is no assurance that such bearings will not be larger than required and consequently uneconomical.

■ Data Needed for Thorough Analysis

To make a thorough analysis of bearing life in a tractor transmission, it is first of all necessary to know the percentage of time the transmission is used in its various speeds. It is also necessary to know what percentage of motor power is normally going through the transmission in each of these speeds. Knowing these data, the bearing-life factors in each speed can be computed and the weighted life of the bearings figured by the formula previously given.

■ Tractor Manufacturers Interviewed

Knowing the data that would be required to make such an analysis of tractor bearing loading, all of the leading tractor manufacturers were contacted to determine if such information was available. Some of the manufacturers had rather extensive data which had been accumulated from their test work and others, in addition to such test data, had considerable information which they had obtained from representative farmers scattered over the country. From these data, it was possible to estimate the percentage of time the tractors were run in each gear. Also, from the information available on fuel consumption of tractors in actual service, it was possible to estimate the percentage of full power that was normally used. Unfortunately, in most cases data were not available as to the percentage of power used in each speed since the fuel consumption data were generally in terms of so many gallons per hour over a considerable number of hours operating time. In this time the tractor might have operated in various gears. The data obtained represented information regarding the percentage of time the tractor was used in each gear and the average power developed over a period of time during which the tractor operated in various gear meshes. In preparing calculations, the assumption was made that the tractor used the same power in all gear meshes. It is believed that such an assumption is reasonably close to actual field conditions and will not materially affect the results obtained. Data, which are based on actual records or on well-founded estimates, were obtained on 14 different models of tractors designed by six different tractor engineering departments. All of these tractors have been in production for a period of years and have been successful designs.

With reference to the percentage of time in which the tractor is used in various speeds, most of the manufacturers are in fairly close agreement. This will be discussed in further detail later in the paper. As might be expected, estimates on the percentage of full power that normally goes through the transmissions varied widely, ranging from 50% of the maximum motor torque to 75% of the maximum drawbar horsepower as developed on the Nebraska test. Practically all of the engineers interviewed agreed that it is incorrect to figure bearing loadings from the peak motor power, and most of them believed that,

in so far as transmission bearings are concerned, the correct method would be to figure back from the drawbar pull. They further agreed that the weight of the tractor controls the drawbar pull that can be developed on a pneumatic-tired unit.

In making this survey an attempt was also made to check on the number of bearing replacements that have been furnished for transmissions and final drives on these 14 tractors. Unfortunately, definite data on this could not be obtained but it is certain that there has been no epidemic of bearing trouble on any of the tractors studied.

■ Data Tabulated

The analysis shown in chart form in Fig. 4 was prepared from the data furnished by the various manufacturers. The first four tractors listed on this chart are 3-speed designs, while the other tractors have four or more speeds. In the case of these latter tractors, only the four lowest speeds were considered in the analysis since it is known that, even basing calculations on full motor torque, the loads in the higher speeds are relatively light. As previously pointed out, tractors equipped with pneumatic tires have been used throughout in this study, and the results, therefore, are not applicable to steel-wheeled units.

■ Explanation of Chart

The first column in the tabulation, with one exception, lists the maximum belt horsepower, corrected to sea-level conditions, as developed on the Nebraska test. This value was used since it was believed that the maximum belt horsepower as developed on the test was as close to the maximum power output of the motor as any value that could be obtained. The exception to this is tractor No. 7. On this particular unit, the motor is governed at a different speed for drawbar work than for belt work. No data were available on the test regarding the belt power with the motor at drawbar speed. Consequently, the power listed is that given by the manufacturer of this particular unit.

The second column merely gives the horsepower figure converted into torque.

The third column in the tabulation gives the weight of the tractor as used in the computations. This weight was obtained by taking the maximum weight as tested at Nebraska and subtracting from it all added weights. This gives a value equivalent to the weight of the tractor in operating conditions with an operator. To this figure was added the weight of water in the rear tires as given in the tire manufacturers' catalogs. Therefore, the weight figure

Tractor No.	Max. belt H.P. corrected to sea level. (From Nebraska test)	Max. torque in ft. lbs.	Weight, with fuel, oil, water, operator plus water in rear tires.	H.P. per 100# weight	1st Gear				2nd Gear				3rd Gear				4th Gear			
					% of time	Tan. force on wheels	M.P.H.	D.P./Wt.	% of time	Tan. force on wheels	M.P.H.	D.P./Wt.	% of time	Tan. force on wheels	M.P.H.	D.P./Wt.	% of time	Tan. force on wheels	M.P.H.	D.P./Wt.
1	16.31 @ 1400	61.8	2280	.717	15	1130	2.89	.495	70	718	4.08	.315	15	371	7.90	.163				
2	25.81 @ 1500	83.5	2501	.952	15	1760	2.42	.701	70	1110	3.84	.443	15	577	7.40	.230				
3	15.21 @ 1850	43.2	2558	.598	15	1400	2.94	.552	70	993	4.24	.391	15	465	8.42	.183				
4	15.26 @ 1400	57.3	2727	.560	20	1524	2.54	.558	60	1139	3.45	.417	20	714	5.85	.261				
5	25.81 @ 1500	83.5	2501	.952	15	1760	2.42	.701	35	1230	3.47	.490	35	880	4.86	.352	15	452	9.46	.181
6	19.81 @ 1500	68.5	2790	.700	10	1420	2.59	.508	30	1070	3.48	.384	50	825	4.57	.296	10	622	6.30	.223
7	22.10 @ 1400	80.0	3006	.735	20	1350	2.60	.449	35	1310	3.55	.437	35	1110	4.38	.369	10	570	9.85	.189
8	24.20 @ 1400	90.5	3417	.705	1	2075	2.30	.607	39	1440	3.30	.423	57	1105	4.28	.324	3	407	12.6	.119
9	31.45 @ 1500	127.0	3571	.880	15	2200	2.54	.617	35	1500	3.70	.420	35	1150	4.90	.316	15	608	9.40	.170
10	32.86 @ 1500	116.0	4113	.800	0	2800	2.28	.680	13	2350	2.72	.570	47	1680	3.80	.408	40	1330	4.78	.324
11	20.84 @ 1150	93.8	4473	.459	15	2645	2.61	.606	35	1743	3.43	.390	35	1283	4.43	.309	15	1101	9.84	.246
12	25.38 @ 1500	116.5	5096	.498	10	2640	2.63	.518	30	1930	3.60	.378	50	1525	4.70	.300	10	1180	6.38	.230
13	30.98 @ 1[illegible]	157.0	5504	.562	15	3235	2.51	.606	35	2595	3.34	.471	35	2050	4.41	.372	15	1665	8.78	.302
14	44.85 @ 1275	185.0	6432	.700	3	3280	2.75	.512	12	2540	3.55	.396	47	1925	4.70	.300	38	1466	6.28	.227

■ Fig. 4 – Tabulation of data furnished by the tractor manufacturers consulted

as used covers the tractor in operating order, with water in the rear tires and with an operator. This value was used in order to put all of the tractors on the same basis and because it seemed to be the opinion of most of the engineers that a large majority of the tractors in actual service have water in the rear tires but do not have any additional added weights.

The fourth column of the tabulation lists of the power-weight ratio of the tractor expressed in terms of horsepower per 100 lb of weight. Note the wide variation in this value between the different tractors studied. There are several reasons for this wide variation which, however, do not come within the scope of this paper and need not be discussed here. The differences in these values of power-weight ratio, however, do indicate the fallacy of checking transmission bearing loadings on the basis of full motor power when only a portion of this power can be used at the drawbar.

■ Tangential Force Calculated

From these data, the balance of the information on the chart has been derived. Each tractor was figured through in each speed and the tangential force on the rear wheels obtained. This tangential force is figured at the rolling radius of the tire as given by the tire manufacturers. In those cases where the tractor builder furnished information as to percentage of full motor power normally used, this percentage has been taken of the maximum belt horsepower and figured through the transmission with an assumption of 1½% loss at each gear mesh. In those tractors where the normal power was estimated as a percentage of the maximum drawbar horsepower, the tangential force on the rear wheels was calculated by using that percentage of the maximum drawbar horsepower at the rolling radius of the tire and adding an allowance for rolling resistance. It should be noted that the tangential force on the rear wheels is not exactly the same as the drawbar pull of the tractor; tangential force being made up of drawbar pull plus rolling resistance. In each gear the miles-per-hour ground speed was also calculated. This speed as shown in the tabulation is the theoretical ground speed and does not take into consideration any slippage. Under each speed is the manufacturer's estimate of the time that each particular tractor was used in that gear.

After obtaining the values for tangential force on the rear wheels based on each manufacturer's estimate of the average power requirements of his tractor, a ratio was struck between this value and the weight of the tractor. This was done for each speed on each tractor and the results are also shown on the chart.

■ Analysis of Data

In order to put the data shown on the above chart in usable form, they must be analyzed from two standpoints: one, the percentage of time the tractor is used in its various speeds; two, the relationship between the ratio of tangential force on the rear wheels to tractor weight as compared to the miles-per-hour speed of the tractor. With reference to the percentage of time spent in the various speeds, the estimates furnished by the various manufacturers have been averaged with the following results:

3-Speed Tractors

Gear	Mph	% of Time
Low	Up to 3	16
Second	Between 3 and 5¼	68
Third	Over 5¼	16

4-Speed Tractors

Gear	Mph	% of Time
Low	Up to 3	10
Second	Between 3 and 4	29
Third	Between 4 and 5¼	44
Fourth	Over 5¼	17

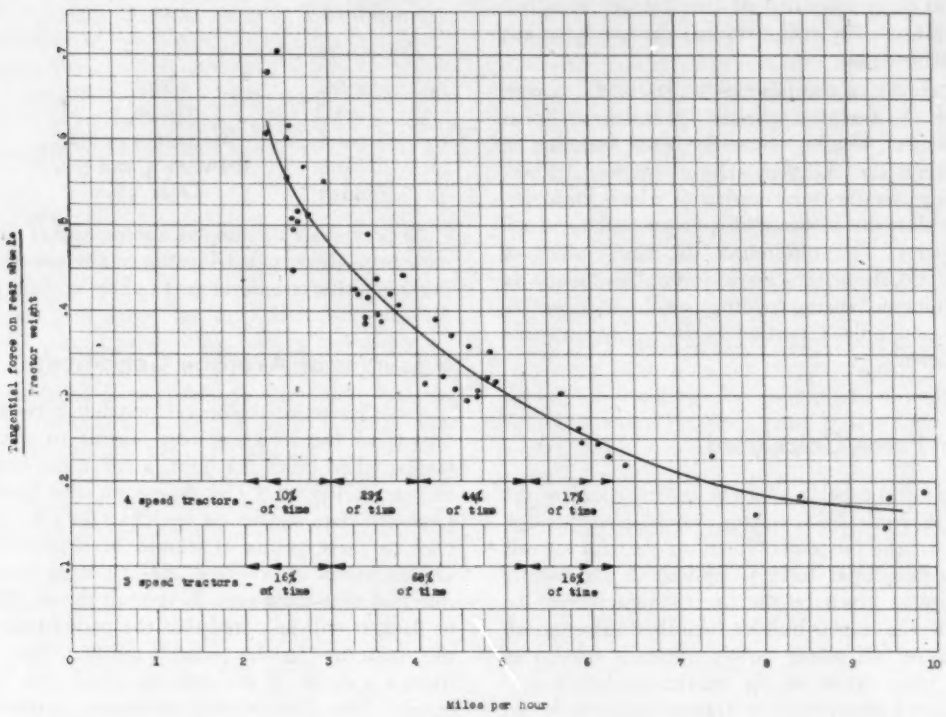
As previously mentioned, speeds higher than fourth gear have been disregarded because of the extremely light bearing loads that are developed in the higher speeds.

■ Curves of Average Conditions

With reference to the relationship between the ratio of tangential force on the rear wheels to tractor weight as compared to miles per hour, a curve has been prepared as shown in Fig. 5. The points on this curve are plotted from the data shown on the chart in Fig. 4. With reference to these points, it should be remembered that they are estimates of average conditions as furnished by six different manufacturers. In spite of the six different sources of data, it will be noted that the points follow a very definite pattern. In the writer's opinion, this, in itself, constitutes a check of the data on which this curve has been based. The actual curve, therefore, represents an average of the estimates furnished by these six different builders regarding average operating conditions on a farm tractor. As a matter of convenience, the breakdown of the time in the various speeds has also been added to this curve in chart form.

Working from the above curve, 18 different tractors have been analyzed, including the 14 on which this study was based, in order to determine what motor torque is required to produce the results shown on the curve. The tangential force on the rear wheels of each tractor was determined in each speed from the curve. Knowing this tangential force, the torque on the rear axle was computed and, the total transmission and rear-axle ratio being known, the amount of torque necessary at the flywheel of the engine to produce this result was calculated. An efficiency of 98.5% was assumed at each gear mesh. The required motor torque was computed in each speed and set up as a percentage of the total rated torque of the motor. The percentage was thus established in each speed and a weighted average arrived at taking into consideration the percentage of time in each speed previously tabulated. From these data a second curve, as shown in Fig. 6, was prepared. This shows the percentage of motor torque required for average conditions plotted against the power-weight ratio of the tractors in terms of horsepower per 100 lb of weight. Again it will be noted that the points follow a definite pattern and, as would be expected, the lower the power-weight ratio of the tractor, the higher the percentage of motor torque required under average conditions.

In order to determine the bearing loads and lives on a new design of tractor, a preliminary check could be made by using the curve shown in Fig. 6. To use this curve, it is necessary to make an estimate of the weight of the pro-



■ **Fig. 5 – Relationship between ratio of tangential force on the rear wheels to tractor weight as compared to miles per hour**

This curve has been plotted from the data tabulated in Fig. 4. The points follow a very definite pattern, despite the fact that they are estimates of average conditions as furnished by six different manufacturers. The breakdown of the time in the various speeds has been added as a matter of convenience

posed tractor in operating condition with water in the rear wheels and to know the rated torque of the motor that will be used. From this information, the power-weight ratio of the tractor can be determined and the percentage of motor torque to be used in making preliminary calculations can be established. Knowing this torque, bearing loads at the various positions can be computed, tentative bearing selections made, and lives checked. The curve gives the torque at the flywheel, and suitable deductions should be made for gear efficiency depending on the number of gear meshes between the flywheel and the bearing being checked. However, for a final check on the bearings, it is recommended that the loads be computed from the curve as shown in Fig. 5, since it is believed that this curve gives a more accurate picture of normal conditions in each speed. In working from the curve in Fig. 5, it will be found that the motor torque is not exactly the same in all speeds, which may be more nearly correct than assuming the same torque in all speed ranges.

To use this curve, it is necessary to know the weight of the proposed tractor in operating condition, including operator and water in the rear tires. It is also necessary to know the miles-per-hour speed in each gear and the tire size that will be used. This tangential force on the rear wheels in each speed can be obtained from the curve in terms of percentage of total tractor weight. This tangential force on the rear wheels multiplied by the rolling radius of the tire will give the torque on the rear axle. From this torque, the bearing loads can be calculated. As an actual example, let us assume a tractor that has a total weight in operating condition with an operator and with water in the tires of 3000 lb. Let us further assume that it will have 9-32 tires having a 23-in. rolling radius and that the low-gear speed will be 2¾ mph, the second-gear speed 3½ mph, the third-gear speed 4¾ mph, and the fourth-gear speed 6 mph. From the curve it can be established that the normal tangential force on the rear wheels in low gear will be 51% of the weight, in second gear 42½% of the weight, in third gear 32% of the weight, and in fourth gear 25% of the weight. Translated to

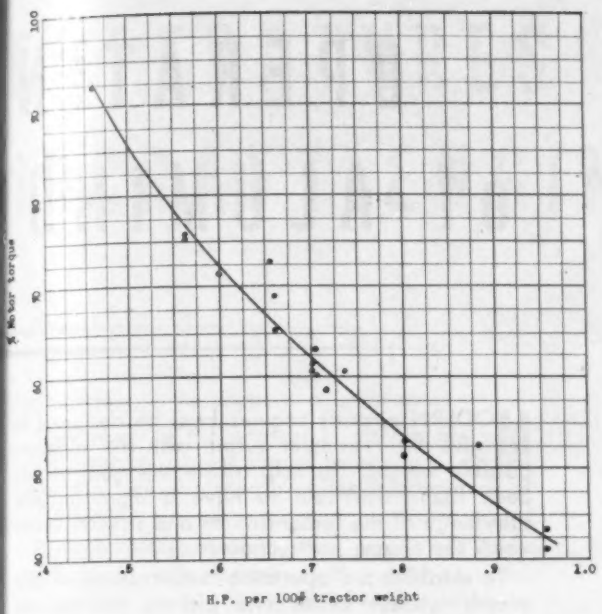


Fig. 6—Curve showing the percentage of motor torque required for average conditions plotted against the power-weight ratio of the tractors

It will be noted that the points follow a definite pattern, the lower the power-weight ratio of the tractor, the higher the percentage of motor torque required under average conditions

pounds; the tangential force on the rear wheels in low gear will be 1530 lb, in second gear 1275 lb, in third gear 960 lb, and in fourth gear 750 lb. With a 23-in. rolling radius on the tire, the rear-axle torque in low will be 35,200 in.-lb, in second gear 29,300 in.-lb, in third gear 22,000 in.-lb, and in fourth gear 17,200 in.-lb. From these torque values, the bearing loads in each speed can be computed and the life factors determined. The weighted life can then be established on each bearing by means of the formula previously discussed.

Checking Transmission Bearings

In actual practice, the problem of checking the bearings in a tractor transmission is comparatively simple. The usual method is first to check all bearings in low gear. In most cases, this will give the maximum bearing loads at all positions. It will then be found that many of the bearings are quite lightly loaded, the bearing selections being dictated by the required shaft size. There is obviously no need of checking these bearings in the other gears where the loads will be lighter. On bearings where load and not shaft size is the determining factor in the selections, a check of the loading should be made in the other speeds. After the life factor has been determined in each speed, the weighted life of the bearing can be determined. In such positions, it is usually desirable to check several different sizes of bearings for each position and determine the life on each of them. Knowing the life and the cost of each of these selections, the designing engineer can make a final selection, weighing one factor against the other.

With reference to the life that should be provided in farm tractor bearings, this is an item which must be decided by the designer on each individual unit. When the loads are known, the engineer can select bearings for almost any desired life. The size of the tractor and the service for which it is intended enter into the life that will be required. For example, on a large heavy tractor where the cost of the bearings is not a large item in the total cost, a little extra put into the bearings which may give a considerably longer bearing life is probably money well spent. However, on the small highly competitive tractors where cost is an important factor, a different decision might be made.

Limitations of Study

In connection with this study, it should again be emphasized that it is based solely on pneumatic-tired tractors. With reference to steel-wheeled tractors, no attempt has been made to accumulate similar data. However, it seems to be generally agreed that to produce comparable drawbar pull, a steel-wheeled tractor requires from 20% to 30% more power than a pneumatic-tired tractor. As to whether the curves worked out for the pneumatic-tired tractors could be used for steel-wheeled units by adding a similar percentage to the values, the writer is unprepared to state.

Furthermore, the study has been based largely on the smaller general-purpose tractors where economy in bearing selection is of paramount importance. Indications are, however, that the tangential force on the rear wheels-weight ratio as shown in Fig. 5 holds true also for the larger models but that on these larger units there may be a slightly different distribution of time between the various speeds.

It should also be emphasized that the data derived applies only to the bearings which are loaded in drawbar work. Bearings loaded in power take-off work or in belt pulley work must be checked on some other basis. No analysis has been made as to average power requirements on the belt pulley, but from general observation it appears that this would be a higher percentage of motor torque than that normally used in drawbar work.

As mentioned at the beginning of the paper, the data derived on average operating conditions of pneumatic-tired tractors may be applicable in the design of other parts subject to fatigue than the bearings.

Method Based on Average Opinions

The writer wishes to point out that this entire study has been based on information furnished by the various tractor manufacturers. To reiterate; the data obtained from these different sources seem to check themselves and is, therefore, presumably very close to being correct.

The results obtained represent the average opinions of the industry as of today. It is entirely possible that changes may be made in the fundamental design of farm tractors in the future and that the data derived in this paper will no longer be valid.

The conclusions reached in this paper were made possible by the cooperation of the engineers of the various tractor companies in furnishing extensive data and the writer acknowledges their invaluable assistance in its preparation.

OPERATIN TE of ALUMINU AI

MODERN aircraft engines must be designed to give the maximum output with the minimum possible weight. To help attain this goal, it has been found desirable to have a more definite knowledge of the temperatures and stresses under which the engine parts operate.

To establish the operating temperatures of aluminum cylinder heads and pistons, the authors have developed a method that takes advantage of the fact that the Brinell hardness of the cylinder head and piston decreases with the number of hours of service and the temperature at which the part operates.

The stresses under which parts operate have also been studied by the authors, who used the brittle lacquer method, extensometers, and wire strain gages. Of these three, they found the brittle lacquer method the most satisfactory. Its ability to operate in constricted areas around the ribs

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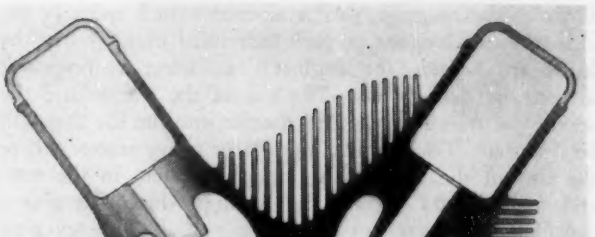
THE AUTHORS: E. J. WILLIS (J '41) has been connected with the engineering department of Aluminum Co. of America since his graduation from the University of Michigan in 1936. At present he is contacting various aircraft engine



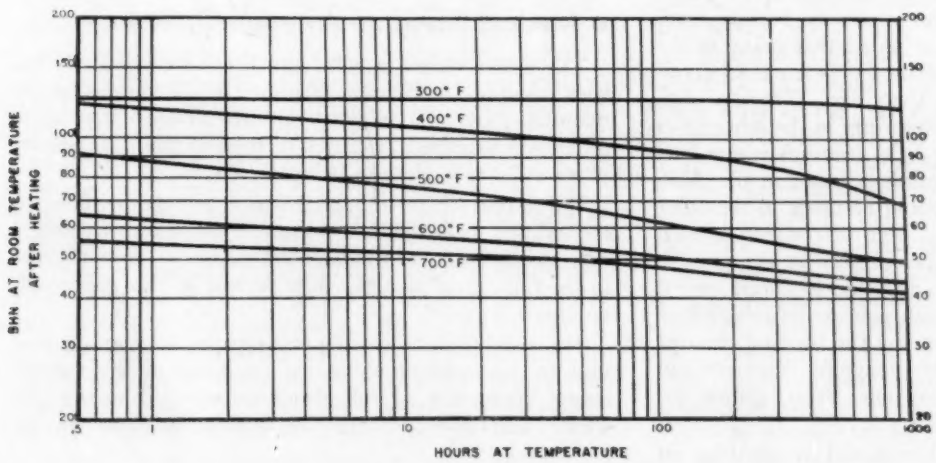
■ Fig. 1 – Cylinder head and piston shaded to indicate temperatures of operation (The hottest areas are shown in black graduating to the coolest areas in white)

AIRCRAFT engines of today require the maximum output with the minimum possible weight. Using the present models as a basis of experimental work, improvements can be made on them by knowing the temperatures and stresses under which the parts operate.

All aircraft-engine cylinder heads are being made of aluminum. The items which determine the value of an aircraft cylinder head are its ability to cool and its ability to carry the stresses imposed upon it.

Fig. 1 shows a cylinder head, barrel, and piston with the temperatures of operation indicated by shading; the darkest areas being hottest, proceeding toward the cooler end in white.

[This paper was presented at the SAE National Aeronautic Meeting, New York City, April 9, 1943.]



■ Fig. 2 – Relation of Brinell hardness to temperature and time for 32S-T aluminum alloy forged slab

TEMPERATURES and STRESSES IN AIRCRAFT-ENGINE PARTS . . .

by E. J. WILLIS and R. G. ANDERSON
Aluminum Co. of America

and interiors of the piston is a great asset.

Since most of the stresses in pistons are of a fatigue nature, pistons should be designed on the material's fatigue strength at the temperature of operation.

To study fatigue stresses, the authors use a hydraulically operated fatigue testing machine, which can produce up to 100,000 lb of load in either or both directions between the master piston and the bolster plate at 1300 cycles per min. The part to be tested is placed in the machine with its head against a cushion with an appropriate connecting rod.

Electric heaters, properly controlled, simulate operating temperatures of the piston. It is quite necessary to maintain the same temperature gradients throughout the piston to produce failures comparable to those occurring in service.

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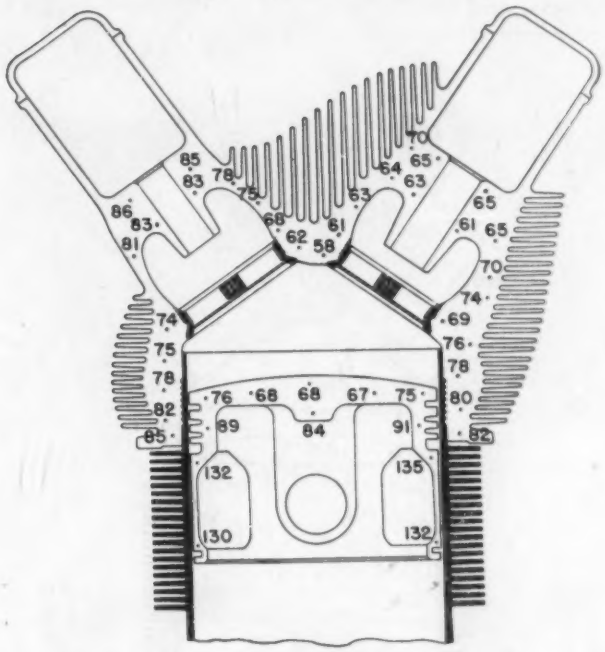
manufacturing plants as well as working with aircraft development in the Cleveland laboratory of ALCOA. R. G. ANDERSON is in charge of the stress analysis laboratory of Aluminum Co. of America. He has also done engine design work and engine testing for the same company.



■ Fig. 3 – Section of a cylinder head that has been Brinelled to show the variations in hardness

At present, the proportions of the cylinder-head fins have not been fixed. Usually, they are dependent upon the ability of the foundry to cast maximum fin lengths with a minimum spacing between fins. It has been a policy to put as much fin area on the cylinder head as it is possible to cast. However, this does not make the best possible use of the material. Due to the advent of the forged head, more leniency in both thickness and spacing of fins is available. Some areas around the exhaust ports of the cylinder head operate at much higher temperatures than others. Therefore, the maximum fin areas should be at this location.

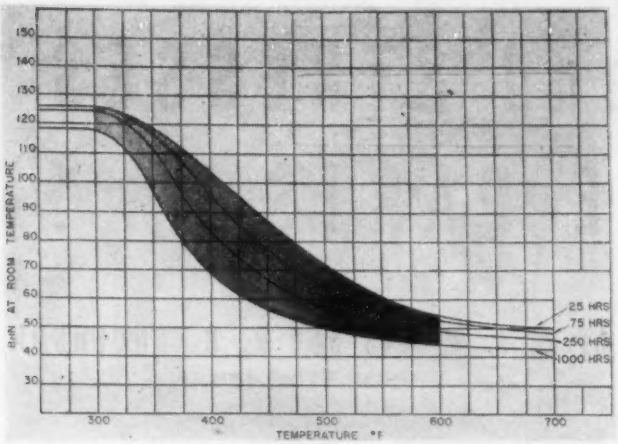
To establish operating temperatures of cylinder heads, a method has been worked out which gives the running temperatures of the head. The Brinell hardness of the cylinder head decreases with the number of hours in service and the temperature at which it operates. If the laws governing the softening of the material are known, it is possible to determine the temperature of any particular spot in the cylinder head by Brinelling this location after running. The length of time of operation must be known and a particular alloy and heat-treatment must be used. Fig. 2 shows the relation of Brinell hardness to both temperature and time for a typical aluminum alloy.

Fig. 3 shows a section of a cylinder head which has been Brinelled to show the variation in hardness. Since the entire head operated a given length of time, all variations in the hardness are indicative of the temperature which the various locations have reached.

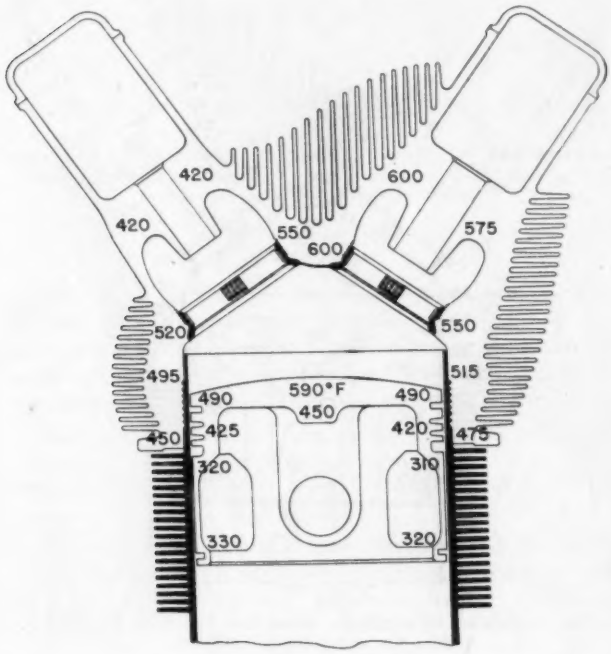
Fig. 4 shows a spectrum by which one may readily see the hottest portions of the cylinder head. The black end of the spectrum is hottest, while the white end is coolest.

Fig. 5 shows a cylinder assembly with the hardness indications converted to temperature.

The stresses under which a cylinder head operates are



■ Fig. 4 – Brinell hardness of 32S-T alloy at room temperature after heating at elevated temperature

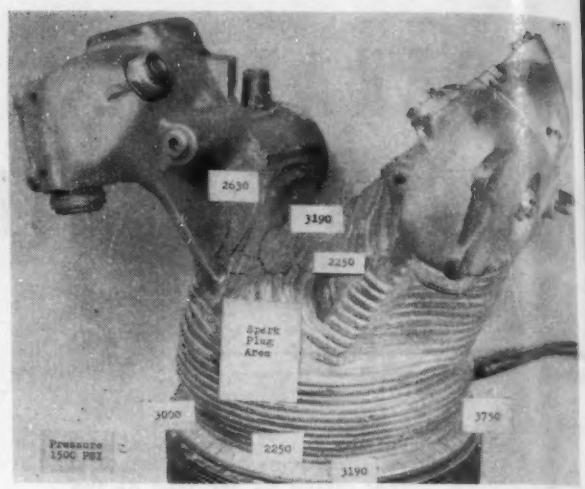


■ Fig. 5 – Cylinder assembly with the hardness indications converted to temperature

of two natures, (1) prestress and (2) fatigue stress. When the cylinder head is assembled with its required inserts, each one of these inserts imposes a tensile prestress in the aluminum. For example, the barrel is a shrink fit in the cylinder head. This is true of spark-plug inserts, valve-seat inserts, valve-guide-boss inserts, and the various studs. On top of the prestresses are the cyclic stresses imposed by the explosion loads. In addition to this, there are high stresses produced by the rocker-arm push-rod sections.

Fig. 6 shows the fatigue stresses produced by 1500 psi in the combustion chamber.

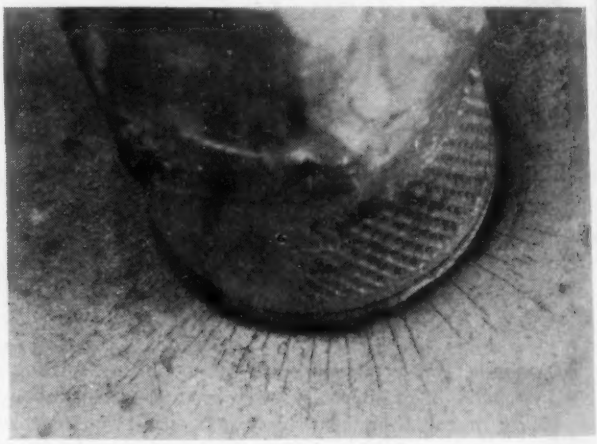
If there were no stress concentrations because of mating parts and sharp radii, it would be relatively simple to calculate the stresses at a particular spot in the cylinder head. However, due to the complicated structure, this is impractical.



■ Fig. 6 – Fatigue stresses produced by 1500 psi in the combustion chamber

Fig. 7 shows the stress concentrations around the spark plug.

The cylinder-barrel threads produce a large number of stress concentrations both in the radii of the threads and at the top of the barrel. The concentrations can be minimized by proper fits and designs but they are always present. Cylinder-barrel threads should match in contour and pitch at the operating temperature of the joint rather than at shrink temperature or room temperature.

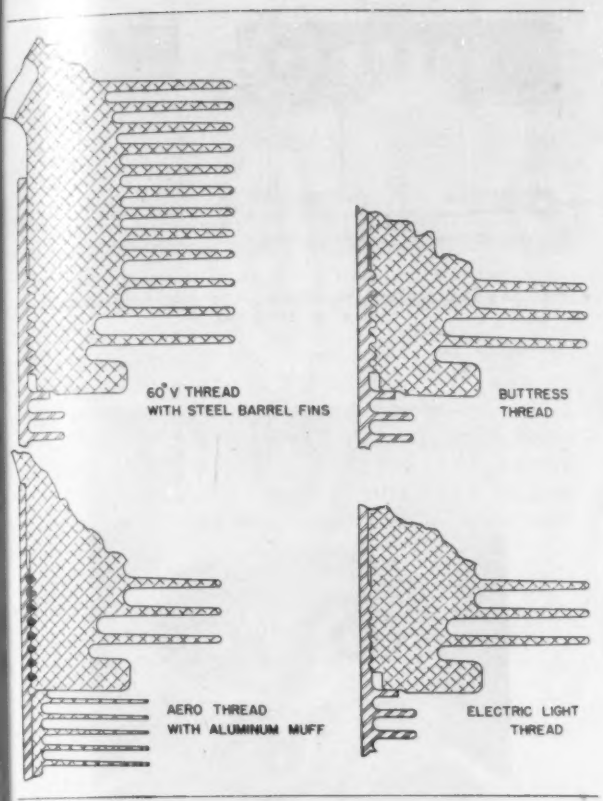


■ Fig. 7 – Stress concentrations around a spark plug

Fig. 8 shows sketches of design proportions and type of cylinder-barrel threads.

The proportion of aluminum-head thickness to steel-barrel thickness in the vicinity of the joint and pilot area should be 4.2 to 1 as a minimum design ratio. This takes into account both the difference in modulus between the two parts and the temperatures at which the assembly operates.

The connection between the cylinder head and the exhaust pipe has caused trouble from time to time. Shrink fits have been tried and worked satisfactorily for the lower output class of engines. When higher horsepowers are



■ Fig. 8 – Sketches of cylinder-head threads

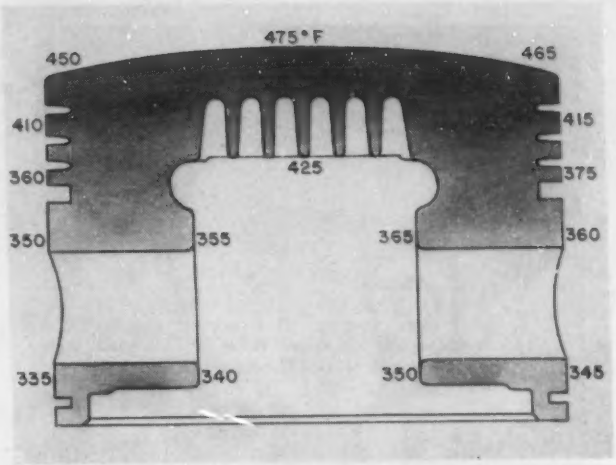
required, it is best to use a flange and studs in the aluminum, keeping the exhaust pipe as flexible as possible. It is well known that the cylinder-head assemblies flex considerably in relation to each other during actual operation. For this reason, a flexible joint is a necessity.

Liquid-cooled cylinder heads are usually constructed with an inner dome which carries the explosion loads and the various inserts, while an exterior jacket retains the cooling liquid. It has been found that this cylinder-head jacket must be extremely ductile to withstand the strains imposed upon it. In some jackets, the first explosion which the head undergoes strains the material past its yield point at points of stress concentrations and causes distortion of the cylinder head under load.

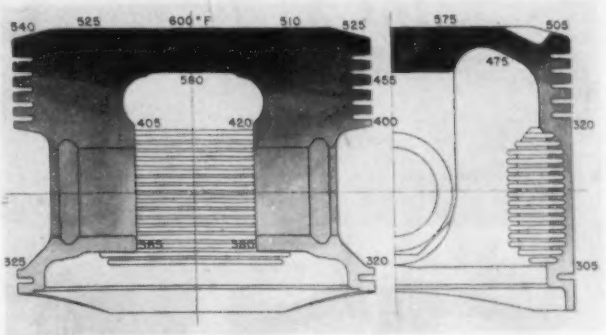
■ Pistons

Progressing inward from cylinder heads to the next major aluminum part, we find the piston operating under even more severe conditions than the aluminum cylinder head. There are three items which greatly affect the operation of the piston. These are (1) temperatures, (2) stresses at various points in the piston, and (3) bending or distortion of the piston and its mating parts. We obtain the temperatures involved in this particular part by the same process as previously mentioned for the cylinder head, namely, the Brinell hardness drop method. Many pistons have been sectioned and Brinell hardness reading obtained.

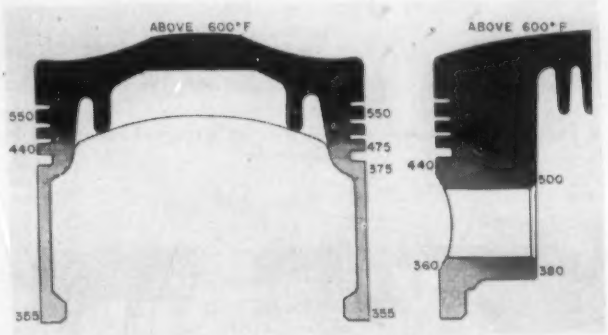
Figs. 9 through 16 are shown to indicate the piston temperature variations between engines and the effects of operating conditions. The hottest areas are shown in black graduating to the coolest areas in white.



■ Fig. 9 – Temperature variations in an aircooled piston – 250 hr at 200 bmep

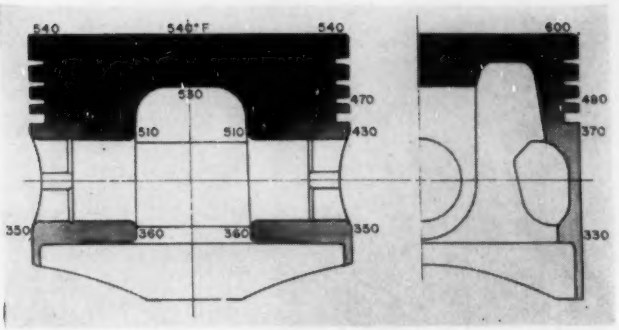


■ Fig. 10 – Temperature variations in an aircooled piston – 1350 hr of airline service

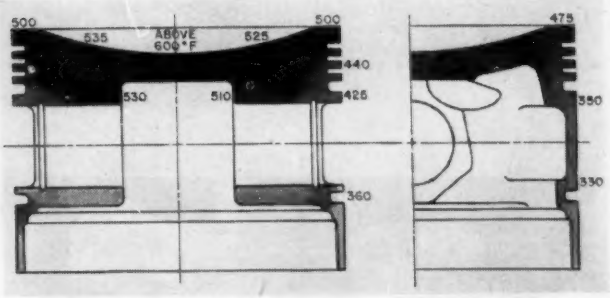


■ Fig. 11 – Temperature variations in a liquid-cooled piston – 280 hr at 250 bmep

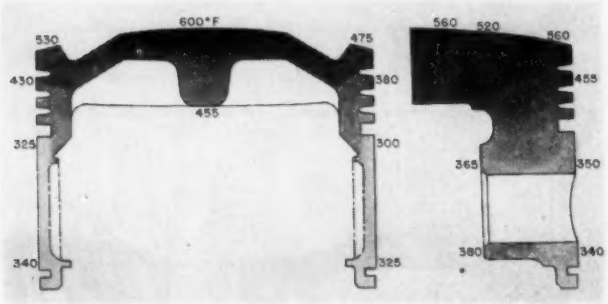
With the temperatures at which these pistons operate fairly well determined, it is logical next to survey the stresses under which these pistons operate. We have found that the strains can be determined quite readily with extensometers, brittle lacquers, and wire strain gages. Of these three, the most satisfactory method seems to be the brittle lacquer method. Its ability to operate in constricted areas around ribs and interiors of the piston is a great asset. This can be only partially covered by mechanical extensometers.



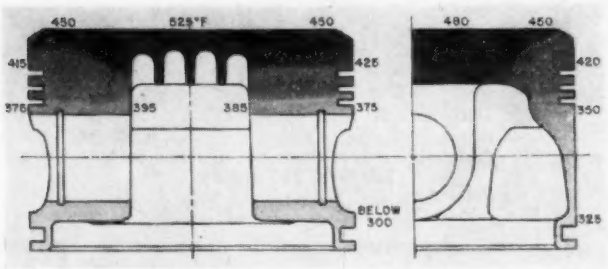
■ Fig. 12 – Temperature variations in a liquid-cooled piston – 175 hr at 250 bmep



■ Fig. 13 – Temperature variations in a liquid-cooled piston – 330 hr at 180-210 bmep



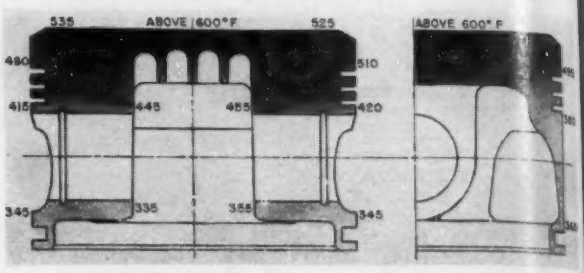
■ Fig. 14 – Temperature variations in an aircooled piston – 25 hr at 225 bmep



■ Fig. 15 – Temperature variations in a liquid-cooled piston – 190 hr at 178 bmep

Fig. 17 shows the concentration of stress around the wristpin hole on the inside of the boss due to a hydraulic load on the head of the piston of 1500 psi.

Fig. 18 indicates both tension and compression in another piston.



■ Fig. 16 – Temperature variations in a liquid-cooled piston – 98 hr at 216 bmep

■ Fig. 17 – Concentration of stress around the wristpin hole on the inside of the boss due to a hydraulic load of 1500 psi on the head of the piston

■ Fig. 18 – Both tension and compression are indicated in this piston

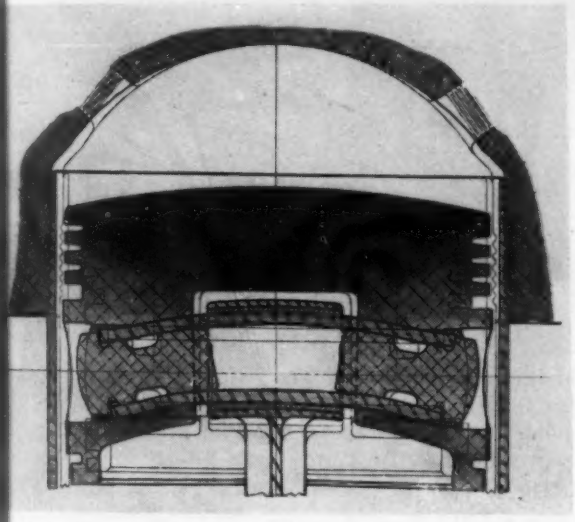
The stresses in the head rib are tensile in nature as well as those progressing radially from the wristpin hole. The rest of the stresses shown are compression. This work has shown that the strength of the wristpin and connecting rod have considerable effect on the stresses in the piston.

For example, by placing a solid wristpin in a given piston, the tensile stresses around the wristpin hole can be reduced as much as 40%. Of course, this is not practical from a weight standpoint, but an approach to this ideal condition can be realized by inserting a plug in the pin to act as a stiffener against both flattening and shear. Small steel rools placed at the maximum shear position give the greatest reduction for a minimum increase in pin weight.

The limiting loads which the piston material can withstand are determined by the stress concentrations at the inside of the wristpin boss. Under load, the piston dome bends inward causing a concentration of stress at the inner edge of the pin bore. The bending of the wristpin upward has the same effect. Pins of larger diameter which have a shorter length sometimes work considerably better even though they have a higher load per unit of projected area. This reduces the amount of bending in the wristpin, thus reducing concentration on the edge of the wristpin boss.

Fig. 19 indicates the distortion of the wristpin in a piston amplified 25 times for clarification.

Piston distortion due to explosion load may show up at other locations as well as the wristpin bore. For example, the ring lands distort.

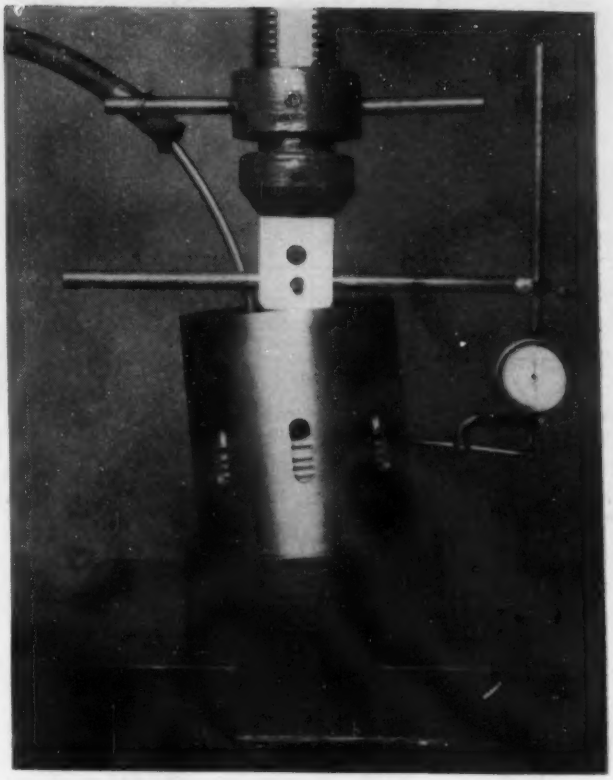


■ Fig. 19 – Distortion of the wristpin in a piston, greatly amplified

This ring-land distortion can be readily measured by the test rig shown in Fig. 20. It should be noted that the piston is tilted for loading at a 5-deg angle. This is approximately its position at the maximum explosion pressure in the cycle. At room temperature, this ring-land distortion may vary from 0.000 to as much as 0.024 in. around the circumference. A variation between ring lands always exists.

Fig. 21 shows the distortion in a satisfactory design of piston under 1500 psi head pressure taken at room temperature.

Fig. 22 shows the distortion taking place in another design of piston with the solid lines indicating room-temperature conditions and the dotted lines showing the ring-land distortion at operating temperatures. We consider designs which show this amount of distortion unsatisfactory. Tests indicate if the distortion of the lands is not excessive at room temperature, it will be somewhat improved at elevated temperature, probably due to redistribution of stress.



■ Fig. 20 – Test rig used for measuring ring-land distortion

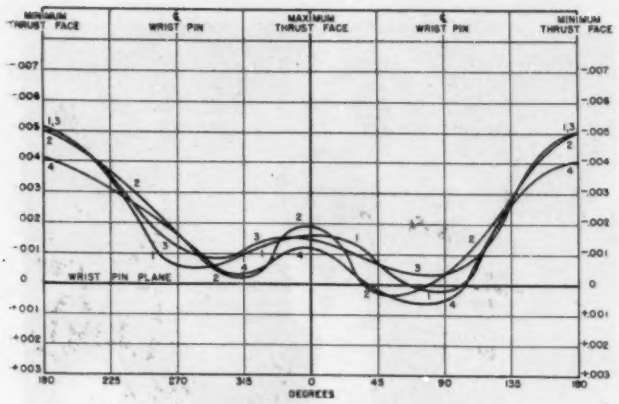
Most of the stresses in pistons are of a fatigue nature and pistons should be designed on the material's fatigue strength at the temperature of operation. Tensile strength and elongation do not seem to enter into the picture except during scuffing or an engine failure where both tensile strength and elongation may be considered as safety factors.

Fig. 23 shows a comparison of the fatigue strengths of 32S-T alloy and 18S-T alloy at elevated temperatures.

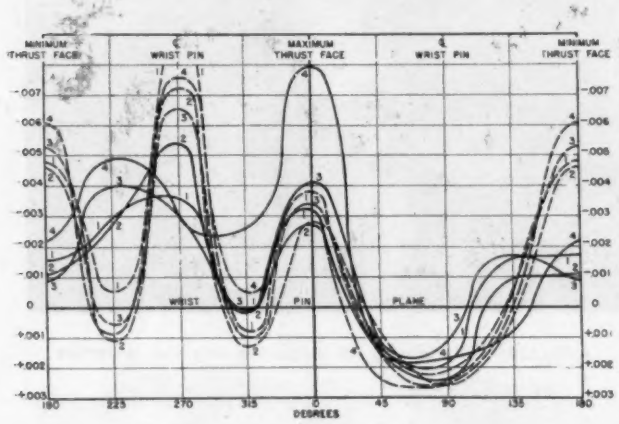
The fact that some portions of the piston operate at different temperatures should be carefully considered in determining the stress which that particular part of the piston can withstand. For example, if a piston operated with the bar across the center of the dome at 500 F, it can withstand approximately 5000 psi in reversals of stress at that temperature for indefinite life. If the temperature could be reduced to 400 F, nearly double that load could be carried.

■ Use of Fatigue Testing Machine

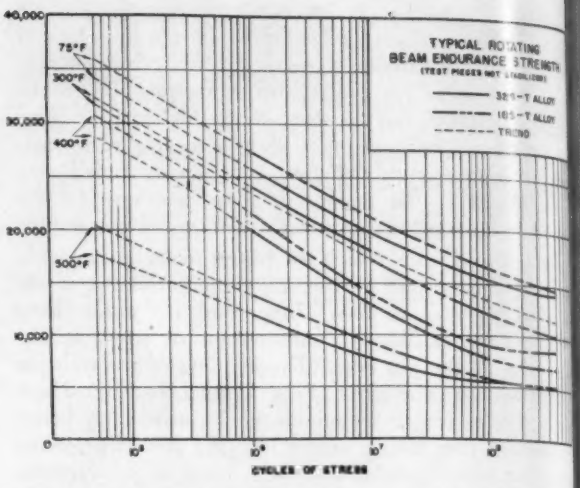
In order to tie the static type of testing to operating conditions, a fatigue testing machine has been procured and used in our laboratory. Fig. 24 shows this machine. It is essentially a hydraulically operated machine, which can produce up to 100,000 lb load in either or both directions between the master piston and the bolster plate at a speed of 1300 cycles per min. The part to be tested is



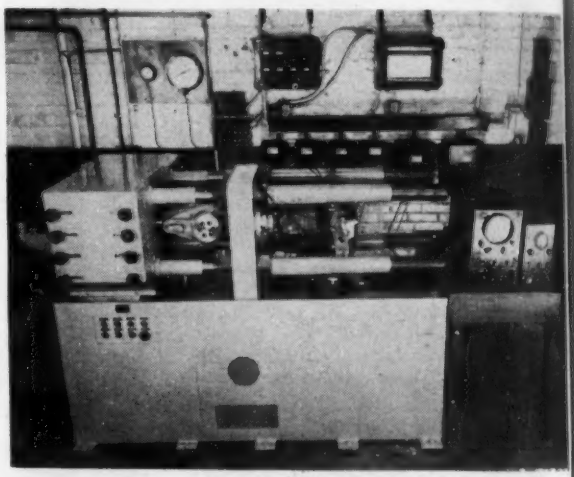
■ Fig. 21 – Ring-land deflections due to 1500-psi cylinder pressure taken at room temperature (connecting-rod angle: 5 deg past TDC)



■ Fig. 22 – Ring-land deflections due to 1500-psi cylinder pressure (connecting rod angle: 5 deg past TCD)
——Room-temperature conditions
——Elevated-temperature conditions
Average temperature inside head: 407 F
Average temperature at bottom of skirt: 303 F



■ Fig. 23 – Comparison of the fatigue strengths of 32S-T and 18S-T alloys at elevated temperatures



■ Fig. 24 – Fatigue testing machine

placed in the machine with its head against a cushion with an appropriate connecting rod.

The operating temperatures of the piston are produced by placing electric heaters around the cylinder barrel, thus heating the test parts. Control equipment turns these heaters on and off, thus holding the test parts at predetermined temperature levels. It is quite necessary to maintain the same temperature gradients throughout the piston to produce failures comparable to service.

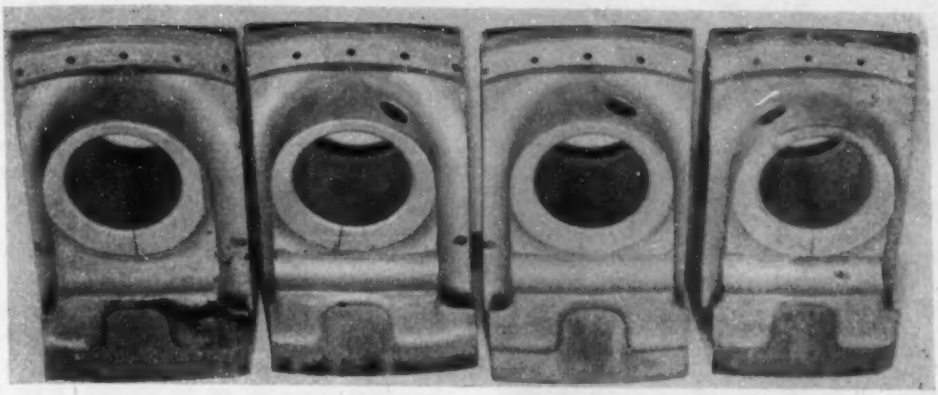
Fig. 25 indicates the results which we have obtained at 1500 psi with 500 F dome temperature. A design recently developed with this machine is shown in Fig. 26. The equivalent length of time in an engine for the same test would be between 50 and 100 hr of take-off condition as compared with 8 hr in the machine. We feel that temperature, stresses, and distortion will dictate the design of the future pistons for high-output aircraft engines.

■ Crankcases

Continuing from the piston to the next major aluminum part, the crankcase comes in for discussion. Aluminum crankcases have been used since before the first World War in cast form. As outputs increased in radial engines, cast material was replaced by forged material due to its higher fatigue strength. The radial engine manufacture has been able to make use of forgings. However, th in-line type of engine does not lend itself to this type o construction.

The method of attachment of the cylinder barrel to th crankcase in most cases is by means of studs in the alum num.[5-6] The fit of these studs is relatively critical becaus of the high loads imposed upon them from explosio forces. Studs will operate satisfactorily in the aluminum if proper care is exercised in their design and assembly Proper fits and assembly methods have been worked ou In the design of crankcases, care should be exercised to b sure that any bending stresses caused by high loads ar not through the sections in which the studs are located The load each stud carries is somewhat dependent upo the amount of flexing in the crankcase. By obtaining th strain in each stud, the distribution around the hold-down circle can be studied. All studs are not uniformly loaded It may vary from no load at the parting line of the tw cases to a maximum for the studs directly over the mai bearing partition wall.

Fig. 27 shows the distribution around this circle as a solid line. The tightening of the hold-down nuts on these stud is especially critical. To indicate what may happen if tw



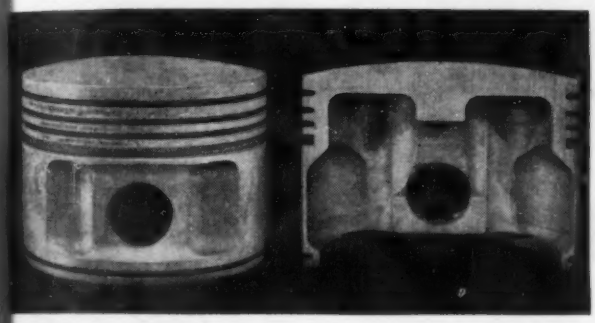
■ Fig. 25 (above) – Results of fatigue tests on one type of piston under 1500-psi load and 500F dome temperature

A
2,342,000 cycles (Standard wristpin)

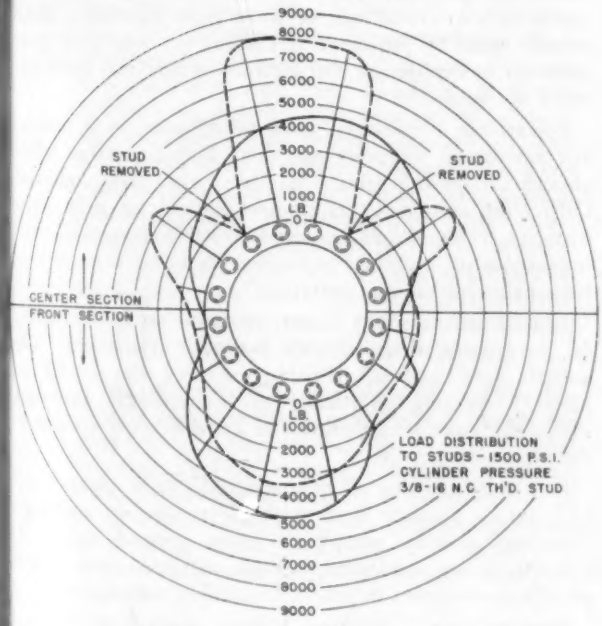
B
1,072,650 cycles (Standard wristpin)

C
314,280 cycles (Standard wristpin)

D
1,000,020 cycles (Standard wristpin) (Magnesium pin plugs)



■ Fig. 26 – Piston design developed with the aid of the fatigue testing machine



■ Fig. 27 – Load distribution to studs attaching the cylinder barrel to the crankcase

nuts are loose, the dashed line has been superimposed upon the normal distribution curve in this figure.

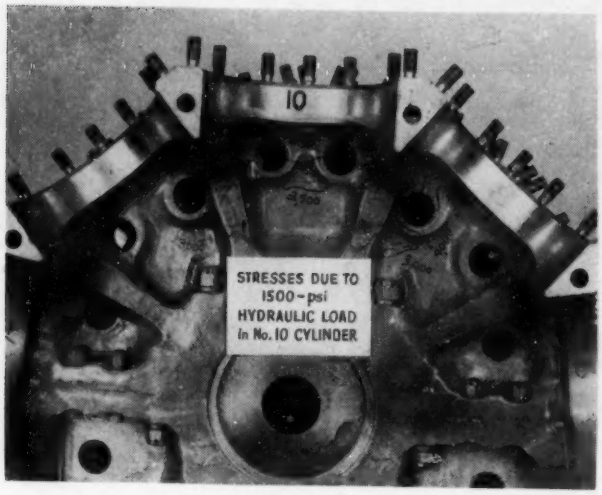
The loads which the two studs over the partition carry are nearly doubled, indicating an extremely dangerous condition. Several methods for improving this distribution can be used: first, by increasing the stiffness of the legs between the cylinder barrels, which forces the studs nearest the parting line to absorb more load; second, by increasing the flexibility of the partition under the two studs which are most highly stressed.

■ Crankcase Ribs

Caution in the use of ribs in crankcases cannot be too highly emphasized. When ribs are used in conjunction with a partition, the ribs take most of the load on their extreme fibers and when thin, round-topped ribs are used, fatigue normally starts at the top of the rib. It would be much better use of the metal if only ribs are used in the entire construction because these ribs could be made broader and stronger. The partition itself, when ribs are used, carries very little of the load and could be made relatively thin. On the other hand, if it is necessary to use a partition type of construction, shallow, wide ribs or bars can be used as strengtheners very satisfactorily.

Experimentally, we have run stress distribution tests on crankcases incorporating a ring nut instead of hold-down studs. This shows much better distribution. The ring was similar to that of the German DB-601 engine having threads below the hold-down flange on the barrel and a nut inside the crankcase.

Fig. 28 shows the stresses in a radial crankcase produced by 1500-psi explosion pressure. The method used is brittle lacquer in a constant-temperature room. Further studies can then be made by means of strain gages.



■ Fig. 28 – Stresses produced in a radial crankcase by 1500-psi explosion pressure

■ Summary

In all the aluminum parts mentioned, the temperature at which they operate is extremely important. Future development will be along the lines of more accurate determination of these temperatures. Since particular localities in a given part may run at much higher temperatures than others, the stresses for that particular locality must be known to make an intelligent improvement of the design. The temperatures of both pistons and cylinder heads are approaching a maximum, and effort should be expended to lower these temperatures.

So far, we have only scratched the surface of investigation on temperatures and stress measurements. We feel great improvement can be made by further study along these lines. Its effect on the aircraft engine will be toward lighter parts. The importance of this method of analysis can be readily understood by all familiar with the necessity of increasing the horsepower-to-weight ratio of a modern aircraft engine.

LESSONS from MILITARY OPERATION of AIRCRAFT

EXCLUSION of dust from airplanes, especially engines, is practically impossible to accomplish. To exclude dust completely, one must exclude air, under desert conditions. As this is an impossibility, the urgency of *adequate* air cleaner equipment at *all* engine openings is obvious.

Protection of engines from dust during operation is only one phase of the service problem. Equally, if not more important is the protection of all engine openings while aircraft are on the ground.

Fighter aircraft generally have had a much greater exposure to dust than bombers, for the latter types are usually operated off of fields with some form of paved or block-top runways, while the fighter is generally based on fields without runways. Another dirt exposure to which the fighter is submitted is the alert warm-up, which frequently is done for 5 min out of every half hour of daylight. The bomber may have only one mission daily or at night, so it is exposed to this warm-up practice only once or twice daily.

One of the outstanding conclusions reached as a result of overseas field service observation, is that everything possible must be done in the design and engineering of aircraft to ensure ease of service and maximum durability under the most primitive field conditions.

It appears particularly necessary that aircraft be designed so that they can be readily broken down into units, each complete in itself, and of such size as to be handled readily on standard transport vehicles and so that a minimum of time is required to remove any given unit and replace it with a new or overhauled assembly. Even though such construction is somewhat heavier, it should be pointed out that the increased percentage of time in the air makes up many times over for the slight increase in weight.

Frequent trouble due to pipes rubbing together and causing leaks is reported. Ground crews seldom will check for this installation difficulty. Careful inspection rarely fails to reveal one or more pipes rubbing. Much trouble could be prevented if designers would give more attention to clip design and location so that this positively could not occur.

Powerplant installations must be designed to withstand any amount of humidity and dirt. Dirt protection while aircraft are on the ground has been especially neglected. Carburetor air cleaner equipment should be fully automatic in action. Pilots are not sufficiently conscious of its importance or interested in engine life to be depended on for satisfactory manual operation.

Despite the humidity factor, electrical equipment gives far less trouble in service than hydraulic equipment. The authors have never talked to a crew chief who would not prefer everything electrically to hydraulically operated. Hydraulic leaks are common and are difficult to correct in the field.

It is suggested that all new service types be given a service test on a rough field, with plenty of dust and high humidity, and by completely green ground crews. If installation designers were to work to such a service test, rapid improvements in field serviceability would result.

Excerpts from: "Servicing Military Aircraft in Overseas Theatres," by C. R. Paton and W. C. Gould, Allison Division, General Motors Corp., presented at the SAE National Aircraft Engineering and Production Meeting, Los Angeles, Calif., Sept. 30, 1943.

SPECULATIONS on AIRWORTHINESS and OPERATING REGULATIONS

by EDWARD WARNER
Vice-Chairman, Civil Aeronautics Board

ALL regulation of civil aviation has a common purpose. Its primary object is the protection of the innocent. The public on the ground must be protected against hazards from overhead. The patrons of commercial services must be protected against incompetence in operators, or their use of unfit equipment. The users of the air space must be protected against each other. The purchasers of aircraft must be protected against the sale of equipment presenting inherent hazards beyond the power of any pilot to overcome, and protected against the offering of equipment for sale on the basis of incomplete or inaccurate information regarding characteristics that directly affect safety.

There is, I believe, very little dissent in principle on the desirability of providing such protection; but there is room for wide differences of opinion regarding the length to which regulation should be carried and the amount of detail that the regulations should contain.

Let me concentrate on the aspects of regulation that most directly affect the technical problems of design and operation. The subject needs lively and continuous discussion, in terms not only of the advisability of particular

[This paper was presented at a meeting of the Metropolitan Section of the SAE, New York City, Sept. 9, 1943.]

THE wartime interruption in the building of civil aircraft is a good period for making a skeptical re-examination of regulations as they now exist, to decide what shape future regulations should take. Now is the time for proposing changes, especially fundamental ones, if they are found desirable. The past 20 years have seen a general trend away from arbitrary methods of airworthiness regulations to more rational ones. This trend will continue, Mr. Warner believes, dealing in the next few years with the establishment of rational relationships between design requirements and operating conditions. It will concern the length to which we should go in replacing rigid requirements by flexible ones, leaving an opportunity of fitting the characteristics of the equipment to the particular purpose for which it is to be used.

Mr. Warner considers the question of whether design and operating requirements should be so interwoven that no aircraft can ever be used in an unsafe fashion without violating a regulation, or whether the user can be relied on to keep within the limits of safe practice and concentrate on getting for him the information that will enable him to decide for himself what safe practice is.

The proper scope of future equipment regulation, Mr. Warner feels, is that it should be framed:

1. To prevent the manufacture and sale of aircraft presenting an inherent and inevitable hazard.

2. To ensure that the purchasers of aircraft will receive adequate and accurate information on characteristics important for safety.

3. To require, in cases especially pertinent to the public interest, that the conditions under which equipment is used should be so related to the equipment's characteristics as to maintain proper margins of safety.

• • •

THE AUTHOR: EDWARD WARNER (LM '17), vice-chairman of the Civil Aeronautics Board and past-president of the SAE, has been active in the aeronautics field since 1911, when he designed and built a glider which competed successfully in the first intercollegiate glider meet. Author of many papers on the technical, economic, legal and political aspects of aeronautics, he has held various public and semi-public positions, among which were chief physicist of the National Advisory Committee for Aeronautics, assistant secretary of the Navy for aeronautics, vice-chairman of the Federal Aviation Commission, and economic and technical adviser to the Civil Aeronautics Authority. He received degrees from Harvard University and M.I.T., and in 1938 was made an honorary doctor of science at Norwich University.

regulations but also of the principles by which future development should be guided. It is easy to secure clear and vigorous expression on particular proposals from those who are to be directly affected. It is much less easy to induce busy designers or operators to find the time for writing papers and contributing discussions on general principles – yet it is important that they should. If consideration in terms of general principle is to be limited to Washington, no one realizes better than the Washingtonians how gravely handicapped the process of regulation will be.

We may all be too much occupied with more urgent matters to formulate new regulations in detail; but it will be helpful, in such snatches of time as we can spare, to give some thought to the shape that future regulation is to take. For the first time in many years, there is a complete interruption of the building of aircraft for civil purposes. Post-war models are likely to be sharply distinguished from their pre-war predecessors. So far as the time for it can be found, this is the ideal period for a skeptical re-examination of the regulations as they now exist, and for challenging the right of established patterns to survive. If there is any idea that the regulations could be improved by change, however fundamental, this is a good season for proposing it.

■ Trend Toward Rational Methods

A general trend in airworthiness regulation in the past 20 years has been the substitution of rational methods for arbitrary ones. A comparatively early instance was the modification of strength requirements to replace certain arbitrarily specified load factors by analysis for the structural consequences of a particular occurrence, such as the encountering of a vertical gust of stated velocity. There have been other such cases; and I believe that the most important single question that we shall have to deal with in the next few years will relate to a particular development in the establishment of rational relationships between design requirements and operating conditions. It will concern the length to which we should go in replacing rigid requirements by flexible ones, leaving an opportunity of fitting the characteristics of the equipment to the particular purpose for which it is to be used.

Closely associated with that question is another. Should the airworthiness requirements be so interwoven with the requirements covering particular types of operation as to ensure that no aircraft can ever be used in an unsafe fashion, for its particular characteristics, without violating a regulation? Or should we rely, at least in certain types of operation in which public safety is least involved, upon the user's own desire to keep within the limits of safe practice, or his instinct of self-preservation, and concentrate on getting for him the information that will enable him to decide what safe practice is?

These questions have been growing in importance; and they should command even more attention in the future than they have in the past; but they are by no means new. They were first presented by the first division of civil aircraft into categories, established much earlier in European countries than in the United States. With different standards of strength established for acrobatic and nonacrobatic aircraft, every purchaser of a nonacrobatic type was put on notice that if he engaged in violent maneuver it would be at considerable personal hazard. Should regulations then prohibit any use of the nonacrobatic types under circumstances from which a special hazard would have resulted? Or should government content itself with having warned the purchasers of the structural limitations?

During the past five years the Civil Aeronautics Administration, the Civil Aeronautics Board, and the aircraft industry have wrestled with a much more intricate problem of the same sort. In lieu of the rigid minimum performance requirements previously imposed on the certification of all new types, the new regulations for the transport category have introduced the measurement and certification of the performance actually realized under various conditions. The airlines are then to be required, in their operation of aircraft so certificated, to adapt the conditions of service and the loads carried to the verified characteristics of the aircraft.

Up to this time, flexibility has been introduced into the type certification requirements principally for the transport category. There is, I believe, every reason to extend it to other categories as well. It has drawbacks – notably the drawback of complexity of initial test – but I think them outweighed by the advantages, whether the problem be regarded from the point of view of the manufacturer of aircraft, the user, or the general public.

Before going further I should emphasize the purely personal nature of these views. I am one member of a board of five, and I am speaking for no one except myself. From a long interest in the subject, however, and from a certain personal share in the present responsibility for its disposition, I have welcomed the opportunity of bringing some suggestions forward for discussion. If those who disagree with them will propose alternatives, or alternatively find arguments in favor of keeping to the present course without change, the paper will have accomplished its major purpose.

Governmental responsibility for the assurance of safe conditions for the users of aircraft is at its maximum in regular passenger transport services. There, more clearly than anywhere else, the establishment of regulatory requirements and prohibitions that will impose on the operator a definite mandate to keep well within the safe capacity of the particular aircraft used is justified. In private flying, on the other hand, the owner can properly be left much freer to decide for himself how he will fly, provided he creates no grave public hazard and no hazard to other users of the air space. In private flying I believe it more appropriate that, if performance characteristics are to be measured, the results should be given to the owner for his guidance, in preference to writing his manner of use of such data into a fixed code which will hang over him with the force of law.

My own concept of the proper scope of future equipment regulation is that it should be framed:

1. To prevent the manufacture and sale of aircraft presenting an inherent and inevitable hazard.

2. To ensure that the purchasers of aircraft will receive adequate and accurate information on characteristics important to safety.

3. To require, in cases especially pertinent to the public interest, that the conditions under which equipment is used should be so related to the equipment's characteristics as to maintain proper margins of safety.

That is all. If there is any paragraph, section, or line in the airworthiness regulations that cannot be shown to promote the realization of some one of those objectives, I favor striking it from the book; and I would add the general

overriding precept: "When in doubt between two courses, choose the less restrictive."

Regulation for Nontransport Types

The possible advantages and drawbacks of increased flexibility of requirement can best be examined in terms of a specific case. The certification of nontransport aircraft, for which no actual changes of regulation with the object of giving increased flexibility have yet been made, will provide an example. Let me select a few of the more important requirements for re-examination.

Such aircraft, whether for private or commercial use, must now be analyzed for the ability to sustain a 30-ft vertical gust at maximum design level speed, V_L. A flexible element already exists in this connection, for V_L can be chosen at the manufacturer's discretion, provided only that the aircraft shall be placarded against exceeding the chosen figure in level flight. It need not correspond to the speed at any particular fraction of maximum engine power; and if the manufacturer elected to have his aircraft analyzed for the 30-ft gust only at a speed corresponding to the use of less than half of the full power, there is nothing in the regulation to prevent his doing so. The maximum speed at which the aircraft is to be checked for a 15-ft gust, V_g, on the other hand, is related to V_L and to the terminal diving speed of the aircraft by fixed formulas.

All aircraft are liable to encounter rough air on occasion. Inability to survive the resultant stresses would certainly be a universal and inescapable hazard. Gust velocities in excess of 15 fps are unlikely to be encountered, however, except in conditions which would have led a competent pilot to suspect rough air and to conduct himself accordingly. It is recognized practice to reduce speed in rough conditions, even to well below the maximum safe value for a 30-ft gust. If the pilot is able to keep below an arbitrarily specified value of V_L in rough air, as the present regulations imply that he will, he could certainly keep below an equally arbitrary value of V_g when the air is smoother. I suggest the possibility that it would be logical to leave V_g quite as flexible as V_L; to let the manufacturer select both; and to require simply that the operating handbook furnished to each purchaser of the aircraft should indicate the maximum indicated air speed for which the structure had been verified for gust velocities of 30 fps and 15 fps, respectively. The pilot with good sense would then keep to below the lower of the two speeds when there was any likelihood of rough air, and never exceed the latter at any time, whether the point was covered by operating regulation or not.

In one respect, present flexibility in determining design level speed may be excessive. The difficulty of holding very closely to a desired speed in turbulent air is universally recognized; and there is consequent danger even for the most skillful pilot, if the allowable range of speeds between stalling speed and design level speed is too small, of being inadvertently carried beyond its limits. There should be some assurance that the stated speed limits could be maintained in practice; and it might be good practice to require that the speed for which the structure is checked for a 30-ft gust should exceed the stalling speed with flaps and other variable elements in the cruising condition by at least 80 mph, or some such margin. Similarly, the speed for that condition might be required to be at least equal to that at which a rate of descent of 2000 fpm is secured with engine throttled and landing gear extended, so assuring the ability to get down through rough air in a hurry without structural danger. In all other respects I believe that the determination of the speeds at which the aircraft would be designed to meet the gust conditions might well be left to the designer, and the decision on whether to accept the implied limits on operation be left to the purchaser, or separately covered by operating regulations for particular types of commercial operation.

Maneuvering Load Factors

There is, I believe, universal agreement that some flexibility is desirable in the requirements for maneuvering factors. Although American airworthiness requirements have long held to uniform standards, with the required values dependent only on the size of the aircraft and its power loading, a tentative decision to recognize an acrobatic category was taken several years ago and work on the development of its standards had made some progress before wartime business interrupted. The establishment of two or more specifically identified categories, with a rigid set of requirements for each, is in accordance with all the precedents elsewhere. There is, however, a possible alternative. The manufacturer might be left to pick strength factors for himself, anywhere along a scale varying uniformly from the minimum factors considered safe under any conditions to the maximum required for the most violent maneuvering imaginable; and the values chosen by the designer and verified by the Civil Aeronautics Administration would then become a part of the certification. Obviously "acrobatic flying" describes no single condition, and no single intensity of load upon the aircraft. Though two categories are certainly better than one, they still cannot fit the needs of all cases at all closely. The private owner who is firm in his intention of never using his aircraft except for cross-country transportation may be quite content with an ultimate maneuvering factor of 5. A graduate of Fighter Command, with a taste for snap rolls and outside loops, may want to buy a machine certificated for a factor of 12. In the wide range between the two lie a variety of other cases, including that of the owner who may merely want to feel comfortable in tight spirals, or gentle dives and pullups, and who would like a factor around 6.5. I suggest that manufacturers might be given the opportunity to make their own decisions about the degree of acrobatic capacity for which there will be substantial demand, and proceed accordingly, subject only to a minimum value below which it may appear that no type of flying could be safely conducted; and that private purchasers might then be left to make up their own minds about what they wanted to do with an aircraft and to pick a model of which the specification showed appropriate strength characteristics. Commercial users could, of course, be obligated by operating regulations to pick an aircraft compatible with the requirements of the intended service. For the giving of advanced acrobatic training, for example, it might be required that the machines used should have an ultimate maneuvering factor of at least 9 – whereas for simple cross-country charter service 5.0 might be enough.

If complete freedom of choice of load factors goes too far, we might consider three categories, probably exacting ultimate maneuvering load factors of about 5, 7, and 10, respectively. I believe, too, that the very rapid variation

of required load factor with size and power loading, which at present increases the ultimate value from 5.4 for an 8000-lb machine to 6.8 for one of 2000-lb weight at the same power loading, or from 5.4 to 6.5 with variations of power loading at a constant weight, ought to be re-examined. Perhaps we ought to give less weight to the qualities of the airplane, in determining load factors, for the characteristics of the pilot have more to do with determining the factors that will actually be applied.

■ Flying Qualities

Certain minimum stability and control standards, which can be attained by good design without handicapping the aircraft otherwise, certainly ought to remain as universal requirements; but they ought to be supplemented by additional information on the characteristics of the particular airplane. The regulations now require for aircraft developed in future for transport use (CAR 04.755T):

"There shall be furnished with each airplane a copy of a manual which shall contain such information regarding the operation of the airplane as the administrator may require, including

"(d) A discussion of any significant or unusual flying or ground-handling characteristics, knowledge of which would be useful to a pilot not previously having flown the airplane."

I suggest that the obligation to supply such a manual should apply to all types of aircraft, and that the checking of the major items of its contents should be a part of the routine of every test for a type certificate.

■ Take-off, Landing, and Climb

Performance characteristics, on which so much has already been done to secure flexibility in the transport category, present a particularly strong case for flexibility elsewhere. The present take-off requirement assumes that an airplane that can get off the ground in 900 ft is typically safe, while one which requires 1100 ft is typically dangerous; yet the take-off areas that may be used vary in length from 800 to 8000 ft, and in condition from something bordering on a bog to the smoothness and firmness of concrete. There have been cases of particularly fine pilots being killed because they attempted take-offs from fields too small to be admissible even with aircraft requiring extraordinarily short take-off runs. There is, on the other hand, no inherent danger in a machine that requires a 1500-ft run to clear the ground, provided it is used only from runways of appropriate length and allowing a proper margin of safety. The present regulation is a survival from the remote past. We have abandoned any maximum limit on take-off distance for transports. Its retention for the nontransport types is an anomaly.

The alternative would be an official certification of the actual take-off distances under standard conditions, to clear the ground and also to reach an altitude of 50 or 100 ft. If I were a manufacturer I should not expect to find a very rewarding market for a private-owner aircraft that required more than 1500 ft to reach a 50-ft altitude; but that is a problem of market analysis, rather than one of establishment of safety standards.

The case of stalling speed is not so clear. The pilot can choose his own point of take-off, but his landing-place may be imposed by factors beyond his control. As the landing speed and landing run of an aircraft are increased, the hazard that it presents to the public on the ground in the event of a forced landing increases more than proportionately. Also, an aircraft making an airport approach at very high speed in bad weather creates a hazard for other traffic.

After much discussion, it was concluded two years ago that the maximum allowable stalling speed for transport aircraft should be 80 mph in landing condition, with flaps fully extended, and 85 mph in the approach condition, with the flaps sufficiently retracted to allow a reasonable rate of climb. The maximum safe speed of bad-weather approach for an aircraft with a single pilot is certainly somewhat lower than for a machine with a two-man crew. I suggest that it might be reasonable to allow the manufacture of nontransport aircraft with power-off stalling speed up to a maximum of either 70 or 75 mph for a flap condition corresponding to a full-power rate of climb of at least 300 fpm. I am not suggesting that I would expect any considerable demand for aircraft with such a stalling speed as that for personal use. I should, on the contrary, be distinctly surprised if such a demand were to develop; but manufacturers would decide for themselves whether the number of ex-fighter pilots wanting to buy such machines was actually large enough to support their production.

For most practical purposes landing distance is more important to the pilot than stalling speed. If the practice of measuring important data and including them in the certificate is to be applied at all, it certainly should be applied to landing runs, or to the distance to come to rest after passing a 50-ft or (preferably) a 100-ft altitude.

If a minimum standard for rate of climb is to remain in general effect, as I believe that it should, it ought to be established at a value considered to represent the minimum at which any type of flight could be safe. That, in turn, would presumably approximate the figure corresponding to ability to maintain altitude in such turbulence as every pilot must expect to encounter from time to time.

There has been too little research on the effect of turbulence on performance. There are great gaps in our knowledge of the subject, which ought to be filled; but while working under present disadvantages in that respect I would estimate a reasonable minimum as 200 fpm under standard conditions at an altitude of 3000 ft, approximately corresponding to an altitude of 1000 ft at 30 F above standard temperature. The actual verified rate of climb ought to appear in the operating manual, with the other information already described, if the practice of collecting performance data particularly affecting safety and presenting them to the user under government certification is to be extended as I am suggesting.

I am very conscious indeed that the practice of determining and certifying critical performance data would add considerably to the length and expense of the certification tests. I am not advocating the more general adoption of this practice in light-hearted disregard of that fact. The regulations adopted last year for the transport category will greatly increase the burden and expense of certificating new transport types. I believe, nevertheless, that both manufacturers and operators will generally agree that the new body of regulations in that case is better than its predecessor – enough better to justify the added burdens. The extension of similar principles of regulation to aircraft of nontransport types will be justifiable only if the change presents such advantages to the public, to the industry, and to the users as to more than offset its complications.

Operating Restrictions in Air Commerce

I have already noted my own belief that the primary purpose of airworthiness regulation for the private-owner market is to give the purchaser accurate information by which he can be guided in selecting an aircraft and in subsequently making his personal decisions, from time to time, on how he will use it. The same types of aircraft would in many cases be used for commercial purposes, but they have then to be discussed in terms of a larger governmental responsibility. The discretion of the owner may be supplemented by a requirement.

I shall not burden the paper with great detail. A few simple illustrations will suggest the general form that operating regulations might take.

Take, for example, the operation of a flying school. I suggest that it might be appropriate to require, among other things, that:

1. The aircraft used for primary training should have ultimate maneuvering load factors of at least 6.5; and those used for acrobatic training, at least 9.0.

2. Aircraft used for solo practice by primary students should have a stalling speed of not over 50 mph and a rate of climb of at least 400 fpm at an altitude of 2000 ft above the field level.

3. Training operations should be conducted from a field large enough so that the aircraft used, making a still-air take-off under standard conditions, could reach an altitude of 200 ft by the time of passing the boundary of the field.

Charter operation presents another case. It has never yet been directly regulated, but there has been much discussion of the form that regulation might take; and I believe there is wide, if not by any means universal, approval of the establishment of some sort of standard for such services. The problems are in some respects akin to those of transport. The characteristics of the fields used cannot be introduced as a factor as readily as in regulating scheduled transport, however; for the charter operator may be called on to go anywhere on a moment's notice, and there is no time for exhaustive analyses of the relation between the characteristics of the proposed landing area and those of the aircraft used. A large proportion of the total number of take-offs and landings will be made at the operator's home base, however, and that at least may be judged by standards based on the performance of the aircraft.

Some possibilities of reasonable safety regulation of that type of service might be:

1. Aircraft to have an ultimate maneuvering load factor of at least 5.0; and to be designed to withstand a 30-ft gust at a design level flight speed at least 100 mph above stalling speed.

2. No use to be made of single-engine aircraft having a stalling speed in excess of a figure ranging from 55 to 65 mph, the exact value depending on the area in which the operations are principally to be conducted and the prevailing altitudes and terrain conditions there.

3. Only such aircraft to be used, and only with such loads, as will permit attaining an altitude of at least 150 ft by the time of passing the boundary of the field at the operator's base, following a take-off under standard conditions.

Any such code would leave it open to the manufacturer to fit his product to the worst conditions that have to be encountered anywhere, thereby making it good for all other conditions as well. No doubt that would be the typical preference; but if a particular manufacturer should believe it worth while to produce a model fitted only for the larger fields, the more open country, and moderate operating altitudes, and thereby to give prospective purchasers who operate over such terrain the advantage of greater economy, should the Government deny him the opportunity of building such an airplane and looking for a market?

I have heard some argument that all charter operations should, in the future, be conducted with multiengine aircraft. The effect of such a regulation, at least as applied to existing aircraft, would be to decimate the total amount of charter business by so increasing its cost as to make it unavailable to many who might otherwise use it. It seems a more reasonable course to seek to establish conditions under which single-engine aircraft may be used with proper safety, even though in so doing it may prove necessary to develop a code of a certain degree of complexity.

Vision

In one respect, at least, airworthiness standards of the future will not only have to be rigid, but more stringent than in the past. Congestion in the air, which was a rarity before the war, is going to be a much more common experience when civil flying is freed from restriction. The avoidance of collisions will demand both adherence to traffic patterns and a proper lookout for neighboring aircraft; and it will demand aircraft from which such a lookout is possible. There will be no place for private-owner types with vision so bad that it is possible to run down another plane without ever having had a chance of seeing it. I am not ready to venture a personal recommendation in terms of a definite specification of visual angles; but I am sure that it ought to be physically impossible for two aircraft to be in such a relation to one another that they can come into collision, while traveling along straight or moderately banked courses, without either pilot ever having been able to see the other. I suggest that the lowest possible line of forward vision for the pilot ought to be depressed at least 5 deg below the flight path at the maximum probable angle of attack, whether looking straight ahead or diagonally across the cowl in a machine with side-by-side seating; and that in taxiing the pilot should be able to see the surface of the runway not more than 100 ft in front of him, both straight ahead and for a considerable distance to each side.

Scheduled Transport

Finally, I return to the transport category. Three main questions of equipment regulation seem likely to seek consideration there within the next few years, as a supplement to the work already done between 1938 and 1942.

1. Should predictable variations from standard atmospheric conditions, and predictable differences in other characteristics of landing areas than their dimensions, be taken into account in determining the loads that particular aircraft may carry?

2. Should the general requirements of the transport category, originally written for large aircraft, be modified in application to small machines used in local transport services?

3. Should present requirements be considered in any

respects as temporary standards, to be raised further as the development of the art permits?

The first question is typified by the case of temperature variation. Regulation for the transport category now require that in the event of the failure of an engine at the most critical instant of the take-off it shall be possible either to bring the airplane to rest within the confines of the take-off area or to continue the flight, reaching an altitude of 50 ft by the time of passing the field and thereafter clearing all obstacles along the course by at least 50 ft. The 50-ft margin of altitude incorporated in the regulation serves several purposes; but its primary purpose is to allow for the contingency that the emergency may arise when conditions are substandard, or that the pilot may fail to get from the airplane the performance that it attained in the certification test.

Temperatures above standard are among the commonest of adverse conditions. The effect of a rise of temperature to 30 F above standard is typically to increase the ground run of a representative twin-engine transport of the present day by about 125 ft, and reduce the single-engine rate of climb by about 60 fpm. Results of calculations for a typical transport[1] on those assumptions are shown in Table 1.

Table 1 – Effect of a Rise in Temperature to 30 F above Standard

Weight (as a Fraction of Normal Take-off Weight)	Performance Under Standard Conditions at Sea-Level, after Failure of One Engine at the Instant of Leaving the Ground				Estimated Performance at Temperature 30 F Above Standard	
	Take-off Run	Rate of Climb—Landing Gear Extended	Rate of Climb—Landing Gear Retracted	Total Distance to Reach a Height of 50 Ft	Total Distance to Reach a 50-Ft Altitude	Altitude That Would be Reached in Distance Which Would Suffice for Reaching 50-Ft Altitude Under Standard Conditions
0.90 W_o	1200	290	400	2450	2900	37
1.00 W_o	1500	140	270	3800	4650	30
1.05 W_o	1700	70	210	4800	6100	26

The effect of a 30 F temperature rise in this case is to reduce the margin of altitude at the edge of the field by approximately a half. Alternatively, the effect is to reduce by about 4% the gross load that can be carried out of a field of given length, while still attaining an altitude of 50 ft at the time of crossing the boundary of the field after having suffered the failure of an engine at the most critical point. The angle of climb at normal load, after passing the boundary of the field, would also be reduced from about 1 in 33 to 1 in 42.

The margin with respect to the consequences of a critical engine failure during take-off, at a given gross load, is greater in spring or fall than in midsummer. For take-off from a runway of given length, the summer margin would be greater in some parts of the country, where inordinately high temperatures are unknown, than in others. To attain the closest possible approach to a uniform standard of safety, the maximum allowable take-off loads would vary with temperature. Obviously it would be impracticable to tie such a variation to the temperature actually existing, since it could not be known at the time of planning the loading of the aircraft; but it would be feasible to allow for the variation of average temperature with locality, or from season to season through the year. The July average temperature in the United States, for points close to sea-level, ranges from 58.9 F at San Francisco and 64.2 F at Seattle to 82.5 F at New Orleans and 84.2 F at Dallas. There is the obvious question of the reasonableness of making take-offs from the latter pair of points with the same load that would be established as the maximum for the same aircraft and for fields of the same length at the former pair.

The present regulations have been in force for too short a time to consider change unless a real emergency need for it develops. They represent a substantial increase in margins of safety over those which have previously existed. When it appears reasonable to consider a further revision, however, I believe that serious consideration should be given to taking the mean temperature of the locality and season into account as a factor in the take-off requirements.

When that time comes, consideration can also be given to an improvement in the present criterion of margin of safety. The effects of changes in temperature, of the occurrence of turbulence, and of imperfections in piloting as compared with the technique used in proving the airplane for certification, are all best measured in terms of loss of rate of climb. A 50-ft clearance at the edge of a 6000-ft field, on the other hand, represents only half as great a margin, measured in terms of the amount of rate of climb that can be lost without causing an accident, as the same 50-ft clearance at the end of a 3000-ft take-off strip. If I were to write my own idea of the specification that may prevail five or ten years from now, it would be very like the present one except that in place of the requirement of a 50-ft clearance at the edge of the field I would put a requirement that the airplane must be able to clear the edge of the field, and all obstacles thereafter by at least 20 ft, in spite of having lost 100 fpm from the rate of climb shown to be attained under test conditions at the altitude of the take-off point and at the temperature normally to be predicted there. That would require a rate of climb under standard conditions, with one engine inoperative, landing gear retracted, and propeller idling, of from 200 to 350 fpm, depending on the season and upon the angle of climb required to clear obstructions after passing the boundary of the take-off area. The present regulations may require rates of climb over almost exactly the same range, but their variation is now dependent upon the stalling speed of the aircraft in its take-off condition. The suggested form would also require a take-off strip of about the same length as the present rules demand.

The present requirement of a rate of climb of 0.035 V_s^2, immediately following one-engine-inoperative take-off, would then become irrelevant and unnecessary.

Much the same reasoning on the effect of temperature applies to flight en route in case of the failure of an engine. In that case the present requirements are for a rate of climb with one engine inoperative of approximately 120 fpm (the exact value varying with stalling speed) at an altitude 1000 ft above the highest point that has to be crossed en route. This, too, represents a substantial advance over the preceding requirements; but if temperature

[1] Made from basic data furnished by C. W. Dycer and Omer Welling, Flight Engineering and Factory Inspection Division, Civil Aeronautics Administration.

runs up to 30 F above standard, the margin of climb of 120 fpm is likely to be reduced to less than half that. It would again seem reasonable to look forward to a time when a reasonable approximation to the temperatures likely actually to be encountered will be substituted for a slavish adherence to standard atmospheric conditions determining in the effects of these requirements upon allowable load.

After having had a couple of additional years to reflect on the matter since the present regulations were adopted, too, I have become very skeptical of the validity of the original conclusion that the rate of climb required for safe continuation of flight should vary with the stalling speed. I believe it would be more reasonable to require (in the present state of development) a flat 100 fpm of climb with one engine out of action, at 1000 ft above the highest point on the course and at the probable temperature for the region and season.

Airport Surface Conditions

Another variable element for which allowance might be made in determining the required field sizes and allowable load is the surface condition of the landing area, as influenced by the surfacing material and by the local climate. I will not take the space to develop that point at length; but it is obvious that the considerable margin of safety which now finds a place in the formula for determining the required length of landing strip is largely required to meet the risk that rain or snow on the surface will reduce the coefficient of braking friction. In some parts of the United States snow is unknown, and rain virtually never falls during a protracted dry season each year. The introduction of the worst probable surface condition at a particular airport as a factor in determining allowable landing loads would complicate the determination of such loads considerably; but I foresee that sooner or later we are going to be challenged by someone who will protest that the allowable landing loads at a point that particularly interests him are being determined by a rule that takes into account the probability of weather conditions such as never have existed at that particular point and never are likely to. I am afraid we shall not have any wholly satisfying answer to make to such a protest.

Possible Future Changes of Standard

In framing expectations for the future in any matter affecting safety, we start with the record already made. It has been remarkable. If the year-to-year fluctuations be eliminated and a smooth curve drawn for the last 10 years, the average reduction of the passenger fatality rate in the domestic airlines of the United States has been about 20% per year. It has been equivalent to a 50% reduction of hazard every three years, or a reduction of substantially more than 90% since 1929. To talk in terms of specific figures for specific periods, the average passenger fatality rate for the years 1929-1933 was approximately 16.0 per 100,000,000 passenger-miles; for the five-year period 1934-1938, it had been reduced to 7.2; and for the five years beginning with the effective date of the Civil Aeronautics Act and ending with Aug. 22, 1943, it has been only 2.5.

The very fact that there has been such spectacular improvement, and that so few accidents happen without an ascertainable cause which might conceivably have been avoided, gives us the confidence of further improvement. I am not optimistic enough to suppose that the trend of the last dozen years can be maintained indefinitely; but I see no reason why we should not expect that the fatality rate will be reduced to 1.0 per 100,000,000 passenger-miles within a short time after the end of the war.

If we are indeed to make further improvements we shall have to be on the lookout for practicable ways of raising standards, as the art progresses, in the matters in which reasonably possible concurrences of circumstances might be most likely to produce a fatal result. There are, of course, features of design and operation in which an increase of standard would make no contribution to safety, since they present no hazard now. To have built the Triborough Bridge to twice its present strength would not have increased its safety; and in the regulation of aircraft we have also to distinguish between the points at which the further elevation of standard would contribute to a still further reduction of hazard and those at which no such benefit would accrue.

It is clear from the accident reports of a number of years past that the major opportunities for improvement are to be found in air navigation facilities, communications, airports, weather forecasting, and operating practices. Equipment factors have been involved in a few cases; and the possibility of further gains for safety through improvement of equipment cannot be overlooked, especially as the qualities of the equipment and those of the radio or navigational facilities may be interlocked as causative factors. If an aircraft is harder to fly than it need be, the resultant strain on the pilot makes it more difficult for him to keep his judgment clear and use his radio effectively. It remains true, however, that the direct contribution of equipment failures to such hazard as still remains in air transport is a very minor one.

A cure for one of the worst afflictions of the past 25 years seems to be close at hand. De-icing by heat seems clearly on the way. It will need no regulation to secure its adoption. Operators will insist on it; and it promises to dispose of the aerodynamic problem of distortion of wing form, either by de-icer boots or by adherent ice, with consequent increase in stalling speed and change of stalling behavior and other flying qualities. With the airplane freed of those handicaps, full advantage can be taken of the benefits of low winter temperatures in permitting increases of load, above the loads allowed under standard conditions, without lowering the standards of safety.

The structural record of existing transport aircraft over the past 10 years has been good. While atmospheric conditions so violent that they would cause structural failures certainly exist at times, the hope for improvement there seems to lie more in better forecasting of such conditions, and in avoiding them, than in increasing the structural strength by enough to endure the worst storm loads. If any new specification were to be imposed at any time to cover the general strength of the structure, it might take the form of a requirement that the structure should be calculated to stand the load factor induced by a vertical gust of 45 fps at a speed at least 80 mph above the stalling speed, to ensure safety under conditions of really extraordinary turbulence over a speed range wide enough so that a pilot can expect to keep within it. Accelerometer records have shown that gusts of 45 fps exist; but they are very uncommon, and I believe that there is no record showing actual encounter with a gust of apparent velocity materially higher than that. Decision on whether it would

actually be appropriate to modify existing strength requirements to take account of gusts of higher velocity than are now assumed should be governed by the accumulation of still more experience, and in particular of still more accelerometer records of loads actually encountered in flight.

Although gust loads are likely to be more critical than maneuvering loads for future transport aircraft, I have some doubts about the latter category. When they are examined afresh in the light of many years' experience, I think we shall question the present variation of load factors with power loading, and also the present practice of setting the required factor 5% lower for seaplanes than for landplanes. If power loading is to be retained as a factor, wing loading probably ought to join it, for the worst maneuvering conditions to which a transport is likely to be subjected are related to the ratio of cruising speed to stalling speed. The liability of large aircraft to maneuvering loads is also likely to be connected with the type of servo control used. Ideally speaking, requirements with respect to maneuvering load factors ought to be connected more with control forces than with airplane size. I think every pilot would like to feel certain that the structural strength of his aircraft was great enough to stand the consequences of applying a 50-lb force to the control column in a pull-up; but that ideal can hardly be realized on small aircraft, and the factors governing control forces are hardly predictable enough to make it a feasible basis of regulation for large ones.

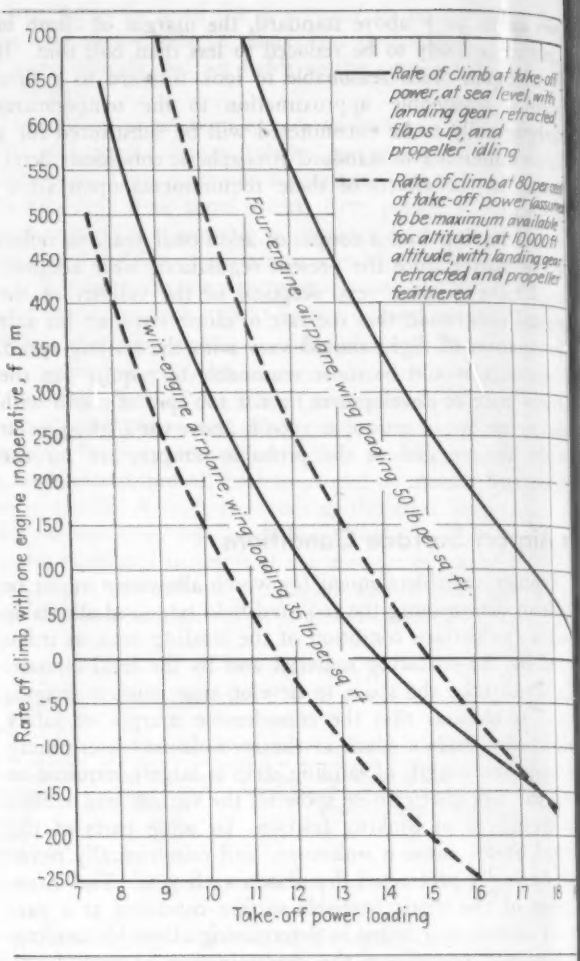
■ Performance Requirements and Economy

Economy is in conflict with the attainment of the very highest performance, or with the maintenance of large margins of performance above the figures required in ordinary operating conditions. A compromise is likely to have to be struck between an extreme of economy gained at the expense of safety, on the one hand, and an extreme of conservatism in seeking large margins of performance at the expense of economy, on the other. Before the location of the compromise solution can be intelligently discussed, we need figures both on the relation between economy and performance and on the relation of performance to certain specific possibilities of accident.

For purposes of illustration of the relation between economy and performance, I have assumed the two basic cases:

1. A twin-engine airplane with a wing loading of 35 lb per sq ft.

2. A four-engine airplane with a wing loading of 50 lb per sq ft. In each case the power loading is used as a variable; and in Fig. 1 the rates of climb with the critical engine inoperative have been plotted against the take-off power loading with all engines operating. Curves are drawn for the two cases of (1) take-off power at sea-level, with the inoperative propeller idling, and (2) maximum power at 10,000-ft altitude, on the assumption that the critical altitude for METO power is 8000 ft, and with the inoperative propeller feathered: corresponding to engine failure during take-off and to engine failure when en route through a mountainous area, respectively. The calculations are not intended to represent the best that is possible, but the probable result of using good present-day design practice. The rates of climb shown for various power loadings with the propeller idling agree almost exactly with those which C. W. Dycer and his associates in the Civil Aeronautics Administration have determined for the DC-



■ Fig. 1 – Rates of climb with one engine inoperative plotted against the take-off power loading with all engines operating

under the same conditions; and the present analysis therefore assumes enough progress in the years since the DC- was designed to make it possible to realize a single-engine rate of climb with a wing loading of 35 lb per sq ft equal to that which the DC-3 secures with about 30% less wing loading.

Increases in powerplant weight must of course be considered as directly subtracted from payload. In relating powerplant weight to power loading I have taken the total powerplant weight as 2.4 lb per take-off hp. On the assumption that additional power would be required only for the purpose of meeting single-engine requirements, as a safety factor, and would not be used to increase the maximum operating speed, the increase in structural weight resulting from increase in power is taken as 0.4 lb per hp. I have further assumed that for every 10% increase in power the fuel load carried will be increased by 0.3% of the gross weight of the aircraft, largely because of the increase of drag with enlargement of the nacelles.

On these assumptions, Fig. 2 shows the probable variation of payload over short distances (about 400 miles for the twin-engine plane, or 600 miles for the four-engine one) as a percentage of the gross load. Separate curves are shown for passenger and cargo operations, on the assumption that the elimination of passenger accomo-

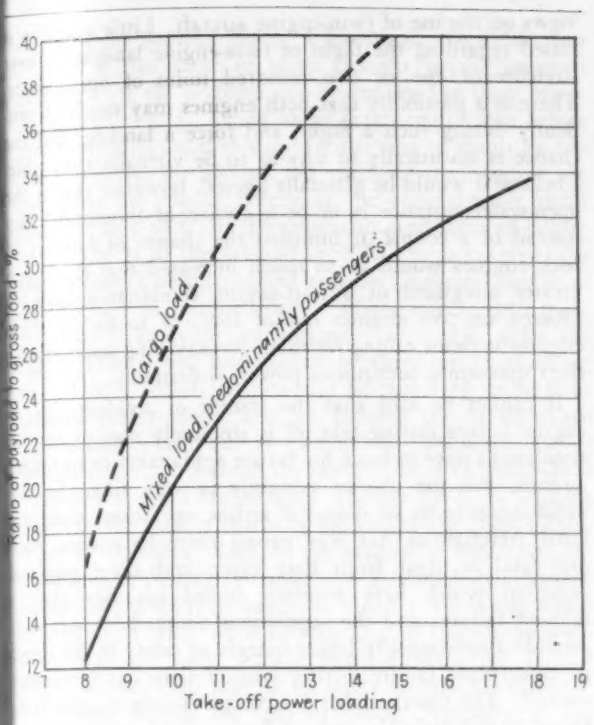


Fig. 2 – Probable variation of payload over short distances as a percentage of the gross load

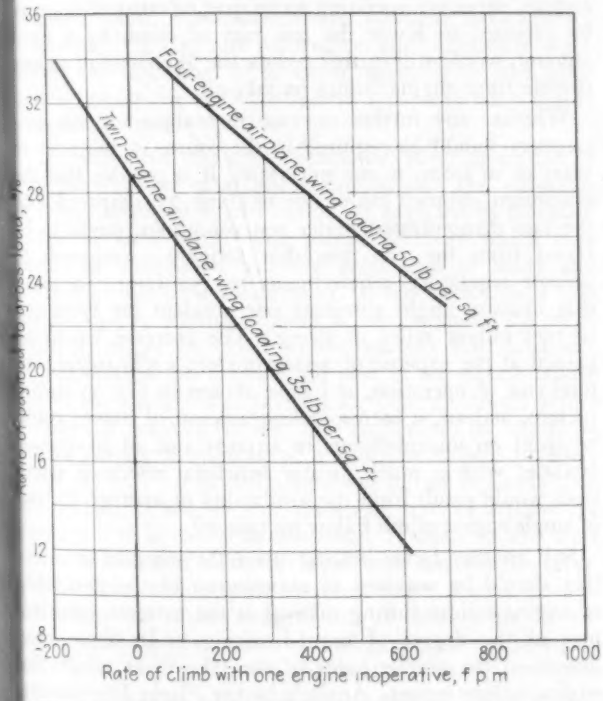

Four-engine airplane, wing loading 50 lb per sq ft
Twin-engine airplane, wing loading 35 lb per sq ft
Rate of climb with one engine inoperative, f p m

Fig. 3 – Anticipated percentage of payload in a passenger-carrying airplane as a function of the rate of climb with one engine inoperative immediately following take-off, with the landing gear retracted but the inoperative propeller not yet feathered

ions and cabin attendants will typically make it possible o carry about a third more payload if it is composed of argo alone than if there is a combination of passengers nd cargo, with the former predominating.

In Fig. 3 the data previously collected are combined to show the anticipated percentage of payload in a passenger-carrying airplane as a function of the rate of climb with one engine inoperative immediately following take-off, with the landing gear retracted but the inoperative propeller not yet feathered.

Curves are drawn for twin-engine and four-engine airplanes. Both are very nearly straight lines. For every 100 fpm of one-engine-inoperative climb that is sacrificed, the payload is increased by about 2.6% of the gross weight in a twin-engine aircraft and 1.7% in a four-engine one.

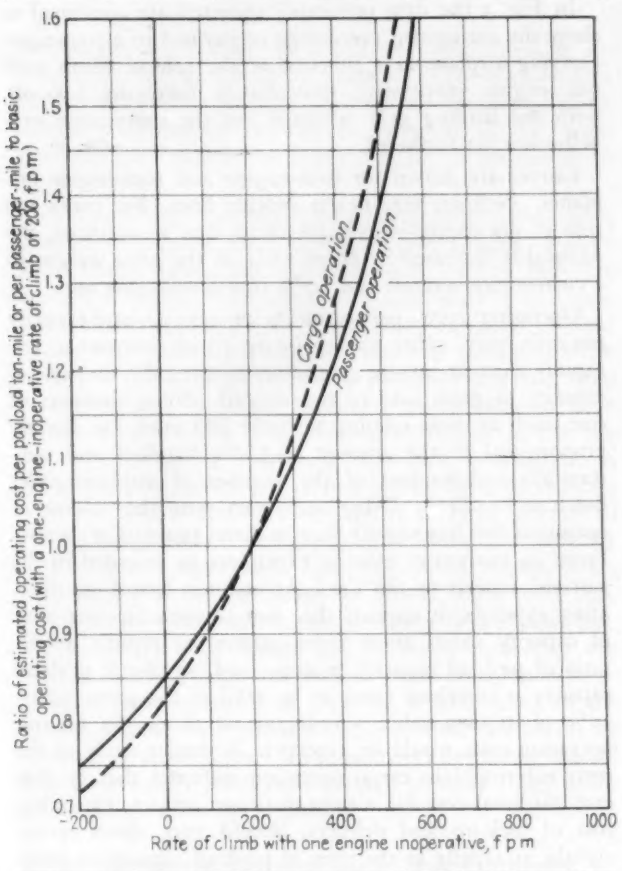
Operating costs per seat-mile or per payload-capacity ton-mile vary, other things being equal, somewhat less rapidly than in inverse proportion to the ratio of payload capacity to gross load of the aircraft. Some elements of cost, such as those relating to traffic and sales, are directly proportional to the amount of traffic handled and substantially independent of the number of airplane miles flown to handle it. Other items vary with the volume of operation, but less rapidly than in direct proportion thereto. From an analysis of existing experience as recorded in the carriers' reports to the Civil Aeronautics Board on operating expenses, it appears that cost of operation per unit of capacity varies about three-quarters as rapidly as the ratio of payload capacity to gross load, so that if payload capacity is increased from 20 to 22% of the gross, or by 10% of its own value, a reduction of about 7% in unit operating costs would be expected. A similar study of the costs entering into cargo operation indicates that in that case the total cost for airport-to-airport service, excluding cost of pick-up and delivery, should vary about seven-eighths as rapidly as the ratio of payload capacity to gross load.

Upon those bases Fig. 4 has been plotted to show the variations of operating cost with one-engine-inoperative rate of climb, other things being equal. No attempt has been made to give costs in absolute terms. Relative values are used. The curve shows the ratio of anticipated operating cost per passenger-mile or per payload ton-mile to the corresponding cost that would exist if the power loading were so modified to give the airplane a one-engine-inoperative rate of climb of 200 fpm.

In very round numbers, every increase of 100 fpm in one-engine-inoperative rate of climb may be expected to increase operating cost in passenger service by about 10% with twin-engine airplanes. The curves are drawn only for the twin-engine case, for a four-engine transport plane with a rate of climb of less than 500 fpm on three engines is not a very probable type; but if the same type of analysis be applied to the four-engine case, with the same range of rates of climb, the effect of increase in the one-engine-inoperative rate of climb on cost appears to be about half as great as for the twin-engine case.

■ Economic Penalties

Obviously, the economic penalty involved in making any unnecessary increase in the single-engine requirement for twin-engine aircraft would be very heavy. Even for the present volume of operations, an increase in single-engine rate of climb of 100 fpm would increase the total operating cost in domestic air transport by about $8,000,000 per year. On the other hand the operation of regular transport service, especially with passengers, under con-



■ Fig. 4 – Ratio of operating cost per passenger-mile or per payload ton-mile to the corresponding cost that would exist if the power loading were so modified to give the airplane a one-engine-inoperative rate of climb of 200 fpm

ditions creating any substantial and avoidable hazard would obviously be neither good public policy nor good business for the industry.

■ Statistics of Engine Failures

An analysis of all the reports of engine failures in carrier service made to the Air Carrier Inspection Division of the CAA in 1942 show a total of approximately four cases of complete engine failure during the take-off proper.[2] There was approximately one such failure for every 25,000,000 miles of flying. None of them caused accidents. There were also about six cases of rapidly progressive failures during take-off, due to loss of oil pressure or other causes, but without immediate stoppage; and there were approximately a dozen cases of complete engine failure during some part of the climb to cruising altitude.

In analyzing hazard statistically, consideration must be given to its probable frequency and duration. A hazard that becomes serious only once in every hundred million miles of flight may be accepted, whereas if its occurrence were 20 times as frequent as that the burdens of preventive measures would have to be assumed. A universally familiar example of such a distinction is found in the prevailing views on the use of twin-engine aircraft. Little question is raised regarding the flight of twin-engine landplanes over stretches of one or two hundred miles of open water. There is a possibility that both engines may stop independently during such a flight, and force a landing, but the chance is statistically so tiny as to be virtually negligible. I believe it would be generally agreed, however, that if the open-water distance is to be a couple of thousand miles instead of a couple of hundred the chance of failures of both engines would be so much increased that the much greater safeguard of a four-engine combination able to proceed on two engines out of four, or to fly on three engines without calling on them for anything approaching their maximum continuous power, is desirable.

It cannot be said that the chance of accident due to engine failure during take-off is absolutely zero under the regulations now in force for future application to transport aircraft, but the chance certainly is very tiny. In the 525,000,000 miles of domestic airline operations since the Civil Aeronautics Act was passed there have been only one fatal accident from that cause and three nonfatal accidents which have involved forced landings after a take-off failure; and the regulations adopted in 1942 will provide a substantially larger margin of safety in the event of powerplant failure during take-off than has heretofore existed. The changes that I have previously suggested as being in the desirable path of future development, taking account of probable temperature conditions in place of using a fixed year-round standard, and assuming that turbulence, slightly subnormal power of the remaining engine, imperfect operating technique, or other causes may be expected to lower the test rate of climb by a fixed amount, would still further reduce the likelihood of serious trouble from engine failure on take-off.

Whether any further increase in margin for this contingency should be required in the course of the next 10 years or so seems to me uncertain. It is possible that the maximum assumed loss of rate of climb, as compared with the rate demonstrated under test conditions, ought to be raised from the 100 fpm that roughly corresponds to present regulatory requirements to 150 fpm. To make that increase might eliminate one accident for every one to two billion miles of flying. The increase would be gained at the expense of approximately a 5% increase of total cost of operation, as I have shown in Fig. 4; and at present, at least, a corresponding amount of money could be spent on improvements to airports and air navigation facilities with a much greater beneficial effect on safety than would result from the availability of another 50 fpm of single-engine climb following take-off.

Still another factor bearing upon the margins of safety that should be required in anticipation of the possibility of engine failure during take-off is the extreme sensitiveness of the degree of hazard arising under those conditions to the precise point of the take-off at which the engine failure occurs. Analysis by the Flight Engineering and Factory Inspection Division of CAA[3] has shown that under the regulations adopted in 1942 a DC-3 with engines rated at 1200 hp for take-off would require a 4000-ft take-off strip in order to reach an altitude of 50 ft at its end if the failure of one engine occurred after 1500 ft of

[2] The figure has to be an approximation, as the reports do not always make it clear how suddenly the failure developed or at just what point of the take-off and subsequent climb it occurred.

[3] See "Determination of the Take-Off Flight Path Defined by CAR 04.7532T for the DC-3 S1C3G Airplane (Including a Method to Correct Test Data to Standard Conditions)," Flight Engineering Report No. 7, Oct. 1, 1942.

run from the starting point. I have used the CAA data to examine the effect of some changes in the assumed conditions. If the performance be assumed to drop below standard to such an extent as to lengthen the run before take-off by 10%, and to decrease the rate of climb by 100 fpm thereafter, the altitude on reaching the edge of the field, after an engine failure at the same 1500-ft point would be only 10 ft. If the engine showed signs of failure before 1500 ft of the take-off strip had been covered, the take-off should be interrupted and the airplane brought to rest. If, however, sudden failure of the engine took place after 1800 ft from the starting point instead of 1500, the altitude that would be attained by the time of reaching the end of a 4000-ft strip would be increased to 89 ft at standard performance, and 37 ft with the performance hypothetically reduced as previously indicated. Every additional 300 ft by which the occurrence of engine failure is delayed is good for roughly another 40 ft of altitude gained by the time of reaching the end of the 4000-ft field. So far as the initial phase of the take-off is concerned, up to the time of passing the obstructions immediately bordering the field, a delay of 400 ft, or about 3 sec, in the actual point of occurrence of a sudden engine failure, as compared with the most critical possible point for such a failure, increases the margin of safety enough to offset the effects of a loss of 100 fpm in climb. Similarly, if one engine begins to lose power after 1500 ft of take-off run, but instead of failing instantaneously and completely develops an average of half power for another 6 sec before failure, the margin of safety would be increased by a similar amount. Instantaneous and complete loss of power is fortunately a very rare occurrence; and the present regulations, in assuming it to occur precisely at the most critical instant, are ensuring against a possible but a highly improbable event.

The standards of safety maintained in common-carrier transportation are of almost unique rigor. The standards of other classes of service cannot be expected to maintain quite the same level. The precise level at which they should be established needs a great deal of study and discussion; but, as a contribution to the discussion, I will suggest that an alternative basis of regulation for the cargo transport might be based on the analysis just described, by substituting the hypothetical case of an engine failure five or ten seconds after take-off for the exceedingly severe assumption, developed for the regulation of common-carrier passenger services, of a complete failure precisely at the most critical instant. Some loss of rate of climb, such as 100 fpm, below the rate shown on tests of a prototype ought still to be taken into account. If this standard were adopted, it would allow the operation, from fields favorably situated with respect to surrounding obstructions, of aircraft having a single-engine rate of climb with take-off power, landing gear retracted, and propellers feathered, of as little as 120 fpm under the expected temperature conditions. Operations with as low a rate of climb as that, even in cargo service, would have to be limited to fields where the take-off could be made without passing over any obstruction more than 50 ft high within two miles of the point of starting the take-off, or anything more than 75 ft within four miles.

If any lower standards than that were to be set for the carriage of cargo, it would have to be with the recognition that the operation was being brought very close to single-engine-aircraft status. There could be no confidence of ability to continue the flight after an engine failure at any time. There are an enormous number of occasions, in the operation of single-engine aircraft, on which the sudden failure of the engine would cause an inevitable crash. The rarity of accidents from that cause is a high tribute to present-day engine reliability. In operating cargo aircraft out of fields bordered by open country, agricultural land, or undeveloped areas, it would no doubt appear allowable to accept a very strong probability of a forced landing in the event of engine failure in the first 30 sec or a minute of flight. It is not to be expected, on the other hand, that heavy aircraft will be allowed to operate in the face of any such prospect from fields where the take-off carries them over thickly built-up districts.

The margins of safety with respect to one-engine-inoperative performance en route also present some questions for the future. In that case, too, the records of the recent past have been good. There have been only two accidents (neither one fatal) on the domestic airlines in the last five years which could be attributed to failure of an engine after the completion of the take-off and inability to continue flight thereafter (excluding cases in which the accident was primarily due to other troubles which an increase in single-engine performance would not have overcome). There have of course been such cases in military transport operation, including a very recent and dramatic one resulting in a forced descent of an entire ship's company by parachute in Burma.

Nevertheless there is always the possibility of an engine failure in highly turbulent conditions, when a substantial margin of power will be necessary if altitude is to be maintained; or of weather conditions requiring the maintenance of an altitude of substantially more than the 1000 ft above the highest ground level which the present regulations require. It certainly would be very convenient at times to have increased margins; but past experience makes it evident that the cases in which such an increase would be of decisive benefit are very rare indeed, and under those circumstances it is again necessary to reckon with the cost of providing it. Again offering my own tentative opinion as of this moment, it seems to me that the present standards, modified as I previously suggested to take account of probable temperatures, will remain adequate for the case in which a relatively short stretch of high ground has to be crossed during a flight; but that in connection with any condition which is to endure for more than 40 or 50 miles, as in flying over terrain of constant altitude without fully adequate landing spaces at short intervals, the ultimate margin of climb with one engine inoperative should be at least 250 fpm at an altitude at least 2000 ft above the ground level.

For cargo aircraft a similar distinction between the requirements for passing over a range of mountains and those for flight over an extended plateau at the same altitude would seem appropriate; and I suggest that the minimum allowable rate of climb in each of these cases might properly be about 100 to 150 fpm less than for passenger operation under similar circumstances.

■ Stalling Speed

Wartime experience and the desire for higher speeds will undoubtedly accentuate the desire, already manifest from some designers, for an authorization to increase stalling speed. The permissibility of such an increase

seems more a matter of navigational method and of operating procedure than of airplane control or ability to land. The critical effects of increased stalling speed are in its reflection upon turning radius and upon approach speed. The critical factors are the reluctance of the pilot to have time slip by any more rapidly than it does at present during the passage from the radio-range cone of silence to the contact with the field, and his reluctance to have to accept an increased radius of turn while maneuvering about the field by visual contact. As long as those factors remain critical, there is little that can be done by the airplane designer to justify an increase in stalling speed. I have every hope, on the other hand, that improvements in radio and in instrument landing means will justify such an increase after the war, and that higher stalling speeds than are now allowed will not only be reconcilable with the present standard of safety but with a considerable increase in that standard; but that cannot be settled by speculation and hope. The improvement in landing aids will have to be proved by demonstration.

I hope that I will not appear an incurably reactionary if I suggest that increase of stalling speed does not seem conducive to increase of regularity of service, other things being equal, and that improvement in regularity is of great practical importance. I am very doubtful that it would be as easy to put instrument landing into practical daily use with aircraft stalling at 90 to 100 mph as with machines having a much lower stalling speed; and if there is in fact any conflict between increase in regularity and increase in stalling speed, it ought to be resolved in favor of the former. With all the great advantages and merits and achievements of air transportation, it still has the Achilles heel of being grounded by weather in which other means of transportation are able to operate. The percentage of scheduled mileage completed by the domestic airlines as a whole has increased more or less uniformly from 84 in 1932 to 94 in 1942. The proportion of disappointed would-be travelers has been cut by half; but there is need for still further progress, and from the point of view of a habitual air traveler I should say that every additional increase of 1% in proportion of trips completed would be worth as much to the patrons as an increase of 20 mph in cruising speed.

■ Problems of Small Transport Aircraft

Although it is the giant airplane that catches the imagination, there is a vast mileage of potential air routes all over the world which cannot justify very large aircraft, or perhaps even the sizes now commonly used, but could make good use of transport planes of from 5-passenger to 10-passenger capacity if such machines can be operated economically enough to keep the necessary commercial charges within the range in which traffic can be attracted. Experience with the cost per unit of payload in operating small aircraft in transport service up to the present time has not been very encouraging; and at least it is clear that there are many regulatory and other burdens which can be borne without serious economic effects by 20-passenger or 40-passenger aircraft, but which would be absolutely disabling if imposed on machines of much smaller size. There will be many routes on which a choice will have to be made between having no air transportation at all and having air transportation of a less appealing type than major trunk lines can support, and without all of their safeguards – precisely as passengers are carried at sea, especially on the less traveled routes, on vessels that lack many of the refinements and safety devices installed on the ships that ply a few of the major trade routes.

The regulatory authority has the problem, in such cases, of deciding whether communities that cannot support the highest grade of transport service should be denied any transport service at all, or whether they should be allowed to receive service which is in some respects operated to lower standards than the heavy-traffic routes can maintain.

The problem of deciding what type of operation it is economically feasible to maintain has already been presented in very practical form in various parts of the world outside the continental United States. There are many existing airlines which are rendering vital service to large areas, but which would have to suspend operations if they were required to adopt in all respects the standards maintained by American carriers.

■ Single-Engine Aircraft

The extreme possibility would, of course, be the use of single-engine aircraft. The use of single-engine aircraft by private owners will no doubt continue. They have been used for most charter service; and although, as I have previously mentioned, there is some challenge of their future use in that connection, I think it would be most unwise to terminate it in any clearly foreseeable future. They have been largely used by the airlines in Alaska, under very difficult conditions indeed. The possibility of their approval for widespread use in common-carrier operations which it might appear impossible to maintain otherwise raises important questions. Discussion, and the development of a competent public opinion on the subject, will be helpful. By way of a personal contribution to the discussion I will repeat a suggestion which I have previously made elsewhere – that the use of single-engine machines in public transport should be allowed on condition that their gross weight does not exceed 10,000 lb; that the stalling speed is not over 55 mph; and that the demonstrated distance to complete a landing from a 50-ft altitude is not over 1500 ft. I have no doubt that a safety record could now be made in the operation of single-engine aircraft in regular transport under such conditions which would be better than the safety record actually existing on the airlines of the United States at any time prior to 1938. The chance of death or injury to any individual passenger, even in many years of habitual use of the service, would be very small indeed; but, at the same time, one could hardly hope to approach the prospective standard of safety of multiengine operation.

Short of the extreme of using single-engine aircraft, there is a possibility of relaxing the standards of single-engine performance for twin-engine machines in use on local runs with light traffic. As previously suggested here for the case of cargo service, the assumptions regarding instantaneous engine failures during take-off might be made less severe than those which apply to the major transport operations. Such relaxation would find additional justification in that these local services will especially serve small communities, and the airports used in such communities are likely to be bordered by open fields, favorable to the prospects of a forced landing in case of need, rather than by thickly built-up industrial or residential areas.

MOTION ANALYSIS by Means of HIGH-SPEED PHOTOGRAPHY

by H. D. JACKES
Wright Aeronautical Corp.

IN analyzing the operation of aircraft-engine parts, the high-speed motion picture camera has many advantages over the older methods, which involved the use of vibration indicators, stroboscopes, and strain gages.

High-speed motion picture studies have the advantages of forming a permanent record and of accurately evaluating transient conditions.

Many subjects have been studied by means of high-speed motion pictures, some of which are here shown and discussed by Mr. Jackes.

■ ■ ■

THE AUTHOR: H. D. JACKES (J '40) who, as project engineer of Wright Aeronautical Corp., specializes in gearing and clutches, has been with that company since 1938, a year after receiving his B. S. degree in mechanical engineering from the University of Kentucky. At the time the high speed motion picture work reported in his paper was completed, he was assistant project engineer assigned to development of cylinder assembly and component parts.

FOR a number of years, the Wright Aeronautical Corp. has employed the stroboscope as a means of studying and analyzing parts of aircraft engines during operation. We have found this device to be of considerable value in discovering and analyzing vibrations and similar recurrent phenomena whose presence would only be surmised in the absence of some stroboscopic device.

However, the stroboscope has two serious limitations: (1) The record is no more permanent than the memory of the person or persons making the observation; and (2) transient conditions are exceedingly difficult to discover and analyze with stroboscopic means.

More recently, the high-speed motion picture camera has been utilized for time-motion studies, as a continuation of, rather than a replacement of, stroboscopic investigation. High-speed motion picture studies have the advantages of forming a permanent record and accurately evaluating transient conditions.

Single- or multiple-flash still pictures are also used to "freeze" parts moving at any velocity and are a valuable asset to any time-motion study.

The problem of determination of the actual motion of a rapidly moving part of an engineering structure has long been hampered by lack of adequate means of measuring the motion of the part. Strain gages and similar devices can be used to great advantage in many cases but it can readily be seen that the use of strain gages in many places is limited by the inability to provide proper attachment, impracticability of the electrical connections and wiring, the physical mass of the strain gage itself interfering with the proper motion of the part (in the case of valve spring surge), and the difficulty of analysis of such records when nothing is known about the motion of the subject part and related parts of the mechanism. The stroboscope steps in at this point, providing a visual means of analyzing recurrent motion and vibrations. The stroboscope is limited in that the part or structure investigated must be located in a manner to provide ready visual accessibility, and there is no record of the motion other than that impressed upon the observers. In addition, transient conditions usually go unnoticed.

High-speed motion pictures and instantaneous flash photography are somewhat subject to the accessibility limitations of the stroboscope but provide a permanent record which can be studied at leisure, and accurate measurements and curves showing motion with respect to time can be made as desired. Often the conditions of extreme personal discomfort accompanying stroboscopic investigations seriously limit the quality of the observation. In addition, it is often possible to locate the camera where it would be

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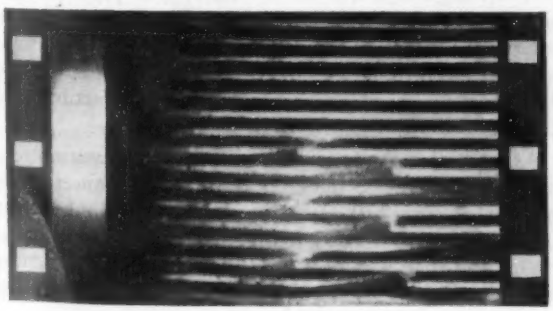
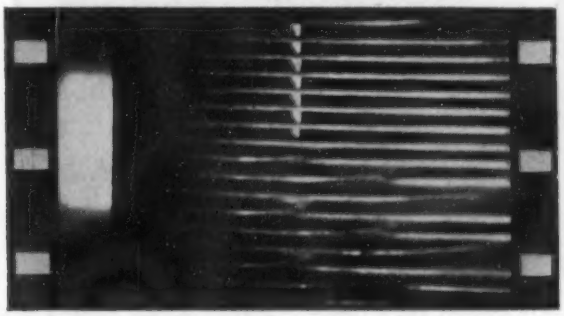
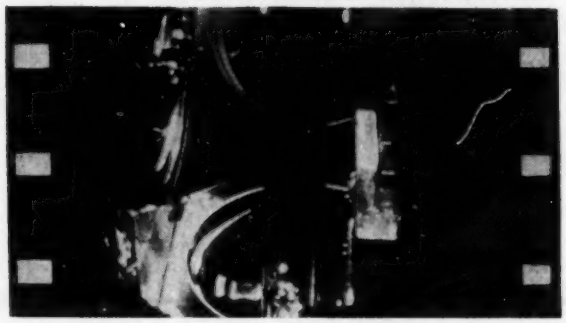




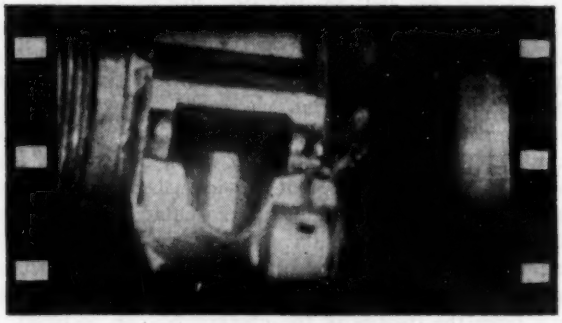
■ Figs. 1 (left) and 2 (right) – Section of C14BA engine cylinder head operating on single-cylinder engine, showing fin vibration without (left) and with (right), vibration damper

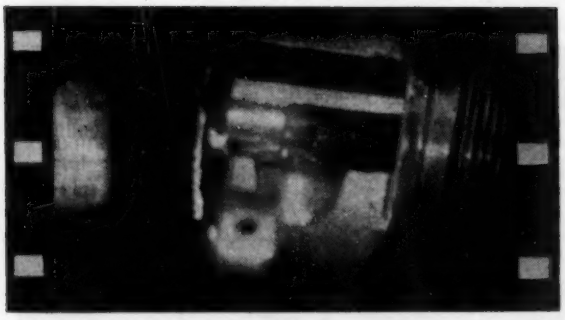
either difficult, impossible, or dangerous for a person to be, and have the camera operated by remote control.

We have been using various types of strain gages, seismic types of vibration indicators, and the stroboscope for many years in measuring and analyzing various vibration and motion problems. In approaching certain problems, it has been found that the above methods are far from adequate. We have found high-speed motion picture analysis particularly useful for analysis of oil flow, cylinder-head-fin vibration, effect of entrained air on oil pump performance, valve mechanism performance, or similar problems where the difficulty of attaching strain gages or the presence of transient conditions makes analysis by other means difficult or impractical. In particular, it should be pointed out that a camera is basically a rugged piece of equipment, which we have found to be capable of withstanding severe vibration, buffeting by exhaust gases and similar conditions, and still able to provide a faithful record of what has transpired.



■ Figs. 3 (above) and 4 (below) – Crankcase scavenging of C14 engines at shutdown





■ Figs. 5 (above) and 6 (below) – Solenoid plunger entering locking slot of E-4 type machine gun synchronizer cam follower plunger – above: sluggish action; below: redesigned to give more rapid and positive action

Mere observation of the film records, projected at normal projection speed resulting in a 150:times magnification, shows many motions to be far different than previously conceived, and will show what has transpired without need of a long and tedious education in the art of reading and analyzing oscillograph records, as is the case with records obtained from strain gages and such. In addition, it is very easy to duplicate tests with high-speed motion pictures, whereas accuracy of duplication of tests by strain gage methods depends upon the accuracy of location of the gages, temperature conditions, and so forth. If a more thorough analysis of the photographic records is desired, a time-motion study of any part is readily prepared with simple equipment usually available in any general-purpose photographic darkroom. Due to the simplicity of analysis of photographic records, and the ease of presentation of the pictures themselves, we have found that an analysis of this type is very easy to "put across" to engineering executives, who do not have the time to study and analyze the sometimes complicated results of other methods of analysis and are more ready to believe the motion picture display. This confirms the saying, "One picture is worth ten thousand words."

We have made an intensive study of many subjects by means of high-speed motion pictures. Some representative pictures are shown herewith, as follows:

1. For some time, we have been troubled with breakage of cylinder-head cooling fins on our higher horsepower engines. We at first thought that the vibration was due to airflow. We attempted to analyze this fin vibration with the stroboscope with negligible success. The first filming of head-fin vibration showed that the motion was a transient condition, excited by deformation of the cylinder head under the gas forces present during combustion. The maximum vibration was definitely confined to a few small areas of the head. This information greatly simplified the test of redesign to cure the trouble. (Figs. 1 and 2.)[1]

2. A problem familiar to manufacturers of radial and inverted engines is that of "hydraulic lock" or drainage of oil past the piston rings of the lower cylinders when the engine is shut down, resulting in the lower cylinders filling with oil. When the engine is cranked, the piston strikes these pockets of oil, locking the engine, sometimes with damage to the connecting rods.

As part of a study of this difficulty, we assembled an engine with one cylinder, piston, and connecting rod missing. A cover was placed over the open cylinder bolting flange. The engine was motored over sufficiently to heat the oil and then was shut down. At the instant of cutting the power, the cover was removed, and the high-speed

[1] Figs. 1-8, 10-12, 14, 16, 17 each show only one frame taken from a roll of high-speed motion picture film. Due to the high speed at which the pictures have been taken, each picture practically "freezes" the motion of the parts.

camera started in order to record where the oil goes when the engine shuts down. This was done on several engine models and demonstrated wide differences in scavenging largely attributed to different crankcase designs. (Figs. 3 and 4.)

3. A subject of much discussion has been the functioning of various types of machine gun synchronizer solenoid controls. During the process of determining just what the action of each type of control is, we were involved inasmuch as our engines utilized these controls. A gun synchronizer housing was cut away sufficiently to show the working parts and the action of the control filmed by means of the high-speed camera. These pictures showed a distinct difference in operation between the two principal types under discussion. That is, the action of one type was rapid and positive, whereas the other was sluggish and not at all positive in action. (Figs. 5 and 6.)

4. In our engine development work, we employ single-cylinder test engines consisting of a standard engine with all cylinders and connecting rods but one removed. These sometimes give trouble because of the large torque variation inherent in single-cylinder engines. A flywheel incorporating dynamic vibration dampers of the type used in Wright engines was constructed. As a part of the development of this dynamic damper flywheel, pictures were taken to show its operation. (Fig. 7.)

5. In connection with oil pump development for high-altitude engines, a transparent oil pump was constructed for a high-speed motion picture study of the effect of entrained air on the oil pump. Multiple-flash still pictures were also employed to "freeze" the motion. These pictures show clearly the path of the air bubbles, how they recirculate, the effect of clearance volume on recirculation of the air, and other things of interest. (Figs. 8 and 9.)

6. One project particularly aided by the high-speed camera was the development of a valve gear to stand increased engine speeds. This is an excellent example of a place where all other methods of determination fall far short for reasons previously mentioned. As we feel that this is one of the applications where the high-speed camera was most useful, a short history of our valve gear development is in order:

In the process of development of the Cyclone series engines, the take-off speed has increased from 1850 to 2800 rpm. In spite of this increase in speed and a corresponding increase in power, the basic engine dimensions have been little changed. This has been particularly true of the valve gear. The valves, push rods, rocker arms, and tappets currently used are heavier if anything, the valve timing is essentially the same, and the same amount of space is provided for valve springs.

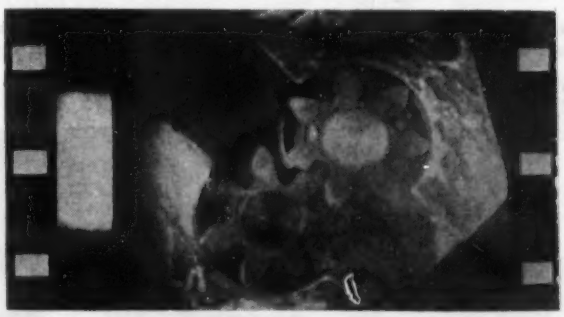
Calculations indicated a valve jumping speed of 3250 rpm. As engine speeds exceeded 2600 rpm, for continuous operation, difficulties were experienced which could only be explained by faulty valve dynamics and valve jumping.

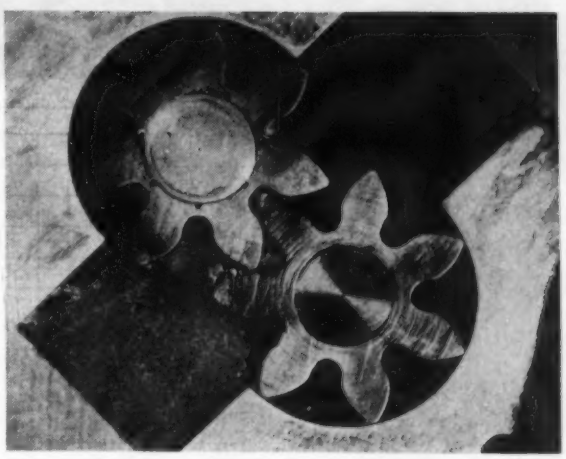
Attempts to analyze valve gear operation by means of strain gages were almost useless because of difficulty in applying the gages to vital points in the system and the difficulty of interpreting the results. Stroboscopic investigation was more successful but was not completely satisfactory due to the difficulty of establishing permanent records, the amount of punishment an observer could take, and the fact (unsuspected at the time) that there are many transient conditions of valve malfunctioning during otherwise normal operation.

A more complete understanding of the problem was reached with almost the first filming of the valve gear with the high-speed camera. The camera not only formed a permanent record which could be studied at leisure, but also allowed operation of the engine at conditions impossible to maintain for the period of time necessary to make a stroboscopic investigation. It only requires slightly more than one second for 50 ft of film to travel through the camera. In addition, the camera could "see" many things not possible visually because of oil fog inside of the engine. We found it relatively easy to film the action of the cam and rollers inside of the nose section.

Information gathered by use of the stroboscope indicated that valve-spring surge was the chief cause of valve gear malfunctioning. A study of the high-speed motion pictures

■ Fig. 7 – Dynamic damper flywheel operating on single-cylinder test engine

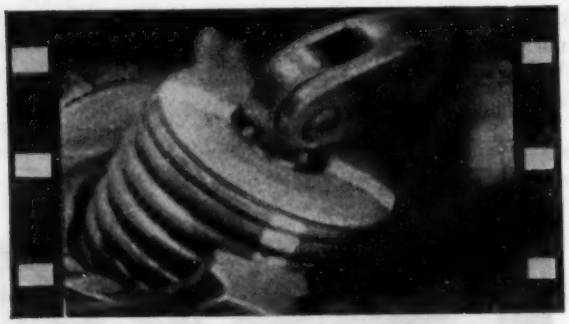


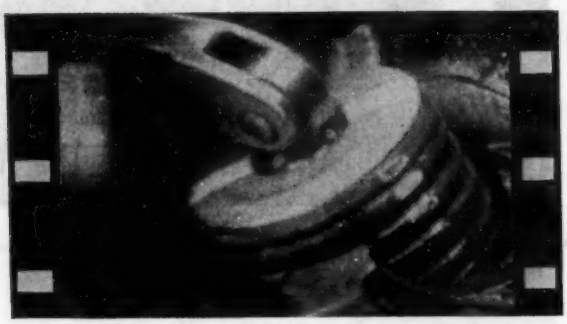


■ Figs. 8 (above) and 9 (below) – Transparent oil pump (test rig) operating with entrained air in oil – above: movie frame; below: multiple-flash still picture

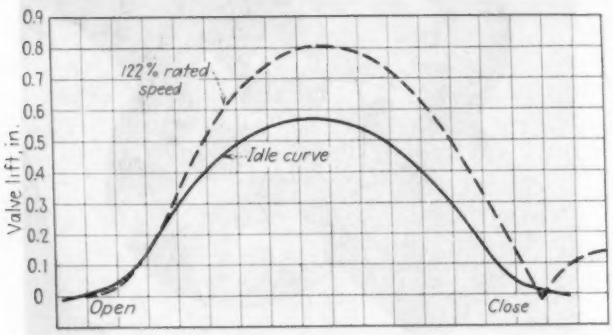
and preparation of time-motion studies showed that spring surge, while a factor, was a secondary contributor towards poor valve dynamics. Flexibility of the valve gear, which was previously thought more than sufficiently rigid, was found one of the principal causes of malfunctioning.

Increasing the rigidity of the entire linkage of the valve gear, plus a redesign of the valve springs to reduce spring surge, enabled the engine to operate at well over 3000 rpm without valve jumping, whereas traces of intermittent valve jumping had been found as low as 2500 rpm on the original valve gear.





■ Figs. 10 (above) and 11 (below) – Cutaway C9 intake rocker box and valve parts on single-cylinder test engine – above: cruising conditions; below: 122% overspeed operation



■ Figs. 12 (above) and 13 (below) – Cam and roller action through cutaway nose section (above) and time-motion study for intake valve – both at 122% overspeed

Normal functioning of the valve gear is shown in Fig. 10. This picture shows the action of an intake valve operating in a single-cylinder engine at normal maximum cruising conditions. There is some spring surge present but the magnitude of the surge is not great, the valve action is smooth and continuous, and the seating rather gentle. It is felt that this valve action is satisfactory.

Unsatisfactory valve action is illustrated in the accompanying pictures. Fig. 11 shows the intake valve of a single-cylinder engine operating at considerable overspeed, using the original valve gear design. In this case, there is severe valve spring surge, the valve jumps to the extent of compressing the valve spring to its solid height, and the valve strikes the seat with violence, rebounding to an opening of more than 0.090 in. Of particular interest is the fact that the valve does not behave the same in successive cycles, sometimes seating comparatively gently and then with great violence.

Fig. 12 shows the cam and roller operation of this engine at the same speed. The intake tappet and roller are to the right. The motion pictures of this operation showed the magnitude of the valve jumping and the violence with which the roller strikes the trailing edge of the cam. A time-motion study at this speed is shown in Fig. 13.

Fig. 14 shows the redesigned valve gear operating at 300 rpm higher speed than shown previously. There is a reduction in spring surge and general smoothness of operation. We feel certain that a development of this nature would have taken many times as long in the absence of a tool such as the high-speed camera, and we would not have been sure of what we had. A time-motion study for the redesigned valve gear is shown in Fig. 15.

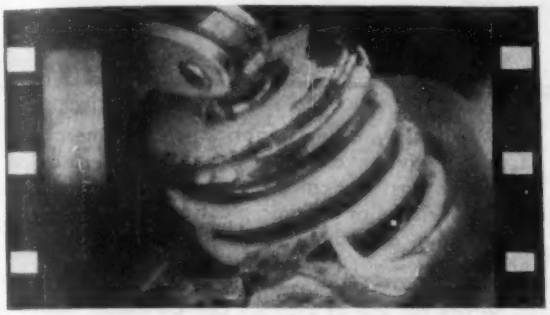
The preliminary valve gear work was carried out on the single-cylinder engines, so to prove our work, we checked it out on a full-scale engine of each model on a dynamometer test stand. Fig. 16 shows the redesigned valve gear operating at rated power in a full-scale C14 engine. We found no essential difference between valve gear operation on single-cylinder and corresponding full-scale engine.

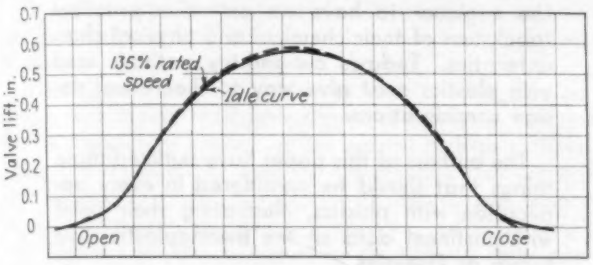
As an excellent example of the amount of abuse a camera can take, Fig. 17 shows exhaust valve operation taken through the exhaust port of a single cylinder while engine operating at take-off power. The camera was in almost the direct blast of the exhaust of this cylinder.

Fig. 18 shows the high-speed camera set up to take pictures of the valve gear on a single-cylinder engine, with Walter Calvert, our photographer, at the controls of the camera. As previously stated, 50 ft of film travel through the camera in slightly more than one second, and the operator need only be at the camera during filming.

From the results of high-speed motion picture investigations, it is believed that investigations by means of high-speed motion pictures are valuable from any point of view. The actual filming is not particularly difficult, although requiring a photographer having a great deal of experience in high-speed work, the results are accurate, analyses are readily made by anyone having general engineering experience, and the analyses are easily checked by projecting the films themselves and reviewing the motion and comparing the motion with the analysis.

There are many possible fields of application of the high-speed motion picture camera. Some of the subjects that have been filmed are undoubtedly familiar and others possibly not so well known. The high-speed camera is used to photograph horse races and other speed events, providing the well known "photo finish." It has also been used to photograph windshield breakage, where birds are shot at windshields by specially constructed guns.





■ Figs. 14 (above) and 15 (below) – Redesigned valve gear operating at overspeed – above: same view as Fig. 10; below: time-motion study

■ Fig. 16 – Similar to Fig. 10 except redesigned valve gear on full-scale engine at rated power and speed

■ Fig. 17 – Exhaust valve operation at take-off, taken through exhaust port

Some more of the uses of this relatively new tool are:

1. Analysis of the action of the escapement mechanism of precision clocks.

2. Investigation of the action of the shutter of high-speed aerial cameras. The shutters are in proportion to the size of the lens systems, so there is a considerable problem in accelerating and stopping such a large shutter.

3. Studies of various manufacturing processes such as the blowing of spun glass, or the action of the various parts of a high-speed weaving machine.

It is proposed that there are many uses for high-speed photography in the aircraft industry. Some suggested uses are as follows:

1. A study of the flutter of wing and control surfaces. It would appear that this is an ideal problem to be approached by means of high-speed photography.

2. A study of the action of wheel struts and landing mechanisms. This would appear to one not well acquainted with the problem to be a situation concerning a violent impact, as when first striking the ground, followed by lesser impacts due to bumps and wheel-brake applications.

3. Exhaust collector ring vibration and failures. The camera could be located in the engine compartment, during flight, to record vibrations and other conditions conducive to exhaust system failures.

In conclusion, it is proposed that high-speed photography, and particularly photography by means of the high-speed motion picture camera, is a valuable tool to research and development engineers. The uses of this tool are growing continually and it is proving a valuable addition to a considerable list of equipment designed to measure the motions and vibrations of parts of engineering structures.

The high-speed motion picture camera, due to its flexibility, ruggedness, and ease of analysis of the films is, in many respects, superior to other methods of determination of motion of parts. It has certain limitations, principally of accessibility and illumination, and occasionally is not fast enough for the evaluation of certain vibrations of ultra high frequency but, in general, is certainly a capable piece of equipment.

This discussion was made possible by the generous efforts of many members of the organization of which the author is a part, and I wish to thank them for their co-operation. Walter S. Calvert, an independent photographer, was of particular assistance with his excellent photography.

■ Fig. 18 – Camera ready to photograph valve gear

CHARACTERISTICS of PLASTICS as

THE term "engineering materials" usually denotes materials that are suitable for use in the construction of stress-bearing structures. This definition can be further subdivided into

1. Materials used for carrying primary stresses.
2. Materials used for carrying secondary stresses.
3. Materials that need only be self-supporting.

The general use of plastics has been largely limited to engineering applications of the latter two types. Part of this limitation has been due to the basic nature of plastic materials and part has been due to a lack of detailed knowledge of their fundamental physical characteristics.

We must admit that the organic plastic materials that we have available to date are limited in their engineering applications. At the same time, we must concede that they have certain characteristics which make them attractive to both the designer and the engineer. The most outstanding of these characteristics are:

1. Low density.
2. Ease of fabrication.
3. Low thermal conductivity.
4. Available transparency.
5. Low unit cost for mass-production items.

Some of these characteristics are of very particular interest in military applications but, in general, they are also of importance in every application. For example, low density or, in other words, low weight per unit volume, is obviously important wherever weight saving is a prime consideration but it is also important in applications where rotational or vibrational inertia must be reduced. The relative importance of the ease with which complex structures can be fabricated from plastics has been very well brought out in some of the work in which we are all now so vitally interested. One-piece bomber noses, one-piece gun turrets, intricate manifold forms, and complex forming dies are now being quite successfully made from plastics at much lower labor costs than they could be made from other materials. The relative ease with which quantity production of small injection and compression molded pieces of plastics can be accomplished is well known. The relatively low thermal conductivity of plastics is useful in numerous applications which involve both the amount of heat transferred and the rate at which heat is transferred. Available transparency combined with reasonable mechanical strength characteristics is probably one of the most important characteristics of plastics in their present military applications. It appears probable that trends of future design in the housing, home furnishing, and automotive industries will continue to make extensive use of this characteristic. Low unit cost of fabrication of small items produced in large quantities is a feature of organic plastics which indicates their use in many wartime and peacetime applications.

No matter how obvious these features may appear to be, it is also obvious that they cannot be correctly and profitably applied in a truly planned manner unless the designer and the engineer have a sufficient familiarity with their general nature or have available an extensive practical tabulation of their chemical and physical characteristics. Most of the engineering advances which have been made in the use of plastics to date have been made through the cut-and-try method rather than through that of theoretical design consideration. This picture is changing somewhat, however, as more data on the various materials are made available in terms of both the common engineering data form as well as that of observations of practical service applications. At the same time, more engineers and designers are becoming familiar with the materials by actually

[This paper was presented at the SAE Meeting on Materials – War and Post-War, Cleveland, Ohio, Nov. 10, 1943.]

LOW density, ease of fabrication, low thermal conductivity, available transparency, and low unit cost for mass-production items are characteristics of organic plastic materials that make them attractive to the designer and the engineer.

Before full use of plastics can be made, however, it is necessary for both the designer and the engineer to have an extensive practical tabulation of their chemical and physical characteristics. Today's cut-and-try methods used with plastics must give way to theoretical design considerations.

The authors of this paper have outlined those things that should be considered in every application with plastics, illustrating their point with pertinent data on the thermoplastic resin known as Plexiglas.

★ ★ ★

THE AUTHORS: W. F. BARTOE, who has been director of Rohm & Haas Co.'s physics laboratory since 1936, is a recognized authority on the physical properties and testing of resinous and acrylic plastics. After having received his M.S. degree from Ohio State University in 1933, he was employed as a physicist by Firestone Tire & Rubber Co. DR. D. S. FREDERICK, first winner of the John Wesley Hyatt award in 1942 for outstanding contributions to the plastics industry, is vice-president and sales manager of Rohm & Haas Co. Joining that organization as a research worker in 1934, he is responsible for introducing Plexiglas to many aircraft companies and to the Army and Navy. A graduate of Miami University, Dr. Frederick received his Ph.D. from the University of Illinois.

as ENGINEERING MATERIALS

by W. F. BARTOE and DR. D. S. FREDERICK
Rohm and Haas Co.

working with them. It appears that it is desirable, however, that the extent of this familiarity should be increased if we are to make the most profitable use of these materials in a truly engineering manner. We would, therefore, like to outline those things which should be considered in every engineering application with plastics. Later, in the course of outlining these considerations, some detailed remarks and data will be presented to illustrate certain points. These data will be taken from the rather extensive information that we have available on methyl methacrylate resin because that is the material with which the authors are most familiar and because such data serve quite well to illustrate points that they wish to make.

The engineer finds it possible to apply tabular data on metals with a very reasonable degree of success in designing its applications because he has learned perhaps without being too conscious of how he has learned, that certain data applied to certain formulas with appropriate factors of indeterminancy, commonly known as "safety factors," will ensure that the structure in the designs will give reasonable service life. This can also be done with any kind of plastic material. Unfortunately, there are several things that must be taken into account which do not enter into the ordinary engineering design formula. The plastics industry itself is partly guilty for having permitted such a condition to arise by publishing and permitting to be published, data on mechanical characteristics of plastics in the same form in which such data are presented for metals and other structural materials, without sufficient explanation of the significance of such tabulations. Fortunately, this situation is being corrected by the extensive improvement of available data on plastics. Further help in correcting it will be supplied by our engineering schools as soon as they give recognition to the fact that the use of plastics is rapidly expanding and, therefore, worthy of a certain share of attention in any engineering educational program.

■ General Considerations

The most important general considerations in the designing of applications of any of the plastic materials that are not covered by ordinary tables of data on their properties are:

1. The effect of the nature and mode of application of stress.
2. The amount of permanent dimensional change with time.
3. The required service temperature range.
4. The previous history of the material.
5. The effect of temperature on their physical properties.

Although we quite commonly speak of thermoplastic and thermosetting compounds as being the two main classifications of plastic materials, both of these types of materials are subject to the same considerations in engineering design. They differ principally in the degree to which they are subject to the effect of a particular variable. All of these materials are subject to plastic flow under any applied stress. The amount of flow is, of course, dependent upon the nature of the material but is also dependent upon the total time of application of the load as well as the temperature at which this load is applied. In general, the thermoplastic materials are more subject to flow than the thermosetting materials but it should be remembered that this is not necessarily or always true.

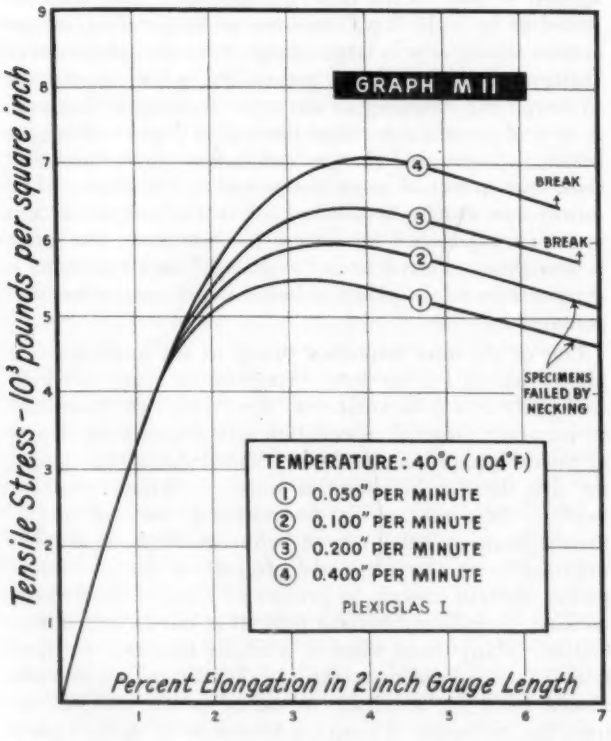
The effect of the mode of application of stress is often quite significant and does vary appreciably among the various types of plastic materials. Stresses applied quickly and removed quickly in the manner of those accompanying vibrations or forced flexing usually have less permanent deleterious effect than much smaller stresses applied for long periods of time. This is almost exactly contrary to that observed for those types of metals which are said to "fatigue" easily. All plastic materials are subject to what we would choose to call "service temperature flow." This is merely a question of an inherent tendency in plastic materials to relax and thereby distort under steadily applied stresses. This flow characteristic, sometimes referred to as "cold flow," becomes an engineering consideration whenever it is large enough to involve dimensional changes in excess of those permissible in the structure or to permit the loosening of any type of clamped fastening. It is well to remember that there is a distinct difference between dimensional changes due to flow under stress and those due to loss of some component of the plastic. The former can usually be allowed for in the original design or later compensated for after it has occurred. The latter is sometimes referred to as "shrinkage" and represents a deterioration of the plastic material which cannot be compensated for.

One of the most important things in any application of plastic materials is the service temperature range to which this material will be subjected. Knowledge of the service temperature range is essential in any engineering design of plastic application. Without a knowledge of this range, the first decision of choice of material; namely, that between a heat-resistant or thermosetting material and a thermoplastic material, cannot be made. Without detailed information on this range, the prospects of success of any plastic material cannot be predicted. Despite these facts, we have often found it most difficult to obtain such information. Many times when it is finally obtained, we find it to be an all-encompassing specification which is both absurd from the standpoint of actual practice as well as from the standpoint of practical attainment with the plastic materials that are known to date. Too much cannot be said at the present time about the need for detailed knowl-

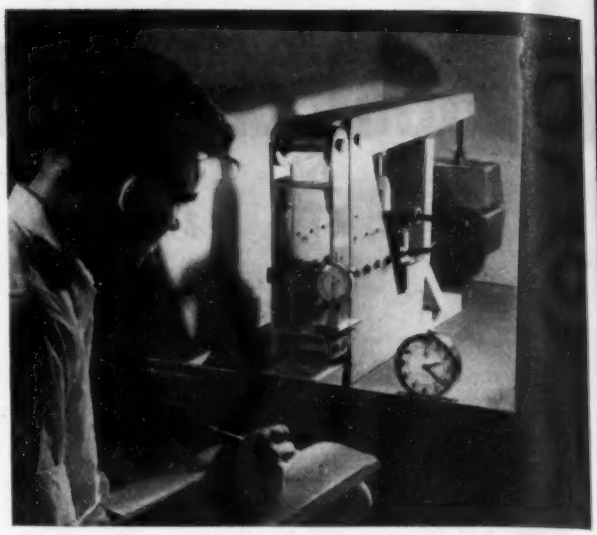
edge and data on the actual temperatures that must be sustained in any proposed application of plastics. Often, the extra 5- or 10-deg range that has been added on to the specified requirement with the idea that it will add an additional factor of safety means the difference between obtaining an otherwise desired type of material for that application and finding this material declared unsuitable. Likewise, failure to state completely the actual service temperature range may mean the difference between a successful application of a given type of plastic and a complete, if not disastrous, failure of that type. We believe that a full and well-considered recognition of the temperature limitations of plastic materials can result in both proper application and better application through engineering design considerations.

The effect of the previous history of the plastic material also enters into the consideration of engineering design. Sometimes this factor is referred to as preconditioning. It is obviously important where this conditioning is done by contact of things other than normal atmospheric variations. The importance of normal atmospheric variations, however, is often vastly underestimated. A plastic material which will function quite satisfactorily under certain applied stresses in a low-humidity climate, may fail quite miserably in a high-humidity climate.

We will leave to the last a discussion of the effect of weathering on the apparent physical characteristics of plastic materials because such are rather widely recognized even though they are not too well investigated in many cases. A number of very serious attempts have been made by those organizations interested in standardizing test methods and specifications for plastic materials to formulate some programs of test which will give a practical



■ Fig. 1 – Tensile stress-strain versus rate of straining



■ Fig. 2 – Cold-flow apparatus

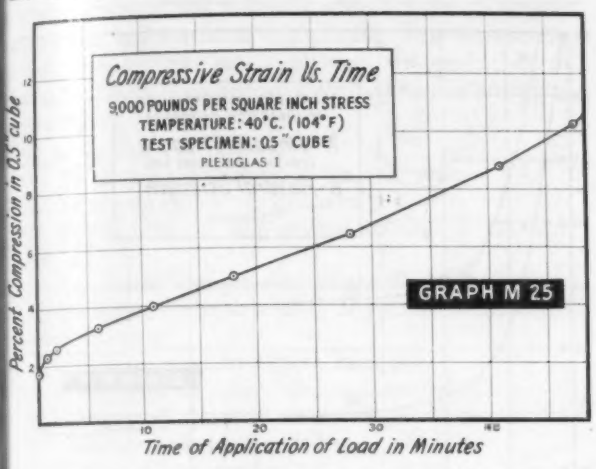
evaluation of the effects of any type of climatic condition on these materials. These programs are progressing quite well but they have not attained satisfactory stages of completion due, in part, to the marked change in status of our global and atmospheric interests. Until the time that such can be established and correlated to actual service information, we shall have to continue to base engineering design on the shorttime effect of sustained temperature and humidity on the physical characteristics of plastic materials.

■ Practical Data on a Thermoplastic Resin

Our discussion to this point has been very broadly generalized to include all plastic materials. Obviously, it cannot be continued indefinitely on this basis to any useful end. Perhaps a citation of some specific data will help to drive home the importance of some of these general considerations. The data that we wish to present have been obtained on one of the most widely known thermoplastic resins, namely, Plexiglas. The characteristics illustrated by these data are not necessarily the same as those that would be observed in data for other thermoplastic materials but they are typical enough to provide us with some definite points of illustration.

Stress-strain data are the usual starting point of any calculation of the application of engineering materials. The fact that such data are usually obtained in a more or less arbitrary manner is often not considered. A great deal of thought and work have been expended in attempts to determine the best methods of obtaining such data but, in general, those that have always been used for metals are still used for plastic materials. This means that the very important factor of extent of flow of plastic materials in comparison with metals is not always illustrated by such data.

The tensile stress-strain relationship versus rate of straining (Fig. 1) shows that within the limits of rate of straining commonly used in testing plastic materials, the tensile strength of a methyl methacrylate base resin will apparently vary from 5500 to 7000 psi. At the same time, up to those stresses and elongations over which a practically


Compressive Strain Vs. Time
9,000 POUNDS PER SQUARE INCH STRESS
TEMPERATURE: 40°C. (104°F)
TEST SPECIMEN: 0.5" CUBE
PLEXIGLAS I
GRAPH M 25
Percent Compression in 0.5" cube
Time of Application of Load in Minutes

■ Fig. 3 – Compressive strain versus time

straight-line relationship exists, the material does not display any difference with difference in rate of strain. This action may be explained on the basis of a practical yield-point for the type of material being tested. In other words, up to a certain stress value, very little of the strain is absorbed in plastic flow. Above this value, pseudoplastic flow begins to take place at a rate determined primarily by the rate of straining of the specimen and at stresses that are inversely proportional to the rate of straining. It is, therefore, obvious that the stress that any plastic part can withstand without becoming permanently deformed will not depend so much on the rate of straining as will the stress at which complete failure takes place. The ratio of the latter stress value commonly called "tensile strength,"[1] to the former, commonly called "elastic limit,"[1] is usually much less for plastics than is that for metals. These considerations immediately argue for the use of a "dead-load," or "cold-flow," or "creep" type of test for the evaluation of plastic materials from the engineering standpoint.

For convenience, data of this type are usually obtained on compression test samples in some sort of test jig similar to the one illustrated in Fig. 2.

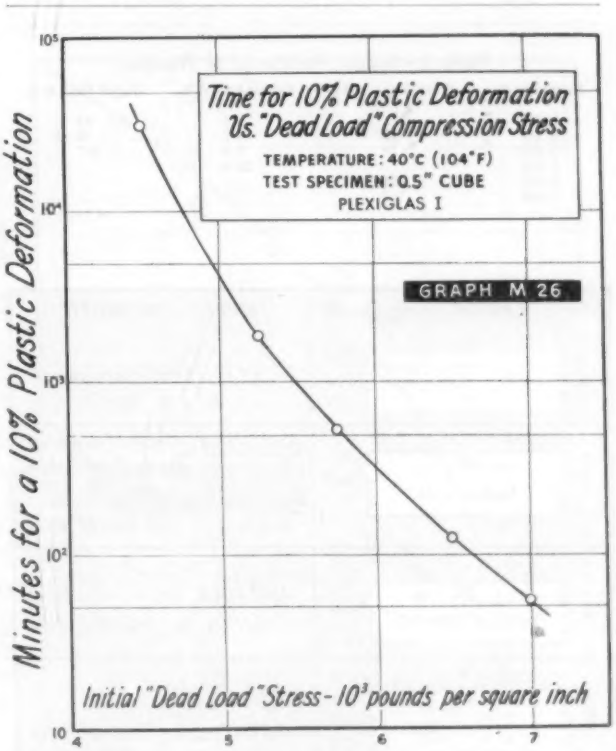
This unit[2] consists of a weighted lever which applies the "dead-load" to the test specimen through two parallel plates. The relative displacement of the two loading platens is conveniently measured by a dial gage.

Data obtained from this type of test may be plotted as a strain-time relationship, as shown in Fig. 3.

It is interesting to note that the instantaneous or elastic compression observed in this test is very approximately equal to the elongation for the straight-line position of the previously shown tensile stress-strain curves. The particular data shown are for a relatively high stress value and, therefore, the time axis is approximately 60 min in length. For a lower stress value, the time axis will be much longer but the general shape of the curve will be practically the same including the amount of instantaneous compression. It is, of course, possible to plot a whole family of curves showing the relationship between strain and time for various values in the initial stress; however, these soon become quite confusing and it is more useful to plot the initial stress against the logarithm of the time required for some arbitrary plastic (total minus first, or elastic, deformation). In most applications, a deformation of 10% would be the maximum permissible so this value is often used in plotting deformation versus stress relationships. However, any other value within the limits of accuracy of measurement of small deformations can be obtained from the original deformation versus time data.

We believe that the type of stress-strain data shown in Fig. 4 is of utmost importance in the engineering application of plastics. Unfortunately, it requires very considerable elapsed time to obtain this type of data. This is well illustrated by the fact that the lowest stress point on this graph required approximately 660 hr or 28 days total elapsed time for determination. Of course, the required time would be shorter for some materials subjected to the same applied stress but the actual determination of the limiting engineering stress value, as indicated by the stress asymptote to the deformation versus stress curve, would probably require the same order of magnitude of testing time. We believe it is worth it, however, and we would urge all design engineers to consider such values as necessary information in the proper design of plastic applications.

We recognize the natural desire of an engineer to design from a simplified table of data such as is commonly given for metals and too frequently given for plastics. We are glad to report, however, that this tendency is being overcome and we feel quite sure that at some time in the near future it will seem just as convenient to design from detailed data on the plastic flow of these in much the same


Time for 10% Plastic Deformation
Vs. "Dead Load" Compression Stress
TEMPERATURE: 40°C (104°F)
TEST SPECIMEN: 0.5" CUBE
PLEXIGLAS I
GRAPH M 26
Minutes for a 10% Plastic Deformation
Initial "Dead Load" Stress - 10³ pounds per square inch

■ Fig. 4 – Time for 10% plastic deformation versus "dead load" compression stress

[1] ASTM D638-42T Tentative Method of Test for Tensile Properties of Plastics.

[2] ASTM D621-41T Tentative Method of Test for Deformation of Plastics under Load at Elevated Temperatures.

manner as design is now being done by those engineers who are interested in the high-temperature applications of metals.

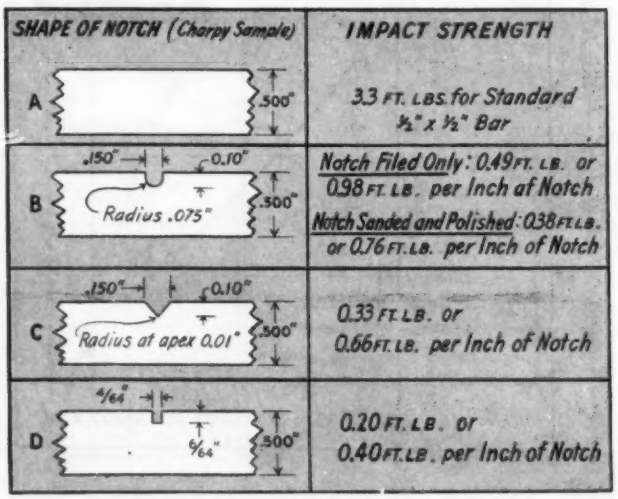
As a final indication of the importance of this method of approaching the subject of the engineering of plastics, we would like to call attention to the difference between the commonly quoted compressive-strength type of data and that obtained by a dead-load stress application method. At room temperature, the former type of data on a methacrylate plastic indicated an allowable stress of approximately 12,000 psi for a 10% deformation at the standard rate of straining of 0.05 in. per min, while the dead-load stress method shows that this same material at the same temperature will deform 10% in 7 days at a stress of 7000 psi.

The response of plastics to loading at very high rates is also of considerable consequence in the proper design of their application. The study of this response, usually called "impact strength," is now being given a considerable amount of attention by various plastics testing organizations. We have found it interesting to prepare a perfectly arbitrary comparison of different responses by comparing the bullet impact resistance, the falling (steel) ball impact resistance, and the falling shot-bag resistance of our material. (Table 1.)

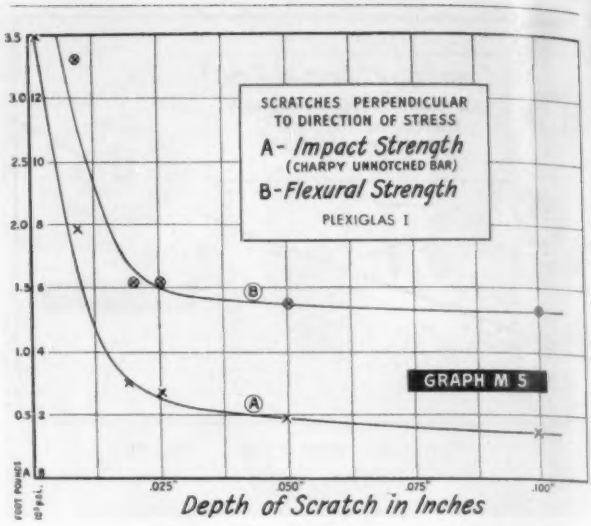
In some cases, for very high rates of impact, part of the energy is actually absorbed in the heating of the test sample. In other cases, the nature of the impacting body necessarily dictates a different type of distribution of stress. It is, therefore, obvious that a simple impact test cannot necessarily indicate true performance characteristics. It is also true that the gross shape and the minor details of the shape of a test specimen have an appreciable influence on the strength characteristics of that specimen. This can be well illustrated by the tabulation of impact strength versus the shape of notch in impact-test specimens shown in Fig. 5.

Table 1 – Impact Resistance of Plexiglas

Thickness, in.	Bullet Impact, ft lb	Falling Ball, ft lb	Bag of Shot, ft lb
0.060	..	1.5	33
0.125	13	3.7	78
0.250	24	9.5	217
0.375	..	23.0	..
0.500	55	....	..
1.000	168	....	..



■ Fig. 5 – Effect of shape of notch in test specimen on impact strength



■ Fig. 6 – Impact and flexural strengths versus depth of surface scratches

This factor is further illustrated by the effect of minor surface imperfections on both the impact and flexural strength of a standard test specimen, as shown in Fig. 6.

It is apparent from these data that relatively minor surface scratches can cause an appreciable decrease in both the flexural strength and the impact strength of plastic materials if they are perpendicular to the direction of applied tensile stress. Additional data illustrating the importance of this "notching-effect" are given in Table 2.

All of the data that have been presented and will be presented here are confined to the results of measurements on unfilled resins. The effect of the addition of fillers of various types is a complete topic in itself. Without any intent of indicating the extent of information that is avail-

Table 2 – Effect of Surface Condition on Impact Strength of Plexiglas

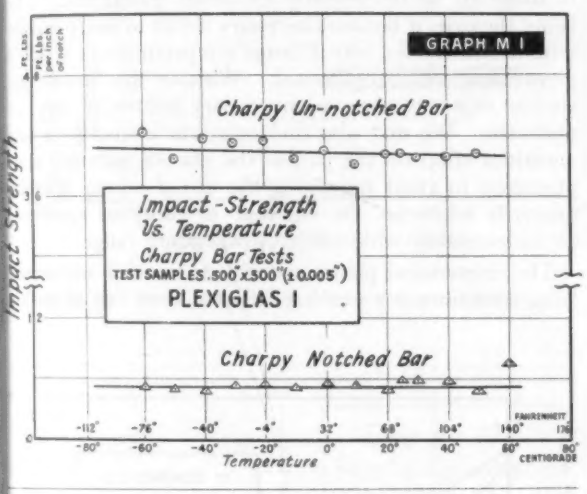
Condition of Tension Surface	Charpy Impact Strength, ft-lb	Maximum Flexural Strength, psi
Polished to remove scratches	3.2	13,750
Sawed	2.9	13,600
Passed through planer so that scratches are parallel to long dimension	3.4	13,400
Passed through planer so that scratches are perpendicular to long dimension	3.1	13,300
Sanded on belt sander so that scratches are parallel to long dimension	3.5	13,500
Sanded on belt sander so that scratches are at 45-deg. angle to long dimension	3.0	12,600
Sanded on belt sander so that scratches are perpendicular to long dimension	1.6	9,530
Tension side sanded so that scratches are parallel to long dimension, compression side sanded so that scratches are perpendicular to long dimension	3.2	13,000
Tension side sanded perpendicular, compression side sanded parallel to long dimension	1.6	9,300
Sanded on belt sander so that scratches were perpendicular to long dimension, then resanded so that scratches are parallel to long dimension	3.3	13,100
Sanded on belt sander so that scratches are perpendicular to long dimension, then coated with one heavy coat ethylene dichloride-Plexiglas solution	2.2	13,000
Same as condition immediately above, except two medium coats ethylene dichloride-Plexiglas solution	2.9	13,000

able on the nature and utility of various types of fillers – and without any intent of depreciating their importance in the processing and use of plastics – we would like to summarize briefly their general function from an engineering standpoint as follows:

Fillers are added to plastics,

1. To change the density of the base resin.
2. To improve their resistance to heat.
3. To improve the dimensional stability of the base resin.
4. To provide strengthening of the base resin.
5. To reduce the susceptibility of the base resin to the "notching-effect."
6. To obtain specialty effects, such as,
   a. Improved or altered electrical properties.
   b. Improved chemical resistance.
   c. Reduced frictional resistance.
   d. Altered heat conductivity.
   e. Altered appearance.

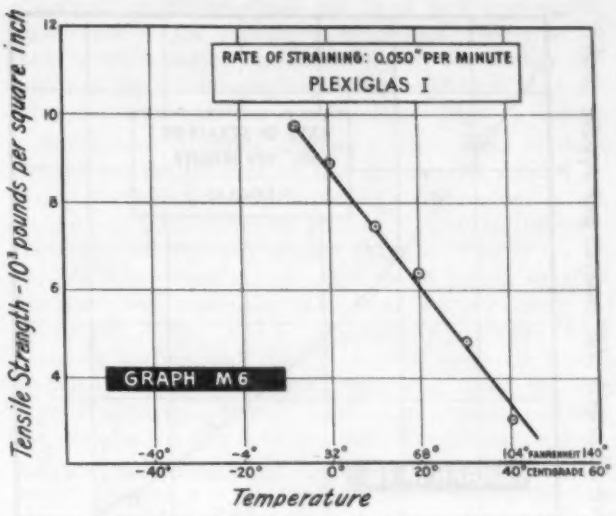
The importance of the effect of service temperature on the strength characteristics of plastic materials is amply illustrated in Figs. 7-12.



■ Fig. 7 – Impact strength versus temperature

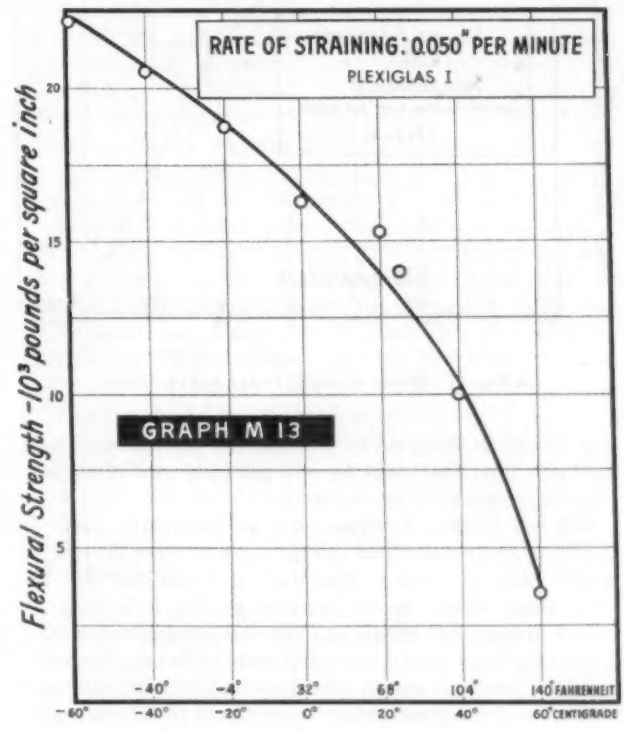
It is clear that the tensile, compressive, flexural, shear strengths, and modulus of elasticity of the material on which these data were obtained are very definitely affected by temperature. At the same time, the impact strength of this material remains practically independent of temperature. Although this apparent anomaly is not equally indicated for all plastic materials, it is generally important enough to introduce us to a discussion that has often confused the interpretation of the results of service experience with plastic materials. We refer particularly to that rather intangible characteristic commonly called "brittleness."

"Brittleness" has too often been judged on the basis of personal impression rather than on a truly objective experimental determination. This may be illustrated by describing the actions of a person judging the effect of temperature on a plastic material by bending a strip of the material at various temperatures. If he has chosen a strip that he cannot break, he will report that this material only becomes stiffer and does not become brittle with reduction

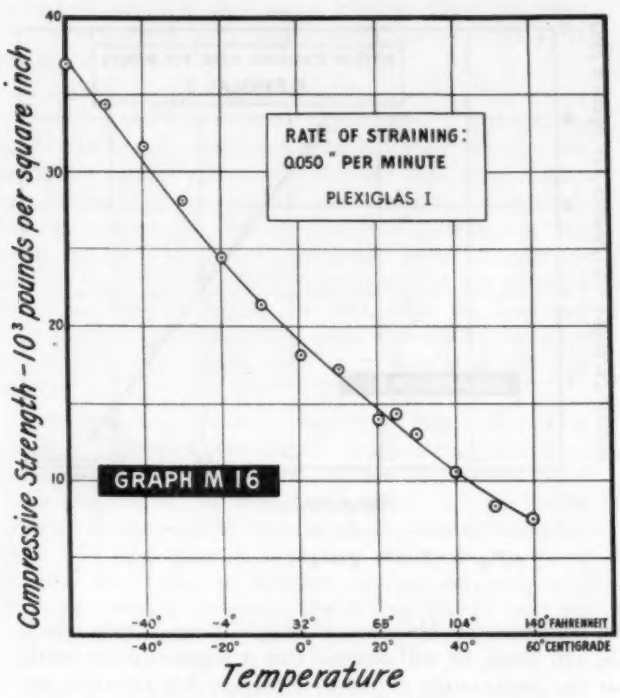


■ Fig. 8 – Tensile strength versus temperature

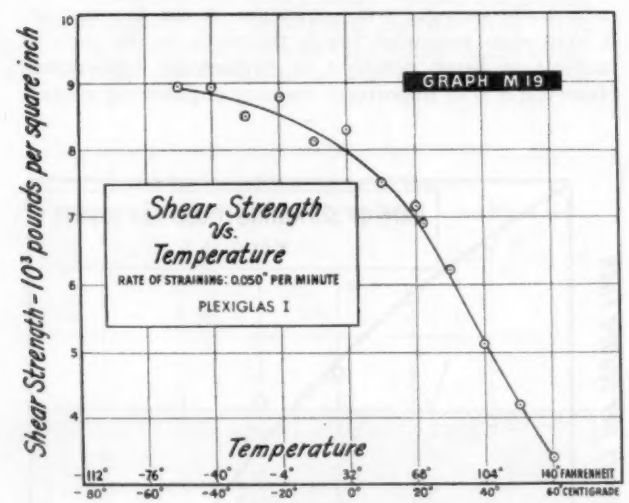
in temperature. However, if he has chosen a strip which he can break, he will contend that it becomes more brittle as the temperature is reduced because his principal observation is that of the distance through which he can bend the specimen without producing failure and because he is unable to integrate the deflection-stress relationship to obtain the observation that the material actually becomes structurally stronger with reduction in the temperature. It may seem somewhat beside the point to cite such an example as being pertinent to engineering applications. However, it is of importance and any engineering applica-



■ Fig. 9 – Flexural strength versus temperature


Compressive Strength - 10^3 pounds per square inch
RATE OF STRAINING:
0.050 " PER MINUTE
PLEXIGLAS I
GRAPH M 16
Temperature
-40° -4° 32° 68° 104° 140° FAHRENHEIT
-40° -20° 0° 20° 40° 60° CENTIGRADE

■ Fig. 10 – Compressive strength versus temperature


Shear Strength - 10^3 pounds per square inch
GRAPH M 19
Shear Strength
Vs.
Temperature
RATE OF STRAINING: 0.050" PER MINUTE
PLEXIGLAS I
Temperature
-112° -76° -40° -4° 32° 68° 104° 140° FAHRENHEIT
-80° -60° -40° -20° 0° 20° 40° 60° CENTIGRADE

■ Fig. 11 – Shear strength versus temperature

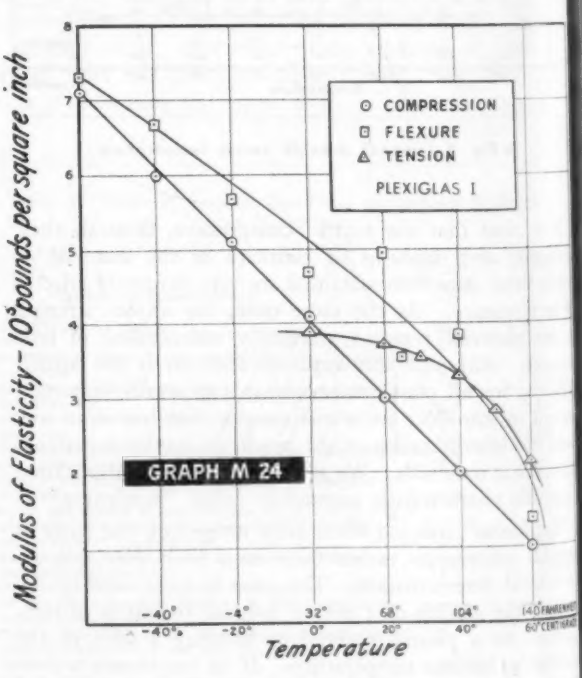
tion should be designed to preclude the possibility of such an observation ever entering into the judgment of the service characteristic of the material.

May we further illustrate with an imaginary case. It would be quite foolish to design a plastic door handle so flexible that, at room temperature, it would not fail because there would be no occasion to flex it beyond its failure strength but which, at a reduced temperature, would apparently break easily under the same deflection. It would be much better to design this door handle with sufficient stiffness at the highest service temperature to preclude giving anyone the impression that it could or should ever be flexed beyond a certain limit.

The term "toughness" is often used to mean the converse of "brittleness." At the same time, this term has been given a factual engineering interpretation as the area under the entire stress-to-failure versus strain curve. This interpretation is quite correct for any type of application wherein the permissible deformation is unlimited. However, for those applications that are engineered to limited deformations, the area under the total stress-strain curve loses importance and the area under the stress-strain curve up to the deformation limit becomes more significant. This, in turn, frequently reverses the relative "toughness" rating that is given to the various plastic materials.

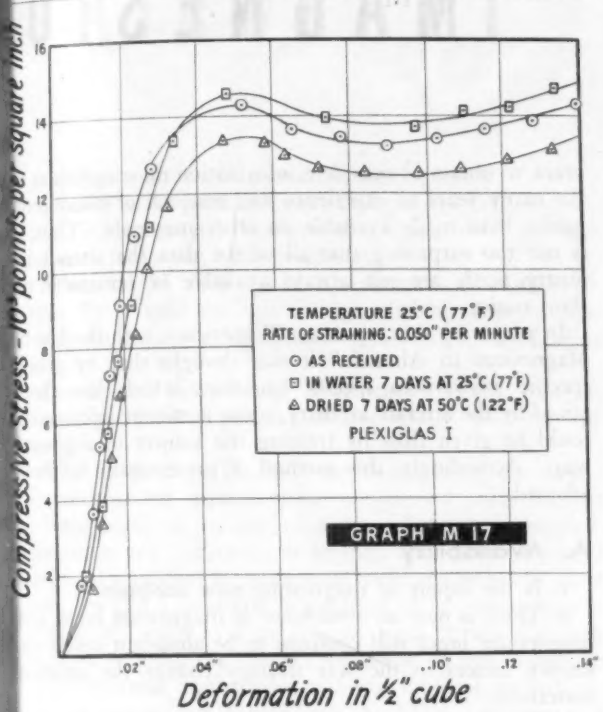
We have previously stressed the importance of having a precise knowledge of the service temperature range to which plastic materials are to be subjected in the engineering of applications of these materials. We all recognize that it would be desirable to have plastic materials which were no more sensitive to temperature effects than are metals. We undoubtedly will have such materials some day, possibly just as soon as the metallurgist has discovered how to obtain all of the strengths that he believes are latent in his materials. Until such a time, however, we shall have to be content with materials whose service temperature range barely encompasses that to which man is subjected in his recently expanded global life. This being the case, it becomes necessary for us to study and determine closely the actual range temperatures to which a given item will be subjected. Without this knowledge we can expect to observe unnecessary failures of their applications. We may also find ourselves limited by a conscientious effort on the part of the plastics materials manufacturers to avoid suggesting the use of certain desirable materials whenever the engineer is guilty of specifying an unnecessarily wide service temperature range.

The resistance of plastic materials to weather and weathering conditions is somewhat different from that of metals

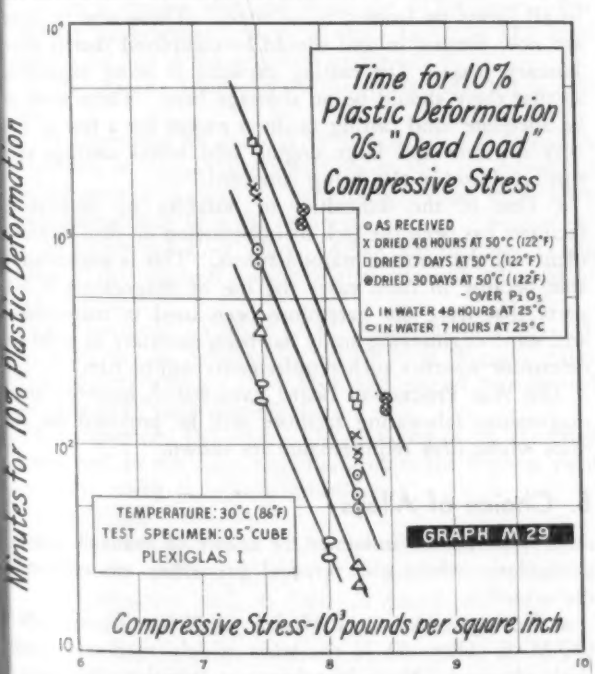


■ Fig. 12 – Modulus of elasticity versus temperature

n that their strength characteristics can become appreciably hanged under certain weathering conditions without the occurrence of any visible evidence of such changes. Of ourse, just like metals, different plastics are changed and how such changes to different extents. There is one weathering factor which is of little or no consequence in the


Compressive Stress - 10³ pounds per square inch
TEMPERATURE 25°C (77°F)
RATE OF STRAINING: 0.050" PER MINUTE
⊙ AS RECEIVED
⊡ IN WATER 7 DAYS AT 25°C (77°F)
△ DRIED 7 DAYS AT 50°C (122°F)
PLEXIGLAS I
GRAPH M 17
Deformation in ½" cube

■ Fig. 13 – Compressive stress-strain versus conditioning


Minutes for 10% Plastic Deformation
Time for 10% Plastic Deformation Vs. "Dead Load" Compressive Stress
⊙ AS RECEIVED
X DRIED 48 HOURS AT 50°C (122°F)
□ DRIED 7 DAYS AT 50°C (122°F)
⊕ DRIED 30 DAYS AT 50°C (122°F) - OVER P_2O_5
△ IN WATER 48 HOURS AT 25°C
○ IN WATER 7 HOURS AT 25°C
TEMPERATURE: 30°C (86°F)
TEST SPECIMEN: 0.5" CUBE
PLEXIGLAS I
GRAPH M 29
Compressive Stress-10³ pounds per square inch

■ Fig. 14 – Time for 10% plastic deformation versus "dead load" compressive stress

application of metals which can be of very considerable significance in the application of plastics. We refer to their moisture content as determined by the relative humidity of the surrounding atmosphere. The extent of the effect of moisture content is illustrated in Figs. 13 and 14.

Atmospheric humidity changes and the effect of these changes on the dimensions and dimensional stability of plastic materials are very important considerations in the engineering applications of plastics. There are two distinct types of responses to such changes. One is the reversible dimensional change displayed by certain so-called "unplasticized" materials. The other is a permanent dimensional change which progressively increases with humidity cycling and changes usually displayed by the so-called "highly plasticized" materials. Both of these types of changes can result in failure in application if they are not given proper consideration. Even though a material may only temporarily expand or contract due to humidity changes, it is possible for it to fail during the cycle due to actual loss of strength or to relative displacement in its assembly with other parts. Of course, when humidity changes cause a permanent dimensional or strength change, the expected service life of the part is shortened.

The subject of the dimensional change of plastic parts due to the relaxation of molding or forming stresses is actually worthy of a separate detailed discussion. A statement of a few fundamental principles – to which we hope considerable thought will be given in future designs of plastic applications – will have to suffice for this discussion. Any plastic material that is subjected to a forming operation during the manufacture of a part, whether this forming is by casting, stretching, blowing, injection, or compressi n molding or extrusion, will contain some residual or "trapped" internal stress. These internal stresses will tend to relieve themselves by producing a shape or dimensional change in that part. The rate with which these changes take place is determined by the service conditions to which the part is subjected. Elevation of service temperature greatly increases the rate with which these dimensional changes take place. Increase of humidity also increases the rate with which these changes take place. It is, therefore, highly desirable that original design should give consideration to the requirements of fabrication in such a way as to permit that fabrication to be accomplished with minimum of internal stress in the parts if a maximum of dimensional stability is sought. This consideration, therefore, indicates the use of cast, machined, or positive-pressure molded parts where minimum permanent dimensional changes due to relaxation are desired. If such methods of fabrication are not suitable, and the part must, therefore, be made by stretching, blowing, injection molding, or extrusion, it should be designed to permit its fabrication by the application of the least possible stress. This, in turn, will result in trapping less internal stress in the finished part and will thereby ensure improved dimensional stability.

■ Summary

May we conclude by emphasizing again that the successful engineering design and application of plastic materials require the consideration of numerous factors that do not ordinarily enter into the design and application of metals.

Plastics may be the wonder engineering materials of the new world – if we intelligently apply them.

Experience wit MAGNESIU

THE tremendous increase in the production of magnesium and the greater incentive to utilize it in reducing the weight of combat airplanes has required information regarding it which is not now covered in the literature or technical handbooks.

In this paper, Mr. Mathes makes available generally some of the more recent information on magnesium which has resulted, in many cases, from questions raised by the aircraft industry itself.

The specific subjects covered are availability, choice of alloys, surface treatment and painting, stress corrosion, joining methods, service experience, effect of gunfire, and primary structures.

■ ■ ■

THE AUTHOR: J. C. MATHES has been with the Dow Chemical Co. since 1935 as development engineer for the Magnesium Division, active in many of the projects which have increased the demand for magnesium from 4,000,000 lb in 1935 to a probably 350,000,000 lb 1943. He has devoted all his time since 1939 to the development of wrought magnesium alloys in aircraft, working with various airplane companies and correlating the research and production of his company in this field to their requirements. Mr. Mathes graduated from the University of Michigan in 1928 with the degree of B.S.C.E.

THE aircraft industry is noted for the meticulous effort it exerts to save weight. The revenue increase per pound of weight saved for commercial aircraft was an adequate incentive for this. Now, however, the importance of weight in military aircraft, whether it be to increase range, load, speed, maneuverability, or climb, has greatly stimulated the effort to reduce empty weight to a minimum. This coinciding with a tremendous increase in the available supply of magnesium, has led to a more than usual stimulus to utilize this metal to reduce weight wherever possible.

As the largest and oldest domestic producer of magnesium, The Dow Chemical Co. has been looked upon as a major source of technical information regarding the design and fabrication of magnesium alloys. In many cases it has been able to answer satisfactorily the questions raised, but in some cases adequate experience or tests from which to supply the information requested has not been on record.

There is a good reason why, in all cases, information pertinent to each specific problem is not readily available. Prior to the war, use of magnesium in aircraft, except for wheels and engine parts, was comparatively restricted and but little opportunity was presented to obtain service experience on what may be common usages in other metals. Lacking service experience, it is necessary to look to the laboratory for answers to many of the questions and, in some cases, proved data cannot always be obtained in this way. Also it has not been possible, in a comparatively few years, to obtain as complete information on magnesium as the many years of experience and research of many companies have made available on other materials. Thus, it is not too surprising that all of the data the aircraft industry needs are not always available or obtainable on short notice.

In preparing this report on "Experience with the Use of Magnesium in Aircraft," it was thought that by giving specific answers to specific questions which have been raised by the aircraft industry, more pertinent information could be given than by treating the subject in a general way. Accordingly, this method of presentation has been adopted.

A. Availability

1. Is the supply of magnesium now adequate?

a. There is now an abundance of magnesium ingot, and magnesium ingot will continue to be abundant unless unknown factors in the war strategy change the situation materially.

b. The fabricating facilities for magnesium alloys are now running at full capacity, but new facilities are being provided that should at least triple the amount of sheet available and should increase the availability of extrusions in all forms by from 5 to 10 times. These new facilities are now coming in and should be completed shortly after January, 1944. Die casting capacity is being expanded so that there should be no shortage here. There seem to be adequate sand casting facilities except for a few of the very intricate and large engine and wheel castings and this condition is also being corrected.

c. One of the difficulties in building up fabrication facilities has been the lack of information on firm requirements by the aircraft manufacturers. This is understandable because in most cases the use of magnesium is for parts that have not heretofore been used in magnesium and some engineering study has been necessary in order to determine whether such requirements will be firm.

The War Production Board have stated, however, that magnesium fabricating facilities will be provided for all cases where firm requirements are shown.

B. Choice of Alloys

1. What factors influence the choice of available alloys, particularly where the physical properties are relatively the same?

a. Castings – Dowmetal H alloy (A-N Specification AN-QQ-M-56, Alloy A) is the most widely used and most easily obtained. Many foundries cast only this alloy. Dowmetal C alloy (A-N Specification AN-QQ-M-56, Alloy C) is used in place of H alloy in the following cases:

[This paper was presented at the SAE National Aircraft Engineering and Production Meeting, Los Angeles, Calif., Oct. 2, 1943.]

the Use of

in AIRCRAFT

by J. C. MATHES
The Dow Chemical Co.

(1) For most permanent-mold castings.
(2) For some pressure-tight castings.
(3) Where greater hardness and static yield strength are more important than toughness.

Dowmetal M alloy (A-N Specification AN-QQ-M-56, Alloy B) is used for cast fittings, such as tank fittings, which must be welded to tanks fabricated of M alloy sheet.

Dowmetal R alloy (AAF Specification 11319, Alloy 1; Navy Specification 46M11) is universally used for die castings.

With either C or H alloy, the as-cast condition should be specified for miscellaneous nonstressed parts because it is the cheapest, the heat-treated condition for maximum elongation and resistance to impact, and the heat-treated and aged condition for maximum yield strength. In some cases, the HTA condition is specified for high-temperature applications, such as some engine parts, where the temperature would cause the HT condition to age in service. This maintains dimensional stability, as some growth occurs in going from the HT to the HTA condition. A new condition, designated as "stabilized" and which can be imposed on either as-cast or HT castings is even more effective than aging as regards dimensional stability and provides a greater ductility than the HTA state. For ordinary applications growth is not a factor as HT castings do not age at temperatures under 200 F.

b. Extrusions – Choice of extrusions is indicated as follows:

(1) Dowmetal M alloy (AAF Specification 11336, alloy B; Bureau of Aeronautics Specification M-314C, Alloy 11) for nonstressed parts or for parts to be welded to M alloy sheet assemblies. It is the cheapest.

(2) Dowmetal FS-1 alloy (AAF Specification 11320, Alloy 1) for parts requiring higher strength than M at minimum cost.

(3) Dowmetal J-1 alloy (AAF Specification 11335, Alloy 1; Bureau of Aeronautics Specification M-314C, Alloy 8) is slightly stronger than FS-1 and should be specified for parts to be welded to J-1 sheet.

(4) Dowmetal O-1 alloy is the strongest of the extruded alloys and, in the aged condition, offers the highest yield strengths, particularly in compression.

c. Forgings.

(1) Dowmetal J-1 alloy (AAF Specification 11345; Navy Specification 46M13, Alloy 8) is used for moderately stressed forgings, such as valve or pump bodies, where forgings are specified primarily to avoid porosity or seepage; or where forgings primarily replace Class I castings.

(2) Dowmetal O-1 alloy (AAF Specification 113 1A; Navy Specification 46M13 Alloy 9) should be specified for forgings requiring maximum strength.

(3) Dowmetal M alloy (Navy Specification 46M13, Alloy 11) can be hammer forged and is the least expensive. It is used where strength requirements are not rigid, and where soundness, as compared to castings, is the major consideration.

d. Sheet.

(1) Dowmetal M alloy (AAF Specification 11339, Alloy 1; Navy Specification 46M2, Alloy 11) should be used in the annealed condition for nonstructural parts requiring the maximum combination of formability and weldability.

(2) Dowmetal FS-1 Alloy (AAF Specification 11340; Navy Specification 46M2, Alloy 18) is stronger than M. In the annealed condition (FS-1a) its formability is equal to, or slightly better than M, particularly for cold forming. It is not as weldable as M, however. In the hard-rolled condition (FS-1h) it is one of the strongest magnesium sheet alloys. However, it anneals at fairly low temperatures (275-300 F), and below these temperatures its formability is not much improved over room temperature. Therefore, it is difficult to use where hot forming is necessary. It is not as readily arc-welded as J-1h.

(3) Dowmetal J-1 alloy (AAF Specification 11338, Alloy 2; Navy Specification 47M2, Alloy 8) is the strongest magnesium sheet alloy in the hard-rolled condition and has been used for all work to date on primary structures. Due to its better weldability, it is usually specified for assemblies to be joined by arc welding. It does not anneal appreciably at 400 F, and can be formed at temperatures as high as that. Therefore, it has been used for applications requiring maximum strength, which involve forming that cannot be done on hard sheet at room temperature.

C. Surface Treatment and Painting

1. Many surface treatments are permitted by current specifications. What schedule has been found to be most widely used?

a. The following practice is widely used for cast H or C alloy engine and accessory parts:

(1) Protection during storage – chrome pickle as put on by producer.
(2) Protection between machining operations – light oil.
(3) Final processing:
(a) Degrease with organic solvent.
(b) Alkaline clean.
(c) Dow No. 7 chemical treatment.
(d) One coat P-27 baked primer.
(e) Two coats Specification E-5e enamel, baked.

b. The following practice is recommended for airframe parts.

(1) Protection during storage – chrome pickle or oil coating.

(2) Protection between operations if above is removed – light oil.

(3) Final processing:

(a) Degrease with organic solvent.

(b) Alkaline clean.

(c) Dow No. 7 chemical treatment (or Dow No. 10 for M alloy).

Note: Dow No. 1 (chrome pickle, either dip or brush on) is generally satisfactory but is not the best.

(d) Two coats P-27 primer (AN-TT-P-656a), air dry.

(e) Two coats (AN-TT-L-51) lacquer, pigmented, air dry.

Note: Two coats of AN-TT-V-118 varnish pigmented with aluminum powder give maximum protection for worst conditions.

(f) Camouflage paint, if required.

2. How do various schedules compare in salt spray test?

a. This comparison is shown in Table 1.

Table 1 – Salt Spray Resistance of Various Protective Finishes

This test on J-1h sheet is still in progress. Panels were exposed facing the spray at an angle 10 deg from vertical. This is a more severe test that the usual vertical position of exposure, and gives more uniform and consistent results. The corrosion was determined by visual observations.

Chemical Treatment	Hours in 20% NaCl Spray Until Initial Failure (Visual Observation)
a. 1 coat P-27 primer + 2 coats L-12 lacquer + Al:	
Brush-on chrome-pickle	486
Dip	491
Dow No. 7 treatment	1162
b. 2 coats P-27 primer + 2 coats L-12 lacquer + Al:	
Brush-on chrome-pickle	857
Dip	872
Dow No. 7 treatment	1869
c. 1 coat P-27 primer + 2 coats V-10 varnish + Al:	
Brush-on chrome-pickle	None at 5500
Dip	None at 5500
Dow No. 7 treatment	None at 5500
d. 2 coats P-27 primer + 2 coats V-10 varnish + Al:	
Brush-on chrome-pickle	None at 5500
Dip	None at 5500
Dow No. 7 treatment	None at 5500

3. What are the recommended treatments for faying surfaces?

a. Proper treatment of faying surfaces is shown in the following tabulation:

Materials	Exposure	
	Usual Weathering or Interior	Salt Atmosphere
Magnesium to magnesium	2 coats P-27	2 coats P-27
Magnesium to 24S-T	2 coats P-27	2 coats P-27 + sealer
Magnesium to steel [1]	2 coats P-27 + sealer	2 coats P-27 + sealer

[1] Should be cadmium or zinc plated.

b. Results of corrosion tests on faying surfaces are shown in Table 2.

4. What protection is required for steel inserts and bolts?

a. Steel inserts should, in all cases, be cadmium plated. If shrunk or pressed in, primer or sealer should be used to form a protecting fillet at the external joint.

b. Steel bolts should be cadmium plated. Bolt and nut should be dipped in wet primer before inserting.

5. How do the American alloys compare with the German alloys on corrosion?

All of the magnesium parts on a JU-88 and a ME-110 were examined. It was found that the impurities in the German alloys were uncontrolled so that the base metal corrosion rate was very high as compared to American alloys, particularly of the J-1 and O-1 alloy type. In spite of this, and of the fact that the protective system was a simple chrome pickle plus primer and an inferior paint, there were no corrosion failures observed.

D. Stress Corrosion, Compressive Yield Strength, and Modulus of Elasticity

1. How serious is the susceptibility to stress corrosion of J-1h sheet?

a. Work on this subject is now in progress, both in the industry's laboratories and as a part of the National Defense Research Committee program. Present data are therefore tentative. However, laboratory tests have clearly demonstrated that magnesium alloys resemble other metals, in that stress corrosion may occur in all of them under proper conditions of stress and corrosive medium.

Stress corrosion tests have included specimens under a constant tensile stress as well as elastically deformed specimens, immersed both in NaCl solution and NaCl + K_2CrO_4. In addition, specimens of both types are undergoing exposure tests in inland and sea-coast atmospheres. The laboratory tests show quite clearly that stress corrosion cracks will result if stresses are maintained constantly at levels over ½ the tensile yield strength and under corrosive conditions.

However, since service conditions are not exactly comparable to these laboratory conditions, it is difficult to interpret the results in terms of service life. Ordinarily, applied stresses of sufficient magnitude to cause stress corrosion cracking are not developed except for short periods during violent maneuvers and, even then, not in an excessively corrosive medium. With usual design factors, the continuous stresses at 1g or 2g loading are below the stress corrosion limit so that stress corrosion failure should only result from "built-in" stresses.

The most serious offender has been the high residual stresses resulting from arc welding. It is believed that stress relieving for 1 hr at 400 F will prevent this. Complete welded assemblies have been exposed at the Atlantic sea coast since November, 1942. Those assemblies which

Table 2 – Effect of Corrosive Exposure on Magnesium

Alloy	Protection	Atmospheric, 2½ Years	20% NaCl Spray at 95 F, 1000 Hr	Sea Coast, 2 Years
J-1	Dow No. 7 + 2 coats P-27 + 2 coats L-12 + Al	No loss	No loss	.....
J-1	Dow No. 7 + 2 coats P-27 + 2 coats V-10 + Al	No loss	No loss	No loss
H	None	4% loss	.....	1% loss
H	Dow No. 7 + 2 coats P-27 + 2 coats V-10 + Al	No loss	.....	No loss
J-1 to J-1 [a]	56S rivets-sealed-painted 2 coats P-27 + 2 coats V-10 + Al	No joint corrosion	.....	No joint corrosion
J-1 to 24S-T Alclad	AM56S rivets painted (no sealer)	No joint corrosion	Joint corrosion in less than 100 hr	Slight joint corrosion
J-1 to 24S-T Alclad	56S rivets-sealed-painted 2 coats P-27 + 2 coats V-10 + Al	No joint corrosion	No joint corrosion	None at 5 months
J-1 to 24S-T	56S rivets-sealed-painted 2 coats P-27 + 2 coats V-10 + Al	.....	.....	None at 5 months

[a] Sealers: Fullers zinc chromate, Watson-Standard P-6712 zinc chromate primer, Horniastic, Alumilastic.

were not stress relieved cracked at the weld area in about five weeks if bare and in about five months if painted. Those assemblies which were stress relieved have shown no tendency toward stress corrosion even in areas that were purposely left unpainted. Cold-formed, unannealed panels have likewise shown no tendency toward stress corrosion even though unpainted. Riveted wings, constructed of J-1h sheet, have been flying for over 1½ years with no evidence of stress corrosion. It appears, therefore, that, properly appreciated and guarded against, stress corrosion should not be a serious factor.

2. Does the modulus of elasticity of magnesium vary with alloy, condition, and temper?

a. The commonly accepted value for modulus of elasticity is 6,500,000 psi. Our own tests on many specimens show the range to be 6,200,000 to 6,600,000, although some laboratories have reported results much lower than this. It is our experience that accurate tests run primarily to determine modulus of elasticity, will always give results in the range indicated. Routine tests run on short specimens may give low values because any error in testing procedure will tend to lower the modulus value, even though they are entirely adequate for determining yield and ultimate strengths. Another factor tending to indicate lower values of the true modulus, is the comparatively low proportional limit of many of the alloys. Because of this, the secant modulus at the design stress level may be a more useful tool than the true modulus.

3. How does the relationship of compressive yield strength compare with tensile yield strength in the various alloys and conditions?

a. Generally speaking, the compressive yield strength is equal to the tensile yield strength for the cast alloys, but is substantially less than the tensile yield strength for wrought alloys. In the latter case, the artificial aging of wrought alloys, when possible, brings the relationship of compressive yield strength to tensile yield strength more nearly unity.

The following tabulation illustrates the relationship between minimum compressive and tensile yield strengths for magnesium in its various forms:

		Tensile Yield Strength, psi	Compressive Yield Strength, psi
Sand-cast Dowmetal	H	10,000	10,000
Sand-cast Dowmetal	HHT	10,000	10,000
Sand-cast Dowmetal	HHTA	16,000	16,000
Rolled Dowmetal	J-1h	29,000	20,000
Extruded	J-1 shapes	23,000	12,000
As-extruded Dowmetal	O-1	25,000	16,000
Extruded and aged	O-1	27,000	22,000
Extruded, heat-treated, and aged	O-1	30,000	30,000

E. Joining

1. Riveting.

a. What standards have been determined for riveting practice?

(1) Use 56S-O anodized rivets for dimpled sheet. Use 56S-¼H anodized rivets for brazier head or machine countersunk rivets.

(2) 56S rivets have practically no electric potential difference when in contact with magnesium and should be used wherever moisture is present. They can be driven dry.

(3) Can A17S-T rivets be used?

We do not recommend A17S-T rivets because they are inferior to 56S in contact corrosion. They have been used successfully for applications not subjected to excessive moisture or corrosive atmosphere. They should be dipped in wet primer before driving.

(4) For flush riveting gages of less than 0.040, we have used dimpled sheet and 120-deg countersunk rivets. Sheet must be heated for dimpling. Satisfactory tools have been developed for this purpose. We have not yet been able to dimple sheet satisfactorily to accommodate the standard 100-deg rivet.

(5) Machine countersunk rivets may be used for gages of 0.040 and greater. When countersinking the hole, a minimum thickness of 0.015 in. should be left at the base of the countersink. Any standard rivet head can be used.

(6) The recommended edge distance is $2\frac{1}{2}d$ with a minimum of $2d$. A minimum spacing of $4d$ is generally recommended.

(7) Design allowables for 56S-¼h rivets and J-1h sheet are as follows:

	Brazier Head, psi	Dimpled, psi	Countersunk, psi
Net tension	40,000[2]	40,000[2]	35,000[2]
Shear	25,000	23,000[3]	16,700
Bearing (4% permanent set)	60,000	60,000	40,000

The following table shows recommended bucktail practice:

Rivet Diameter, in.	Nominal Bucktail Dimensions		Minimum Bucktail Dimensions to Develop Strength	
	$h = 0.5d$	$D = 1.4d$	h	$D = 1.33d$
3/32	0.047	0.131	0.038	0.125
1/8	0.063	0.175	0.044	0.167
5/32	0.078	0.218	0.057	0.208
3/16	0.094	0.263	0.070	0.250

d = Rivet diameter
D = Bucktail diameter
h = Bucktail height

2. Spot welding.

a. What is the recommended finishing practice for spot welding?

(1) For parts which will not be subjected to weathering, such as interior furnishings or tank baffles, our practice is as follows:

(a) Before welding remove the chrome pickle at the weld seams from both the faying surfaces and outside surfaces by wire brushing, steel wool, or abrasive cloth.

(b) Spot weld.

(c) Chrome pickle (Dow No. 1) the entire assembly.

(d) Prime and paint in the regular manner.

(2) For parts which will be subjected to weathering, the following practice is recommended.

(a) Clean as in (1)a, above.

(b) Paint both faying surfaces where the chrome pickle has been removed with Watson-Standard P-6712 primer.

(c) Spot weld through this primer. The primer remains tacky for sometime and can readily be welded through for 30 min after priming.

[2] In joints subjected to excessive local bending, such as single-row lap joints, these values should be reduced to 25,000 psi for brazier head and dimpled rivets, and 18,000 psi for countersunk rivets. Double-row lap joints are, therefore, preferable.

[3] This figure applies to 56S-0 rivets.

(d) After welding, clean the outside of the welds with a grinding compound such as Lea L on a buffing wheel.

(e) Chrome pickle (Dow No. 1) the entire assembly. (Do not use boiling treatments, such as Dow No. 7.)

(f) Prime and paint in regular manner.

(g) This latter schedule has been proved best in laboratory tests but no production runs have been attempted using it. In all cases, wire brushing prior to welding is necessary even where using plain, oiled sheet rather than pickled sheet. Solvent or alkaline cleaners do not adequately remove the thin oxide film which is on all plain sheet.

3. Arc welding.

a. What type of jigs are necessary for arc welding?

(1) Due to the great pressures exerted during arc welding, jigs must be very rigid and the parts firmly secured.

b. Does stress-relieving for 1 hr at 400 F cause serious distortion if the assembly is not jigged in the oven?

(1) We have not found it necessary to stress relieve in jigs except in an assembly which does not have the ends tied together structurally. Assemblies which are held in shape by formers or bulkheads have caused no trouble. Large surfaces with but a slight curvature may have a tendency to buckle inward as the stress is relieved. To prevent this, it is not necessary to stress relieve in the entire welding jig as light formers can be temporarily secured in place during stress relieving as required.

c. What practice is recommended for stress relieving assemblies which are combined riveted and arc-welded structures, and where the relieving temperature would burn out primer at the faying surfaces?

(1) A recommended procedure follows:

(a) Chemically treat (Dow No. 1, No. 7, or No. 10) the component parts.

(b) Remove chemical treatment at weld edge surfaces with steel wool and, by shearing or grinding, obtain a freshly cut edge. Usual cleaning methods, such as wire brushing, will not penetrate a sheared edge that has been pickled.

(c) Assemble riveted parts, using alumilastic paste or Fuller's chromate compound to protect faying surfaces. These preparations will withstand the stress relieving temperature.

(d) Make arc welds.

(e) Stress relieve at 400 F for 1 hr.

(f) Chrome pickle the areas that have been cleaned for welding by the "brush on" method.

(g) Prime and paint the entire assembly according to recommended schedule.

Note: The stress relieving temperature does not affect the corrosion or strength properties of 56S rivets, but does affect some of the other aluminum alloys.

d. Is there a satisfactory method of inspecting arc-welded joints?

(1) It is our opinion that visual inspection will detect any surface cracks which might lead to failure. X-ray inspection is, of course, the most reliable inspection but it is expensive and certain joints do not permit a satisfactory radiograph. We have done considerable cross checking between X-ray and visual inspection and have found that in practically all cases shrinkage cracks discovered by the X-ray were also discernible by visual inspection if the joint was chrome pickled prior to inspection and if a sufficiently powerful glass was used. It appears, therefore, that shrinkage cracks will always appear on the surface.

Work is now being done on the use of "Zyglo" for the inspection of arc welds. This looks very promising to date. Internal porosity can only be determined by X-ray. Experience has shown, however, that if the surfaces and edges are properly and adequately cleaned, any porosity in a weld that passes visual inspection will be so minor that the strength of the weld is not reduced.

F. Service Experience

1. How have magnesium parts stood up in service?

a. Castings, forgings, and extrusions have been practically 100% satisfactory. There are several million pounds per month of sand castings being used in engines, wheels, and airframes. We rarely hear of any service troubles from these. Although not used extensively, forgings and extrusions have apparently given equally as good service. The case of sheet is somewhat different, and practically all instances where magnesium has not done the job expected have been applications utilizing sheet. Many, of course, have been satisfactory and sheet is widely used for oil tanks, wheel fairing, instrument panels, turret parts, miscellaneous cases, and so on.

Most of the failures have occurred on so-called non-stressed parts, using thin gages which, in many cases, have been substituted for the stronger aluminum alloy on a gage for gage basis. Such parts, while not primarily stressed, are subjected in many cases to considerable vibration and abuse. Thin magnesium sheet, perhaps due to its lower ductility and greater notch sensitivity, does not seem to stand this service as satisfactorily. Many of these failures have occurred at points of attachment and in some cases, the trouble was overcome by proper edge reinforcement and the use of doublers.

It is apparently the current practice, on aluminum, to reinforce large, flat panels with stiffeners which do not necessarily tie into the main frame. Doors and panels constructed in this manner of magnesium sheet have not worked out and failure has occurred by cracking at the ends of the stiffeners where the vibration is concentrated. Similarly, hinges or fittings which have been attached to sheet and not reinforced at such points by doublers have often failed by cracking.

Flooring is another example where conventional aluminum design has not always been satisfactory for thin magnesium sheet. Here the local overloading and impact have sometimes caused premature failure. Our observation has indicated a satisfactory service experience in both British and German machines for such parts as fairing, cowlings, doors, seats, control surfaces, and fuel tanks. In all cases, sheet corresponding to Dowmetal M was used although the minimum gages were somewhat greater than usual American practice. Very few cases were observed where sheet thinner than 0.040 was used.

G. Effect of Gunfire

1. What would be the effect of gunfire on a magnesium structure?

a. Some work has been done to determine the effect of gunfire on magnesium. This has included firing tests on complete structures, such as wings, as well as on small

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STAINLESS STEELS— Hot and Otherwise

by H. A. CAMPBELL
Solar Aircraft Co.

AIRCRAFT exhaust systems must be designed to withstand service conditions that include severe vibrations, high temperatures from the hot exhaust gases passing along the inside of the pipes at relatively high velocities, while the outside of the pipe system may be subjected to rain and sea atmosphere.

These extreme conditions of operation led to difficulties in the application of stainless steel to the exhaust system.

Failures caused by intergranular corrosion from carbide precipitation need no longer be feared when properly stabilized 18-8 stainless steel is used.

Stainless steel can be stabilized with either titanium or columbium, the cheaper and more easily worked titanium-type steel being satisfactory for most installations, although there are applications where the columbium type must be used because of its higher yield and tensile strengths.

This improved stainless, however, has not solved the problem of corrosion at high temperatures in the presence of zinc. Zinc, as bits from drop hammer dies, plating, or in any other form, should be kept away from stainless steel at elevated temperatures.

Another problem that is sometimes annoying is the cracking of welds in thin stainless sheets such as are used in exhaust systems. Rigid control of the composition of the welding flux has been found to eliminate this trouble.

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THE industrial expansion of the last two years has seen fabrication of aircraft exhaust systems by many new manufacturers relatively inexperienced in this field. These fabricators, together with many newcomers to the aircraft industry as a whole, have been confronted by problems seeming to cause them great concern, of which a discussion is offered here. These problems arise from the engineering applications of stainless steel to exhaust systems and have to do with:

1. The comparative advantages of titanium- and columbium-stabilized 18-8 steel.
2. Carbide precipitation effects.
3. Welding of thin sheets.
4. Metallurgical effects of zinc alloys on stainless.

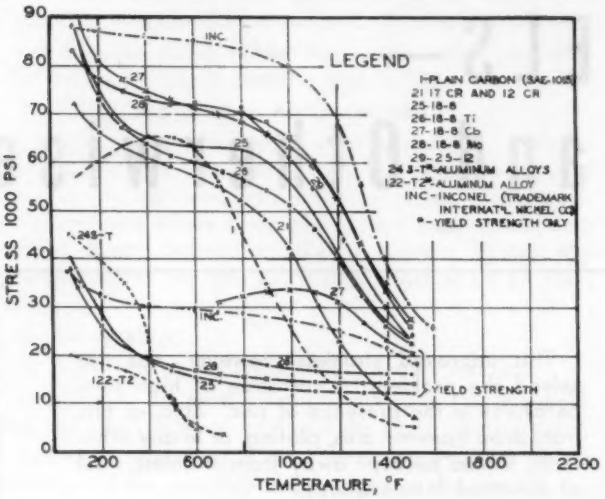
These problems relate to service life and offer difficulties in discussion because of a marked scarcity of test data, accurate or otherwise, on manifold service life. While no standardized method of predicting service life has appeared, it is nevertheless possible to draw tentative conclusions concerning it from the few data available, as well as from some intelligent ignorance derived from a study of failures. The conditions of service to be met include severe vibrations in structures usually supported as interconnected cantilevers (in the case of the collector ring and some liquid-cooled in-line applications) or as cantilever tubes or shapes (in the case of stack-type exhausts). On engines of 1000 hp or more, temperatures of from 1000 to 1700 F are found in these structures. Hot exhaust gases pass along the inside at relatively high velocities, while rain and sea atmosphere are frequent visitors on the outsides of the pipe system.

Under such service conditions it appears that titanium-stabilized 18-8 stainless meets all the requirements for most installations, although there are some places where titanium-stabilized has failed and columbium-stabilized has done the job. The statement holds with the proviso that the welding is adequate and that the steel structure is free from zinc, and with the memory that we are here discussing high-powered engines for combat service and long-distance transport.

■ Columbium- and Titanium-Stabilized Steels

For the larger portion of designs titanium-stabilized (type 321) steel does the job well although there are cases where columbium-stabilized (type 347) steel is applied because of its higher values of yield and tensile strengths.

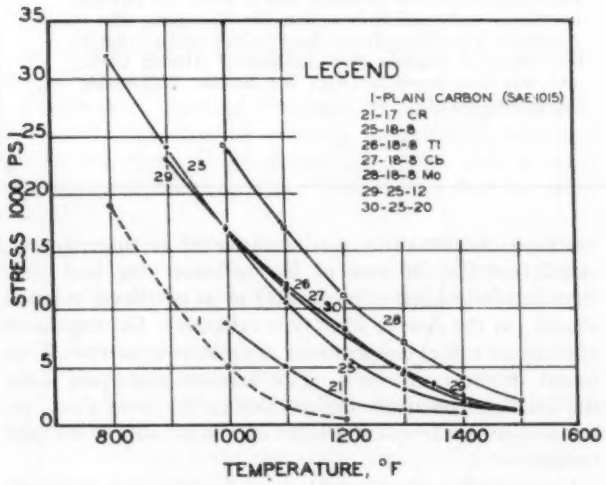
[This paper was presented at a meeting of the Southern California Section of the SAE, San Diego, Calif., May 14, 1943.]



■ **Fig. 1 – Trend of tensile and yield strengths for various heat-resisting materials with increasing temperature (short-time tensile tests)**

Note: Curves 25, 26, 27, 28, and 29 are for austenitic stainless steels. The tensile strength and yield strength values shown for them are average curves, except the yield strength of curve 27, which is from data from a single test only. While these materials exhibit static yield and tensile strength spreads of 8000 psi, in actual use no substantial superiority is exhibited by any one material because other factors, such as abuse, design, and workmanship, have much greater effect.

The curves for aluminum alloys 24S-T and 122-T2 are included for comparative purposes.

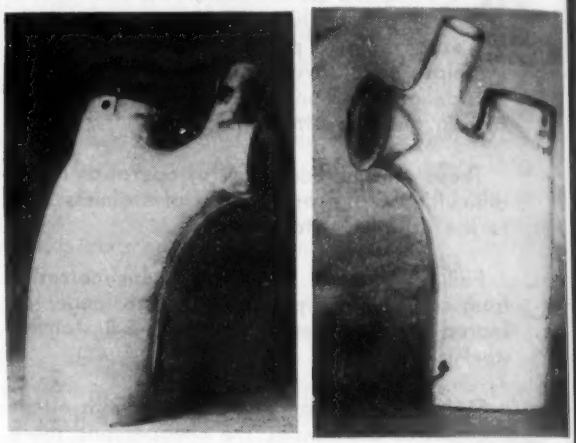


■ **Fig. 2 – Trend of creep strengths of various heat-resisting steels at elevated temperatures (1% elongation in 10,000 hr)**

Note: Of the two opposing processes – strain hardening from plastic deformation and relief of strain from softening at temperature – going on simultaneously in any tensile-type test, relief of strain will be favored by slow rates of loading and increased temperatures.

These creep curves compared with the tensile strength curves demonstrate the increasing effect of relief of strain in slow tests. The values are averages and are not intended for exact design purposes.

Figs. 1 and 2[1] summarize average values for some mechanical properties of these alloys. One may say, "Why not just use type 347 steel in all cases, building in some accident insurance?" Type 347 costs more than type 321. Type 347 is somewhat harder and more expensive to work due to faster work-hardening than 321, and in the majority of flying high-powered jobs, 321 is giving satisfactory service. Thus, 347 accident insurance is not often necessary. Even in cases where the operating temperature is such that columbium-stabilized steel seems called for, gain in overall service life of the complete exhaust system and its shrouding may possibly result from better cooling of the system. A well-done cooling job could permit the use of the cheaper steel. Under current conditions, short ferro-columbium supplies make use of type 321 steels a conservation measure also. Fig. 3 shows a manifold section[2]



■ **Fig. 3A (left) – This is a section from an airplane exhaust system collector ring that was in service over 3000 hr on 800-1200-hp engines. The ring was made up of two different types of metal welded together. This side was made of type 321 stainless steel: 18-8 stabilized with titanium**

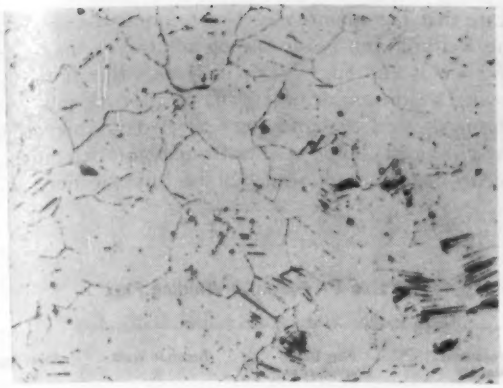
■ **Fig. 3B (right) – This is the other side of the ring shown in Fig. 3A. It was made of type 347 stainless steel: 18-8 stabilized with columbium**

■ **Fig. 3C – Interior of stack at arrow shown on Fig. 3B – shows mechanical erosion occasionally encountered, reason unknown**

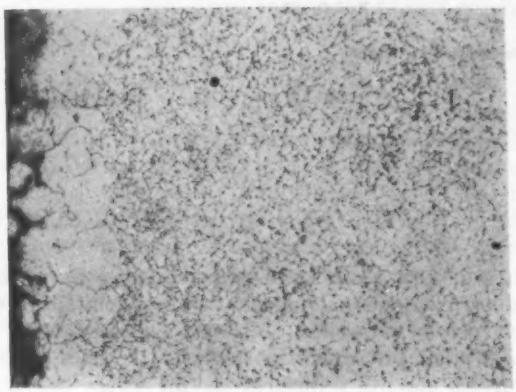
composed of both types of steel. Fig. 4 shows microstructures of these materials after 3000 hr service on an 800-1200 hp engine.

[1] Data compiled from all known sources from 1930 to date, by Carnegie-Illinois Steel Corp.

[2] Solar-TWA manifold. Compare Carnegie-Illinois Steel Corp. Work Works Report 131; 1941.

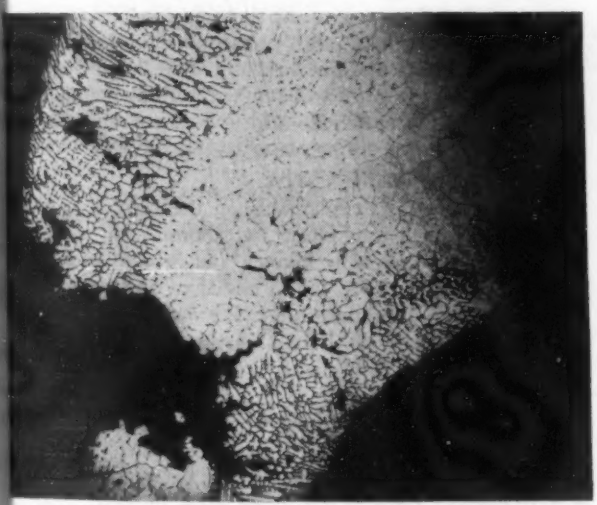


■ Fig. 4A – Microstructure of a section of Fig. 3A – 100X magnification, after corrosion testing in boiling $CuSO_4$-H_2SO_4 solution: not abnormal



■ Fig. 4B – Microstructure of a section of Fig. 3B – 100X magnification, after corrosion testing in boiling $CuSO_4$-H_2SO_4 solution: not abnormal

a phenomenon discussed by Bain, Aborn, and Rutherford in 1933[3], and suggestions are heard still that present-day failures are due to this same phenomenon. The effect of intergranular corrosion is indicated in Fig. 5, in which it is noticeable that grain boundary attack of the material has occurred and that some of the grains have fallen away from the structure. When we take a sample of a stabilized material and test it for sensitivity to intergranular attack by heating it for 2 hr at 1200-1250 F, then subject it to a boiling copper sulfate-sulfuric acid solution for 48 hr and subsequently examine a section of it under the microscope, structures such as that shown by Fig. 6 are revealed, but these structures do not reveal intergranular attack unless carbon in an amount greater than that for which the material was stabilized has been added to the constitution by faulty processing or otherwise. Under conditions of exhaust system service, some part or parts of the system will be operating in this so-called sensitization temperature range, which naturally produces an agglomeration of titanium or columbium carbides or cyanonitrides. Fig. 7 shows a section taken from an exhaust system[4] made from type 321 steel, which was in service approximately 3000 hr and which was retired after it could no longer be repaired in the field. The appearance was such as to suggest that much carbon was present, but the test described above revealed no intergranular attack on this material below the surface. It would appear, therefore, that, in material in which titanium is present to the extent of at least four times the carbon content or in which columbium is present to the extent of at least eight times the carbon content and to which carbon has not been added during the processing of the product, there is little to fear from carbide precipitation leading to intergranular corrosion and subsequent failure of the part.



■ Fig. 5 – Effect of intergranular corrosion

■ Fig. 6 – Intergranular corrosion does not occur with a properly stabilized material

Carbide Precipitation

A few early failures of exhaust system parts were attributed to intergranular corrosion from carbide precipitation,

Products of combustion in exhaust gases usually appear as carbon dioxide, oxygen, carbon monoxide, hydrogen, methane, together with nitrogen, with traces of oxides of

[3] See ASST Transactions, Vol. 21, May, 1933, pp. 481-509: "The Nature and Prevention of Intergranular Corrosion in Austenitic Stainless Steels," by E. C. Bain, R. H. Aborn, and J. J. B. Rutherford.

[4] Solar Laboratory Specimen 119ZM; 1941. From retired manifold from Pan-Am S-42.

■ Fig. 7 – Section taken from an exhaust system made of type 321 steel that was in service about 3000 hr – no intergranular corrosion is present

lead.[5, 6] These substances usually cause no immediate structural troubles in exhaust systems because of attack of the steels. In the cases of severe failures examined in detail over the last three years, no evidence of intergranular corrosion was found, due to the material constitution as such, although two cases have appeared in which the effects of welding gave rise to such attack.

Of course, if during a welding operation the weld is carburized because of excess acetylene in the welding flame or if a carbon-bearing substance such as shellac is used on a seam or other joint, as a vehicle for welding flux, carbon is then dissolved in the weld zone and may give rise to trouble.

A stabilizing heat-treatment consisting of holding the material at 1550-1650 F for two hours, more or less, has been used where it is desired to establish the maximum possible resistance to intergranular corrosion in some specific media. In exhaust systems, however, this stabilizing heat-treatment appears to be unnecessary inasmuch as there seem to be no media encountered under service conditions so highly corrosive as to require the establishment of such maximum resistance. Stabilization in this sense is believed to occur, to some degree in any case, from exhaust collector service over lengths of time at which the structure is held at temperatures ranging from 1000 to 1600 F.

■ Welding of Thin Sheets

Cracking of welds in thin stainless steel sheets such as are used for the fabrication of exhaust systems often becomes a very annoying problem, especially during the training of new welders. In an investigation of this subject started long ago, it was found that in addition to the ever-present variables of welding heat, flame adjustment, and fitting of the joint, the constitution of the welding flux itself plays an important part. Experimentation seems to indicate that the quantity of boron-containing compounds in the welding flux is capable of influencing the tendency of butt welds and edge welds to crack either immediately following welding, during repair, or after the pipe section has been allowed to stand for several days after welding. Table 1 indicates some results from boron determinations in weld metals made during the investigation. Eight groups of butt welds were prepared using conventional welding techniques, in which the flux constitution was very carefully controlled. All welding was done by the same operator on about 100 samples. The welds exhibited tendencies toward brittleness as the borax and boric-anhydride constituents increased in the constitution of the flux. These two substances are found in many stainless steel welding fluxes. After the welding flux constitution was put under rigid control with respect to these compounds and with respect to the slagging constituents, much less trouble was evident with new welders and the volume of repair welding was cut in half. The flux being used currently is compounded to be mixed with methanol under close control of particle size and is kept continuously agitated so that the resulting paste is very smooth and uniform in consistency.

Table 1 – Boron in Welding Flux

Specimen Number	Flux Boron Constituents, %	Boron in Weld, %	Sheet Gage, in.
Acetylene Welds			
282 Z	B_2O_3 17.0	0.044	0.061
286 Z	B_2O_3 10.8 $Na_2B_4O_7$ 7.0	0.006	0.061
289 Z1	B_2O_3 49.7 $Na_2B_4O_7$ 33.3	0.175	0.061
290 Z1		0.098	0.030
Atomic-Hydrogen Arc-Welds			
293 Z1	B_2O_3 49.7 $Na_2B_4O_7$ 33.3	0.235	0.061
293 Z2		0.150	0.030
297 Z1	Boron Nil	Nil	0.061
297 Z2		Nil	0.030

The results appear explainable on the basis of work in 1932 concerning the effects of boron on austenitic chromium-nickel steels by Bennek and Schafmeister,[7] who showed that about 0.5% boron added to 0.15-0.16 carbon, 18-8 steels would produce precipitation-hardening effects sufficient to increase the Brinell hardness from 150-205 up to as high as 315-317 following a water quench from 1150 C and prolonged heating at 700 C. Cornelius[8] in 1939 reported a study of six groups of steel among which was 18-8 steel containing 0.9-1.5% of boron which exhibited similar behavior. Boron seems to be in wider use today as a hardening element for steels than formerly, due to war conditions, a fact which led to the turn of our attention in this direction during the investigation of the cracking problem. While the hypothesis may be completely

[5] See SAE Transactions, Vol. 38, March, 1936, pp. 90-98, 116: "Relation of Exhaust Gas Composition to Air-Fuel Ratio," by B. A. D'Alleva, and W. G. Lovell.

[6] See "Exhaust Valve Materials for Internal-Combustion Engines" by S. D. Heron, O. E. Harder, and M. R. Nestor. In ASTM Symposium on "New Materials in Transportation," pp. 1-26; 1940.

[7] See *Archiv für das Eisenhüttenwesen*, Vol. 5, June, 1932, pp. 615-620: "Precipitation Hardening of 18-8 Steels by Additions of Beryllium, Boron, or Titanium; and Effect on Corrosion Resistance," by H. Bennek and P. Schafmeister.

[8] See *Archiv für das Eisenhüttenwesen*, Vol. 12, April, 1939, pp. 499-505: "Hardening of Boron-Containing Austenitic Chrome-Nickel Steels by Tempering after Quenching," by H. Cornelius.

■ Fig. 8 – Field failure of exhaust system part

wrong in the case of the 18-10 steels we customarily use, it is thus far the explanation which seems to fit circumstances best. Certain it is that since the institution of flux control as above described, the welding results have improved and new welders find their work to be controlled more easily.

■ Metallurgical Effects of Zinc Alloys

In the course of the cracking investigation mentioned, it was discovered that bits of zinc alloy from drop hammer dies could also produce cracks in stainless during the annealing or other heating operation if the bits of zinc were allowed to remain on the metal surface, and this is no new knowledge as the point was brought out by Anderson, Edmunds, and Siller[9] in 1940 and again by Henry and Schroeder[10] in 1942, while in 1939 a private report of the Carnegie-Illinois Steel Corp. Metallurgical Department[11] set forth pretty much the same knowledge. The die-casting industry has known for some time that zinc alloys will dissolve the iron of the melting pot. It was thought, however, that cracking of this sort had been confined to the factory, but several failures in the field, similar to that shown in Fig. 8, showed otherwise, and there appeared to be cause for some alarm when it was decided recently by National production authorities to substitute zinc for cadmium plating on aircraft bolts, many of which are used in exhaust systems. In view of the evidence developed with the cooperation of manufacturers and others, it is hoped that the use of cadmium for such bolts will be ordered in the near future, as it is possible that an exhaust system may fail because of the presence of zinc, thereby putting an aircraft out of service. Fig. 9 shows at 200 magnifications the surface of type 321 stainless after 10 min contact with molten lead at 1950 F. No attack is evident. Fig. 10 shows, at 200 magnifications, the surface of the same steel after 10 min in contact with molten cadmium at 1950 F. Still no attack is evident. Nor is any attack evident if the steel is held in contact with molten lead and molten cadmium for 2 hr at 1500 F. With zinc the story is different. Fig. 11 shows the results of 2 hr in contact with zinc at 1500 F, in unstressed condi-

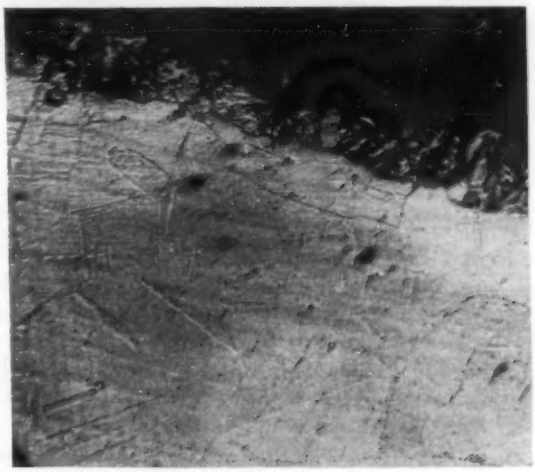
[9] See AIME Transactions, Vol. 137, 1940, pp. 400-413: "A Study of the Action of Molten Zinc Alloys on Pressure Die-Casting Equipment," by E. A. Anderson, G. Edmunds, and C. W. Siller.

[10] See *The Welding Journal*, Part II, Welding Research Supplement, Vol. 21, June, 1942, pp. 287s-290s: "A Study of the Embrittling Effect of Zinc upon Stainless Steel," by O. H. Henry and R. W. Schroeder.

[11] See Carnegie-Illinois Steel Corp. Wood Works Report 90: "Zinc and Lead Penetration of Stainless Steels and Inconel in Airplane Exhaust Manifolds," by P. W. K. Menard and E. H. Wyche, 1939. (Reprinted in Solar RD Report 299.)

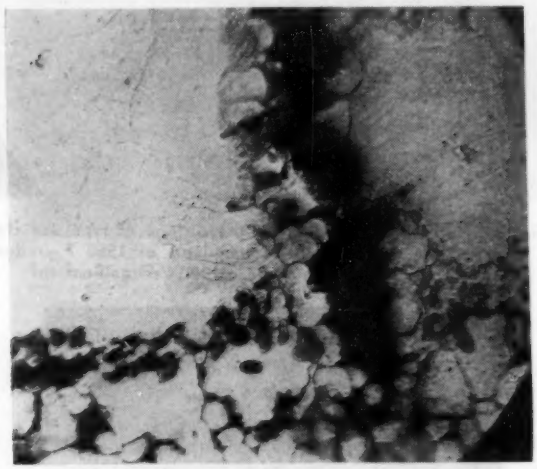
■ Fig. 9 – No attack is evident on the surface of type 321 stainless after 10-min contact with molten lead at 1950 F – reduced from photomicrograph taken at 200X magnifications

■ Fig. 10 – Still no attack is evident on the surface of the same steel shown in Fig. 9 after 10 min in contact with molten cadmium at 1950 F – reduced from photomicrograph taken at 200X magnifications

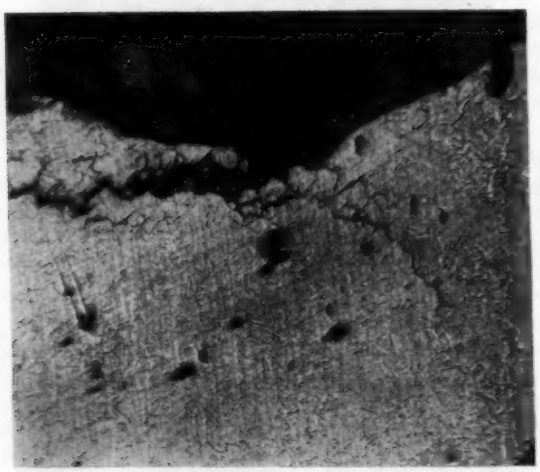


■ Fig. 11 – Stainless steel after 2 hr in contact with zinc at 1500 F, in the unstressed condition

tion, and Fig. 12 shows the same steel several grains away from the surface after 10 min contact with zinc at 1950 F, unstressed. Fig. 13 shows a surface of stressed stainless after contact with molten zinc at 1350 F for a very short time, on the order of minutes. These illustrations speak for themselves in indicating the necessity for keeping zinc away from contact with stainless steel at elevated temperatures, and help to show why cadmium-plated bolts rather than zinc-plated ones should be used in exhaust systems.



■ Fig. 12 – Same steel as that shown in Fig. 11, but several grains away from the surface, after 10 min in contact with zinc at 1950 F, in the unstressed condition



■ Fig. 13 – Surface of stressed stainless steel after contact with molten zinc at 1350 F for a few minutes

■ Additional References of Interest

1. *Vestnik Metallopromishlennosti,* Vol. 16, No. 17, 1936, pp. 73-86: "Investigation of Corrosion of Modern Valve Steels in Gaseous Media and at Increased Temperature," by V. Zinovyev and I. Levin.

2. *Archiv für das Eisenhüttenwesen,* Vol. 11, September, 1937, pp. 131-138: "Heat-Resisting Steels under Attack by Gases," by E. Houdremont and G. Bandel.

3. *Stahl und Eisen,* Vol. 61, 1941, pp. 852-860: "Present Status of Wrought Heat-Resisting Steels," by G. Riedrich. (Report No. 555 of Committee on Materials of VDE.)

Experience with the Use of Magnesium in Aircraft

continued from p. 66

panels. The preliminary conclusions reached may be stated as follows:

(1) Magnesium is not ignited by incendiary bullets.

(2) Solid shot does not affect magnesium structures much differently than it does aluminum. There is much less "tuliping" than with aluminum and the holes are less ragged.

(3) Explosive shells are more destructive to magnesium structures than to aluminum structures. Our observation has also been that solid shots into a large liquid container, such as a gasoline tank, produces an effect similar to an explosive shell and causes more damage to the magnesium surrounding it than to aluminum. Perhaps paradoxically, the British have had a very satisfactory use of magnesium as gas tanks in their Spitfire airplane. These tanks are constructed as a welded framework of extrusions with the sheet welded in, patch fashion, as small panels. Their observation is that these tanks are not only considerably lighter (12 gal more capacity and 9 lb less in weight) than the aluminum tank previously used, but are also more resistant to gunfire because the magnesium does not "tulip" and prevent the external bullet-proofing rubber from sealing effectively. Such tanks are readily repaired.

2. Could magnesium plate be substituted for aluminum as "deflector" plates?

a. Ballistic tests have shown that magnesium deflector plates will not replace aluminum with any savings in weight. On the average, magnesium will have a "ballistic limit" approximately 10% less than 24S-T aluminum plate of the same weight.

H. Primary Structures

1. How far has the development of magnesium primary structures progressed?

a. Several structures have been designed, built, and satisfactorily tested both statically and in vibration. In all cases a substantial weight savings was accomplished, particularly in control surfaces where appreciable weight is saved in balance weight. Only a few of these structures have reached the flying stage but those have been satisfactory so far. The longest service to date is a pair of wings in service over a year and a half and having about 500 hr of flying time. The British are using magnesium sheet as covering for control surfaces and over portions of the wing on some planes. They also have one model, of which about 100 were built in 1939, having the rear fuselage of monocoque magnesium design. A new model also incorporates a similar design.

The following tabulation gives the weight comparison of some of the magnesium structures which have passed static and vibration tests:

Type of Plane	Description of Part	Weight, lb	Saving as Compared to Present Construction, %
Advanced trainer	Outer wing	153.5	17
Advanced trainer	Aileron	16.0	28
Advanced trainer	Flaps	5.6	11
Pursuit	Stabilizer	34.0	24
Pursuit	Elevators	22.7	19
Pursuit	Tail assembly[4]	118.1	13

[4] Incorporates heavier gage than conventional to reduce the number of internal stiffeners.

OIL AERATION

by R. J. S. PIGOTT
Gulf Research and Development Co.

THE condition commonly termed "oil foaming" has been the cause of a considerable amount of trouble, both in airplanes flying at high altitudes and in ground vehicles. The term is not too well applied, as the visible evidence of trouble – foam escaping at breathers or vents – is a result and not a cause. A better term is "oil aeration," and it may now be said the cause is entirely mechanical. Much oil has gone over the dam (the designers, of course), and a quantity of somewhat simple-minded hunch-playing has got us nowhere. The real reason we have been in so much trouble from oil aeration is that until we flew high, or ran tank engines badly cramped for decent location of accessories, nobody recognized there was a trouble, and certainly almost nobody paid any attention to ordinary hydraulics in designing the oil supply system. Until the extreme conditions now experienced came in, almost any clumsily assembled tank, pipe, and pump would oil an engine pretty well; but not now.

The original cause of most aviation-engine oil aeration is the scavenge pump. It has generally been the custom to design pressure pumps on aviation engines 40 to 50% greater in capacity than the maximum demand of the engine; the excess normally is bypassed around the pump so that the full capacity of the pressure pump is never delivered to the engine sump, only the engine demand.

The scavenge pump is then usually designed 40 to 50% larger than the pressure pump. As a result, in level flying, the scavenge pump is 2 to 2.25 times the engine demand. Consequently, the scavenge pump must satisfy its desire to fill, with air and exhaust gas (blowby). This means that a large percentage of air is mixed with the oil and returned through the cooler to the tank. In the case of double-row aircooled engines, or any in-line engine, the situation is even worse, since there will then usually be duplicate scavenge pumps, each full size, to take care of clearing the crankcase either climbing or diving; the volume of air returned to the tank is then more than doubled.

Big bubbles would give little trouble; they bounce out easily. It is small bubbles that cause trouble, and unfortunately the scavenge pumps do a fine job of making little ones out of big ones. No matter how big the gulps of air are on the suction side, no bubble going through a gear

[This paper was presented at the SAE National Fuels and Lubricants Meeting, Tulsa, Okla., Nov. 4, 1943.]

AN extensive study of oil foaming has brought Mr. Pigott to the following conclusions:

1. All suction-side resistance of either pressure or sump pumps must be reduced as much as possible.

2. Velocities in suction piping should be kept below 5 fps at all points. Bends that cannot be eliminated should be long radius. Valves and other restrictions must be avoided.

3. Spur-gear pumps can be improved by optimum exposure of teeth on the suction side, use of side pockets, and reduction of clearance. Without help, these pumps still won't be good enough for the highest altitudes.

4. Oil tank pressurizing will provide boost for the pressure pump.

5. Centrifugal boosters are useful, except with cold oil or with much aeration. For the pressure pump, the booster should be located directly at the oil tank, and for the scavenge pump, at the sump. Both must be drowned, no lift.

6. One form of positive-displacement rotary is available for fixes or new designs that is almost as good as a first-class centrifugal in low suction-side losses, and satisfactory in thick oil or high aeration. Boosting is not needed.

7. Oil tank efficiency as air separators varies with oil level and with scavenge pump pressure. Bad tanks vary from 16% air to 3%; good tanks from 6% to 2%; as varied by tank level and pump pressure – 185 F oil.

■ ■ ■

THE AUTHOR: R. J. S. PIGOTT (M '18), chief engineer of Gulf Research & Development Co., has been engaged in work on heat distribution in internal combustion engines and the behavior of lubricating and fuel pump systems. He is also the originator of rotating intake screens for circulating water in power stations, the vacuum tripper for turbines, side wall ventilation for stokers and of rubber expansion joints for water and low pressure steam lines. In World War I, after installing the power and service station in the Remington Arms plant at Bridgeport, Mr. Pigott managed the Bridgeport Brass Co. for two years, and then the Crosby Steam Gage & Valve Co. for the same period. He was consulting engineer for Public Service Production Co. for several years. Mr. Pigott, who has about 30 patents covering equipment for power stations, oil burners, meters, safety valves, and so forth, is active in Coordinating Research Council work.

pump can be larger than a single tooth space, and since they are all mingled with oil in most violent agitation in the pump, they are not likely to be so big. Since most of the scavenge pumps will take from 25,000 to 30,000 bites a minute at the mixture, it is apparent that a very large number of small bubbles will be formed. Proof is available since several experimenters, ourselves included, have made studies on glass-front pumps, in which the cloud of fine bubbles can very readily be observed, and photographed.

The problem of bubble removal is controlled by Stokes law, if the flow is viscous; another formula applies for turbulent conditions.

Stokes law $$v = \frac{2gd^2}{36} \quad \frac{(\rho_1 - \rho_2)}{\mu}$$

Turbulent flow $$v = c\sqrt{d\,\frac{\rho_1 - \rho_2}{\rho_2}}$$

Where:

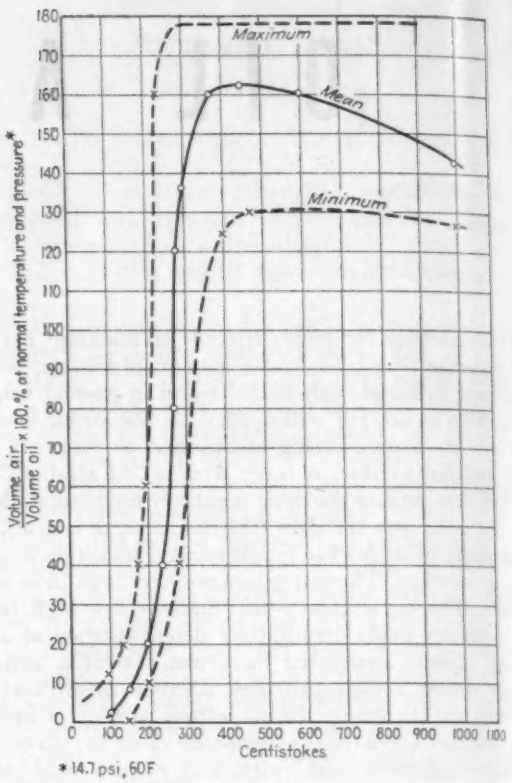
v = Slip velocity, fps
d = Diameter bubble, ft
ρ_1 = Air density, lb per cu ft
ρ_2 = Oil density, lb per cu ft
μ = Oil viscosity, fps units (Reyns = centipoise × 0.000672)
c = A constant varying from 9 for spheres to 3.5 for flattened bodies

It follows from either of these formulas that the smaller the bubble, the slower the rise. When starting and warming up, the oil in the tank is relatively cool, viscosity may be 40 or more times the value at full operating temperature of 180 or 185 F; it is apparent that bubble rise is greatly slowed down and deaeration is extremely difficult. Foaming is therefore most probable at starting or during a climb thereafter, particularly as the stability of foam is high when the viscosity is also high.

The stability of foam for the usual aviation oil is about as shown in Fig. 1. This curve has been substantiated by the British and by several laboratories in this country. At 60 F, a good aviation oil may be 1430 centistokes (1293 centipoises); at 180 F, 44 centistokes (40 centipoises). For cool oil, therefore, the rate of bubble rise is greatly reduced and the stability of foam formed at a free surface is high. The foam loss from engine breathers at low flying levels is usually traceable to a combination of poor tank design and cool oil. The effect, if from this cause, will usually be avoided with high-temperature oil. It is to be remembered that with cool oil, the engine resistance to flow is greatly increased so that flow reaching the scavenge pump is much reduced, hence more air than normal will be passed back to the tank.

The above foam-producing cycles are typical of any dry-sump engine. For wet-sump engines (ground vehicles), this particular trouble is unlikely, unless a very poor pump and piping sytem are involved.

The other principal cycle for producing foam occurs in transmission transfer cases and differentials, where the method of lubrication is by limited submergence of the running gears in a pool of lubricant. Running at any peripheral speed above rather low values, the teeth of the gears tend to cut a trough in the lubricant, as the oil cannot flow into the teeth fast enough; air is carried down in the teeth and beaten into the oil. This mixture breaks loose on the exit side of the gears as foam. Since the lubricant is, in general, more viscous than engine oil, is compounded, and the free surface for break-up of bubbles is small in any gear set, the chances of making foam are good. A number of Army vehicles have shown this trouble.



■ Fig. 1 – Variation of foam with viscosity for aviation oils

While straight mineral oils do not show any large variation of foam stability, compounded (or inhibited) oils, unless treated with an antifoam agent, tend to foam much more than straight oils. Details of these properties will be left to the chemist.

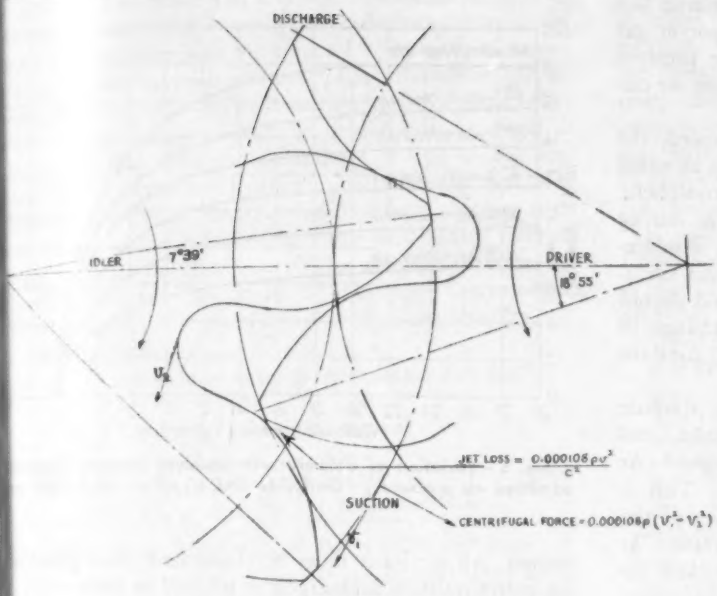
Since there is not much chance of mechanical improvement, unless we go over to dry sumping and spray lubrication in gear sets, this is one place where a good foam-suppressing agent can be very valuable as an immediate fix.

■ Lubricating Pumps

As stated, oil-foaming is a result: but *oil-aeration* is one of the major causes of pressure pump failure at high altitude. It is, therefore, of great importance to examine the characteristics of lubricating pumps, so as to be able to determine the normal internal losses in pumping liquid, and the additional losses due to the presence of air or any other compressible gases.

Referring to Fig. 2, it is at once apparent that the resistance to entry of liquid consists of two principal elements – the centrifugal force due to the ring of oil in the teeth of the gears, and the jet and turn losses entering between the partially meshed teeth, together with the smaller inlet port losses due to changes of direction and skin friction. Analogous losses would subsist in any other kind of rotary (such as a vane pump), if the oil is taken in at the periphery.

We discovered the magnitude of these losses in 1933 when testing a rotary pump design on gassy crude oil in East Texas. Cavitation, involving noise and vibration, was heavy, as approximately 25% gas by volume of oil was coming out of solution, and the discharge pressure was 600 psi. A simple centrifugal pump mounted on the same



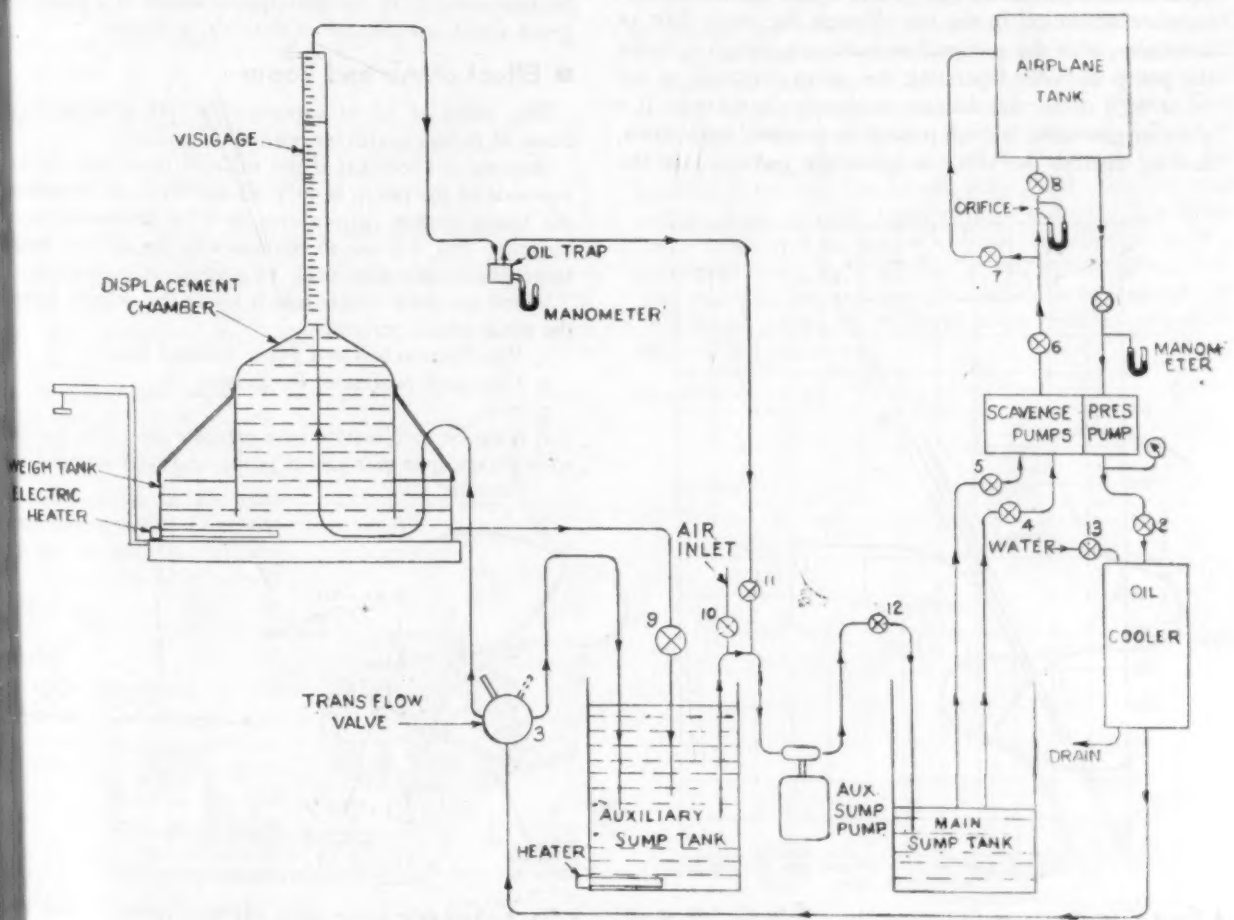
■ Fig. 2 – Spur-gear tooth loss

haft as the rotary and integral in the same housing solved his trouble completely.

It was not until early in 1942 that much further attention was given to such pump losses when we became interested in the subject for high-altitude aviation lubricating pumps, where such suction-side losses become of paramount importance. In April, a report was prepared giving the method of analysis of such pump losses, both with and without air present. The analytical predictions were quite closely checked by tests on a single spur-gear pump, all that we had at the time.

During the last 16 months we have tested about a dozen different pumps of several types, and have fully confirmed the methods developed in the report. As this report was treated as more or less restricted, it has not been published, but has been circulated among those entitled to restricted information.

In June, 1942, John Dolza of Allison presented a paper on the same general subject, and we were encouraged to find that he had arrived at much the same answers as we had. In the meantime we had developed two methods of measuring the suction-side loss directly, and we are therefore no longer dependent upon any kind of calculations or estimates to determine its value.



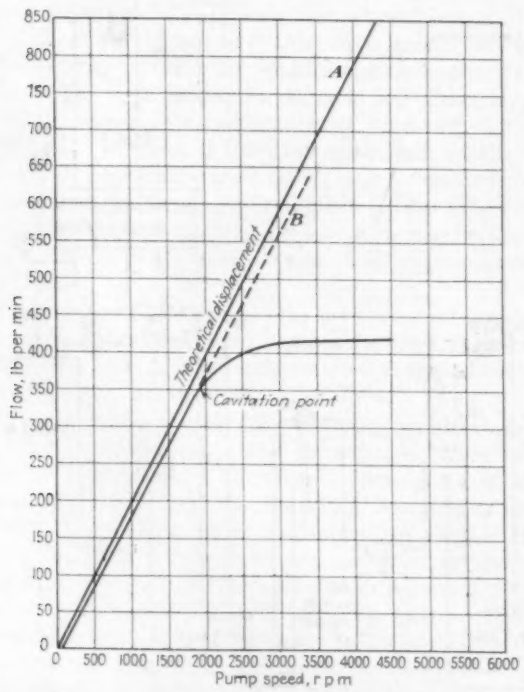
■ Fig. 3 – Laboratory test set-up for measuring entrained air in oil in airplane tanks

The total suction-side loss will consist of the pump loss proper, plus the partial pressures due to any vapor or gas present. If we use a liquid of very low vapor pressure which has been carefully freed of all dissolved gases, we can test for the pump loss alone.

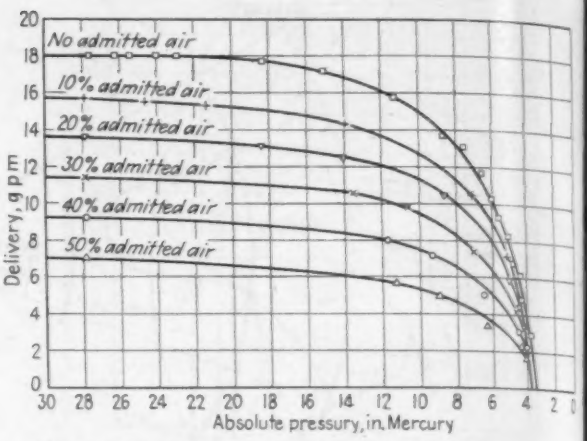
Two methods have been used, one varying speed, the other by restricting inlet. If the pump is set up as usual for delivery test, Fig. 3, with a suction tank at atmospheric pressure, and differential pressure is held constant during test, the delivery versus speed is parallel to the displacement line, since leakage or slip is a function almost exclusively of differential pressure/viscosity. The liquid should be held close to constant temperature to avoid change of viscosity and density, the differential pressure held constant by adjusting a valve on the discharge line.

Referring to Fig. 4, as speed is increased the absolute pressure at suction flange of the pump decreases, and pump suction-side loss increases as square of speed. At some speed these two values will become equal. This is the highest speed at which the volumetric capacity of the pump can be fully satisfied by the absolute pressure at suction, provided vapor pressure is insignificant, and no air is present. As speed is further increased, the pump losses for full flow are greater than the absolute pressure available for driving the liquid into the pump, and the delivery line falls away from the displacement line.

To get a satisfactory low vapor pressure it is only necessary to use an ordinary refined oil such as a lubricating oil SAE 10 to 60; to eliminate dissolved air, it is necessary to circulate all the oil in the test through the pump four or five times, with the suction line restricted enough to make the pump cavitate. Operating the pump cavitating at say 75 or 80% of normal delivery assures this condition. If a globe or gate valve is used, it must be mounted stem down, flowing against the disc, to avoid air leakage into the suction. All air leaks must be eliminated; shaft gland on the pump must be submerged in oil, and so forth.



■ Fig. 4 – Scavenge pump performance test – Gulfpride SAE 60 oil at 250 F, 25-psi differential pressure (constant)



■ Fig. 5 – Variation of delivery with absolute pressure (constant admitted air pressures) – Gulfpride SAE 60 oil at 180 F, 2600 rpm

The second method differs only in using constant speed, and restricting the inlet only, until drop-off shows cavitation. This method gives curves as shown in Fig. 5, which also shows the corresponding performance with measured amounts of air at inlet condition, admitted with the oil. This method is more convenient than the first, as a rule, because usually the complete performance of a pump at a given speed, say take-off or cruising, is desired.

■ Effect of Air and Vapor

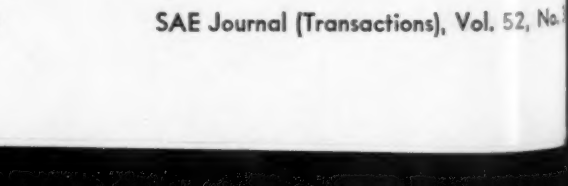
The effect of air or vapor on the pump is the major cause of failure to deliver normal oil quantity.

Anyone can see that if the mixture taken into the suction side of the pump is 90% oil and 10% air, by volume, the pump cannot deliver over 90% of its normal liquid capacity. But it is not so obvious why the delivery should in general be *less* than 90% of normal, as it always is.

There are three effects which lower the delivery beyond the air at intake percent:

1. Pipe friction loss and pump internal loss.
2. Clearance volume of the pump.
3. Slip.

It is easiest to illustrate by a concrete case. Fig. 6 shows an ordinary spur-gear power pump and tank system flying



■ Fig. 6 – Spur-gear power pump and tank system flying at altitude – SAE 60 oil at 180 F

at altitude. The pump has a clearance volume of 19% of the displacement; the latter is the difference between the tooth space volume and the trapped volume, or clearance volume shown between the two pairs of teeth at the instant of sealing off from both suction and discharge. (See Fig. 7.) This clearance volume obviously is returned to the suction side. The slip or leakage is 5% of displacement for the pressure difference and oil viscosity used. For this setup, 13.6% of air by volume at tank absolute pressure of 7.6 in. of mercury, the line and pump internal losses were 4.4 in. of mercury. The absolute pressure has therefore dropped to 3.20 in. of mercury. Consequently the air volume has increased to 27.1% of the mixture.

We have then:

1.000 Volume of tooth spaces taken as 1.000
0.271 Air volume
0.729 Tooth space liquid content

Slip in terms of tooth space volume:

$$0.050 \times \frac{1.00}{1.19} = 0.042 \text{ Slip back to suction}$$

Therefore $0.729 - 0.042 = 0.687$ Delivered to discharge

Meanwhile in passing from suction to discharge, the air has been recompressed as:

$$\frac{0.271 \times 3.2}{142.5} = .0061$$

The mixture in the clearance volume is therefore 0.994 oil and .006 air so that we have returned to the suction

$$0.19 \times \frac{1.00}{1.19} \times 0.994 = 0.159$$

fraction of tooth space.

Whence:

0.687 Delivered
0.159 Returned
0.528 Net delivery, fraction of tooth space

Converted to displacement:

$$\frac{0.528 \times 1.19}{1.00} = 0.628 \text{ Volumetric efficiency}$$

We therefore have under these conditions of 13.6% air at the tank, a reduction of volumetric efficiency from 95% to 63% instead of to 81.4%, which might have been expected.

At low or ground levels, the effects are not so noticeable, because the pipe and pump loss are not so large a percentage of tank absolute pressure. For example, at ground level the volumetric efficiency would be reduced from 95%, no air, to 78%, 13.6% air at tank.

The first effect of air present is to expand from the tank condition as measured, in proportion to the ratio of absolute pressure in the tank to absolute pressure in the *interior* of the pump. The pressure in the interior of the pump is always less than the tank pressure by the loss in the suction pipe from tank to pump, plus the internal suction-side loss of the pump. The mixture in the tooth spaces therefore contains more air by volume and less oil, than the proportions in the tank. The second effect is that the compression of the air on the discharge side fills the clearance volume with a much higher percentage of oil than was delivered by the tooth spaces under suction conditions. In fact, for the pressure pump amounts of air entrainment, the clearance volume returns nearly 100% oil.

It will be noted that since the centrifugal force is a first-power function of density, it will fall off as air is entrained or as the pump cavitates. Centrifugal force will also be proportional to square of rotative speed. The simplest form of the formula is:

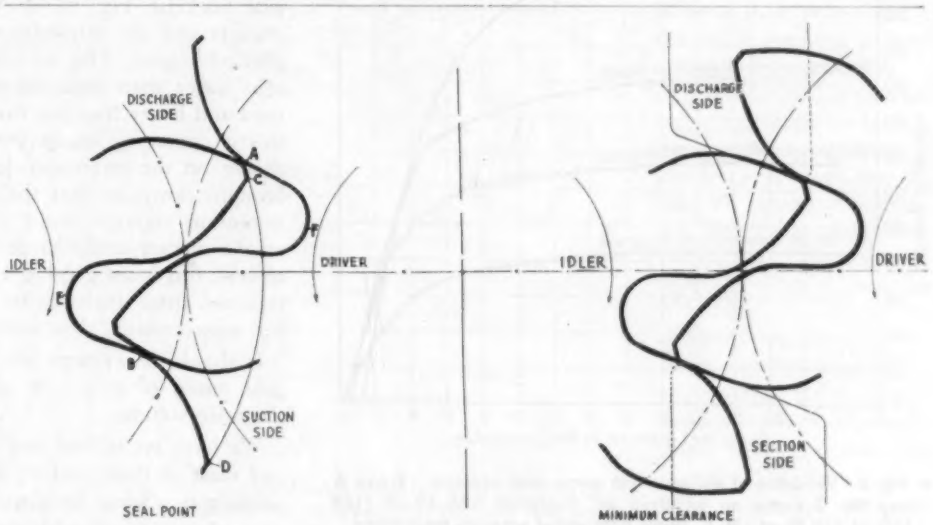
$$p = 0.000108\ \rho\ (v_1^2 - v_2^2)$$

where:

p = Centrifugal force, psi
ρ = Density of the *mixture* in the tooth spaces, lb per cu ft
v_1 = Peripheral velocity, tips of teeth, fps
v_2 = Peripheral velocity, roots of teeth, fps

The pump tooth loss consists of some skin friction, which in the usual viscous-flow condition subsisting, will vary as first power of velocity (quantity of oil, chiefly). But much more important are the turn losses and the reaccelerations into tooth spaces, between the teeth in the meshed region. These vary as square of velocity of fluid.

Another variable also shows up. The head required to fill the teeth varies greatly with position. In the meshed region the losses can be several times centrifugal force (for comparison) on account of the small area jets between teeth. In the open region, out of mesh, the tooth loss approaches the centrifugal force closely as a low value. As the pump goes into cavitation, the tooth loss drops off quite rapidly, as less and less of the filling is done in the meshed region, more and more in the open region. This condition which is difficult to calculate and still more difficult to segregate experimentally, accounts for the rounded corner of the fall-off curve, and then the long comparatively straight curve to zero delivery. The minimum value will be the centrifugal force alone, with a density of about 21% of that at full noncavitating flow. The reason is that slip, say 5%, and clearance volume, say 16%, must always be circulated in the teeth, at zero delivery and full pressure.

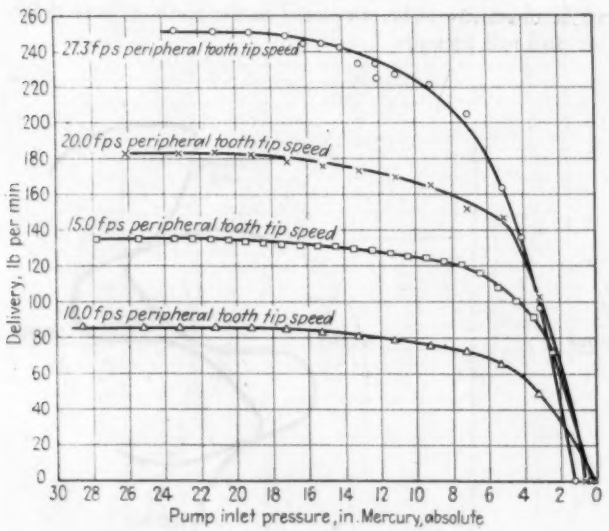


■ Fig. 7 – Spur-gear pump seal point and minimum-clearance diagrams

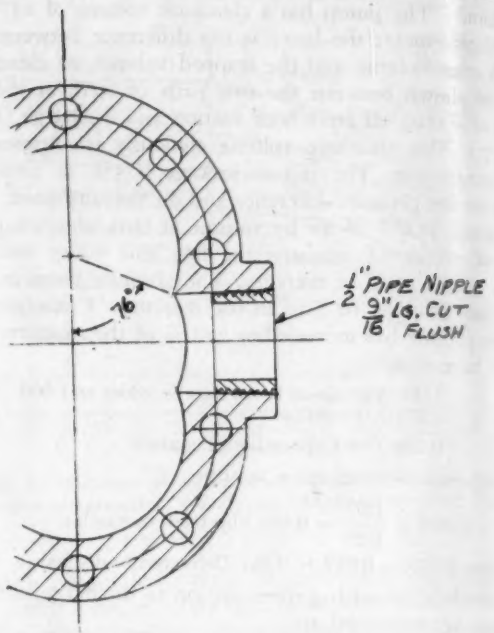
Fig. 8 shows the speed effect as well as the change of minimum head at refusal.

All of the foregoing test data and discussion are based on pumps alone with no relief valve operation, with which we are not directly concerned in a discussion of oil aeration. It is worth noting in passing that the relief valve behavior can give some weird anomalies in pump behavior, which are always blamed on the oil, the tank, or the pump. For example, the usual method of setting pressure-regulating or relief valves is to operate the pump at a specified speed, with a specified quantity delivered, and adjust the valve to the required pressure, obviously partly open and bypassing liquid. This method gives a standardized setting but no information as to the characteristic of the valve, or the seating pressure. If the valve has a stiff spring and the passages to and from the valve proper have high flow resistance, the valve will have a "steep" characteristic, or, as safety valve manufacturers call it, a large "accumulation." If a flat pressure characteristic is wanted, this is wholly wrong. We have had cases where such a valve set at 75 lb did not seat until the pressure had fallen to 38 lb, which occurred at about half the set speed: meaning that the relief valve was open almost down to idling speed. This condition would naturally give wide pressure variation with speed, not what the designer wanted. The valve was corrected by enlarging the inlet and outlet passages, and using a softer spring.

It should also be remembered that high resistance in inlet and outlet connections induces valve chatter, which is very bad for both valve and pump. Most people are unaware that seat contours have an important bearing on valve lift versus pressure. In many actual designs, this is not a simple straight-line relationship but has humps that yield irregular and puzzling behavior. It ought also to be remembered that any relief valve must operate over its whole range in rather small differences of pressure between relatively high pressures. It is therefore very easy to design a "sticky" valve, always troublesome.



■ Fig. 8 – Variation of delivery with pump inlet pressure – Brown & Sharp No. 2 pump, no admitted air, Gulfpride SAE 60 oil (120 SUS at 210 F) at 185 F, 70-psi differential pressure maintained



■ Fig. 9 – Brown & Sharp No. 2 spur-gear pump – condition No. 1

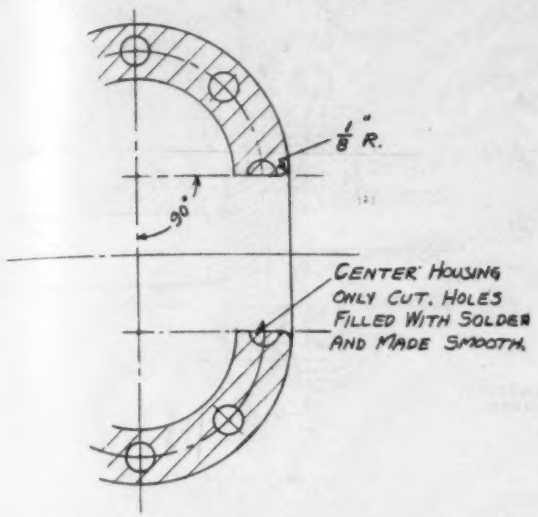
■ Spur-Gear Pump

The internal losses of the spur-gear pump are noticeably affected by the design. We have found that some pumps are designed with the housing close-fitted around the gears too far on the suction side. Some pumps are built with the housing fitting the gears around to reach the diameter of the inlet pipe. (See Fig. 9.) This is definitely too little opening, as shown by tests for cavitation point. In general, the housing should not be close-fitted to less than 90 deg from the centerline of the gears (See Fig. 10.); above about 135 deg the pump begins to lose out again on account of increased slip. It is obvious that as the housing is cut back as in Fig. 10, the capillary seal length has been cut back over the tips of the teeth, and down the sides of the gears.

Trapping, as well as cavitation, are both aggravated in those pumps (and there are plenty) designed without any side pockets. Fig. 7 shows the seal point without side pockets and the minimum-clearance volume point, for a particular gear. The volume at seal point is several per cent larger than the minimum when no side pockets are used and heavy trapping may result with close-fitted gears; this is damaging to the pump bearings. Side pockets as shown on the minimum-clearance diagram need only be brought down so that the seal point coincides with the minimum volume. Some designers put the pockets only on the suction or discharge side, clear down to the line of centers; this is not good as it increases the clearance volume returned from discharge to suction, and shortens the side seal unnecessarily, thus increasing slip.

It should be evident from the diagrams that the spur-gear pump is, as a type, difficult to lay out with small clearance volume.

We have yet to find one with less than 14% clearance, and most of them run 17 to 19%. This is obviously unsatisfactory. Some betterment can be attained by using square-bottom teeth with small fillets, in place of the usual semicircular finish, but in any case the clearance cannot be



■ Fig. 10 – Brown & Sharp No. 2 spur-gear pump – condition No. 1A

reduced much below 10%, on account of the fact that the teeth do not fill the tooth space.

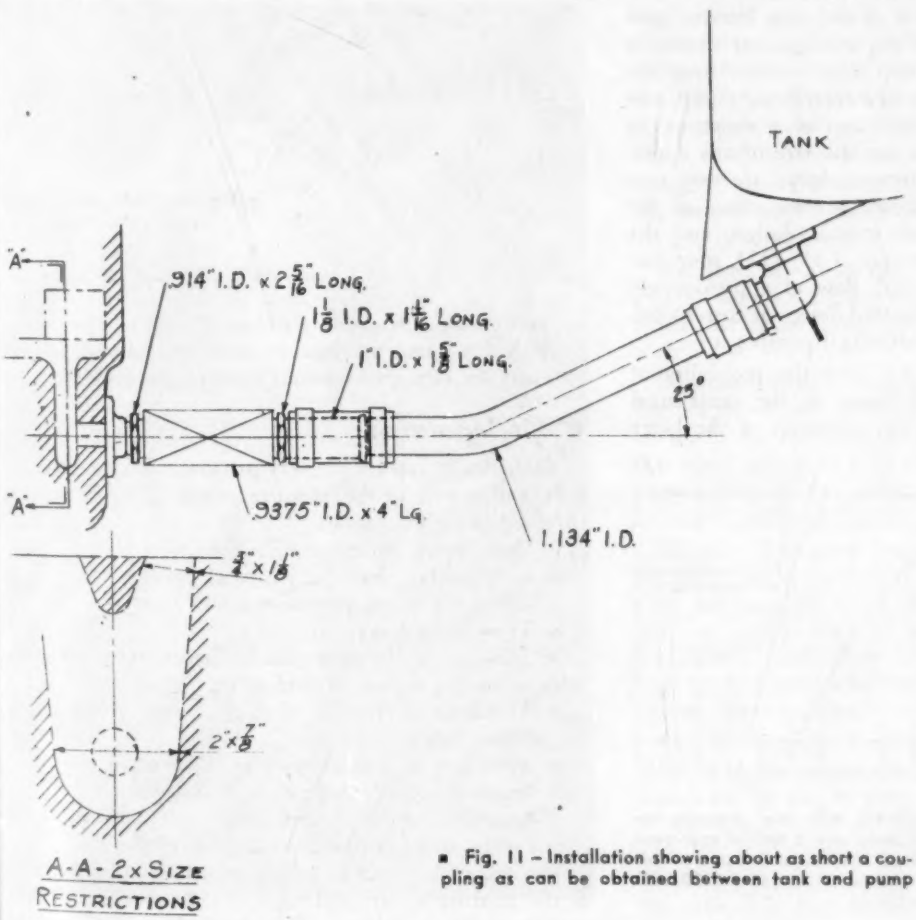
The piping from tank to pump must be carefully reconsidered. It should be as short as possible, as few bends as may be, and with no restrictions. Referring to Fig. 11, which shows about as short a coupling as can be obtained, with 180-deg aviation oil, the total piping loss is 2.19 in. of mercury, at 120 lb per min flow. Starting up with 60-deg oil, Japs overhead, the loss is 18.61 in. In the normal operation case, 0.43 in. is the pipe friction, and sharp bend at the tank plus the resistance due to restrictions at the drain valve and couplings is 1.77 in. of mercury, or about four times the pipe friction. Moral, use more radius on the bend and cut out the fool restrictions. We have found quick disconnect valves in such suction lines that cause a loss of 2 in. of mercury or more; sometimes antiseep valves are used, resistance 5 or 6 in. These things *cannot be permitted;* we have to recognize that the suction-side resistances on any pump must be kept as *low as possible,* if successful high-altitude flying is to be attained securely.

■ Spur-Gear Limitations

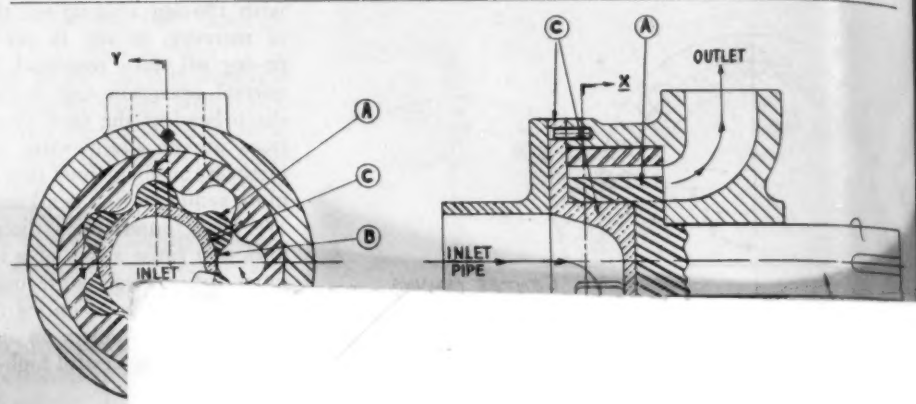
As any compressor designer knows, for best handling of gases, the clearance volume of the pump must be reduced as much as possible, but unfortunately the ordinary spur-gear pump has rather serious limitations in this respect. Other designs of pump are available that can be built with low clearance, but in the middle of this war, such a solution is not generally desirable; it involves too drastic a change. What we often need is a "fix" to take care of present production and engines already operating.

Pressurizing the oil tank is one simple fix. A few pounds pressure on the tank can be very simply provided by putting a spring-loaded relief valve in the vent line back to the engine. The pressurizing is obviously a substitute for a higher barometric pressure, but the objection usually offered is that the leakage will be high if the tank is shot up. Perfectly true, but in an unprotected tank the leakage will not be zero even with no pressure on the tank. In a rubber-lined tank, pressurizing might still be feasible, if the lining can be made to stand 2 or 3 lb pressure above the static pressure it now withstands successfully.

If pressurizing is not considered desirable, a booster pump can be employed. It is a fact that up to now the properly designed centrifugal pump has the lowest suction-side loss of any type. If one is used for boosting it should be mounted drowned at the bottom of the tank; if it is mounted on the engine, it cannot correct for the pipe losses between tank and booster. The drawback of the centrifugal pump is that it fails much more sharply than a positive-



■ Fig. 11 – Installation showing about as short a coupling as can be obtained between tank and pump



■ Fig. 12 – Diagram of internal-ported pump operation

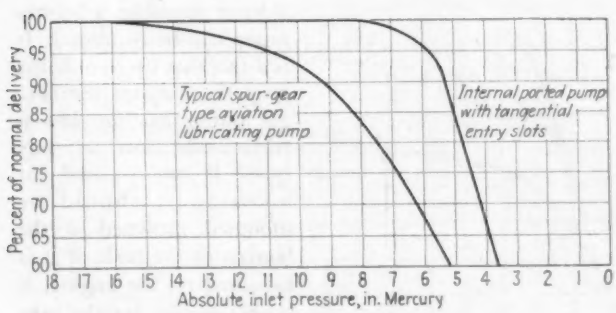
displacement pump when air is present. Further, it is very little good on oils more viscous than 3000 SUS, no good whatever at 5000 SUS. These viscosities might easily be obtained in a hurried cold start. But fortunately, that start will be with full barometer, and the rotary can do the job alone. Mounted drowned in the engine sump for assisting the scavenge pump, the centrifugal pump has no value as long pump capacity is great enough, unassiste oil, and is therefore handling plenty of air

But when the rotary begins to fail, less the centrifugal pump begins to develop the time the sump level has risen enough inlet completely, most of the air is elir centrifugal does a full job. It, therefor sound bet, provided the centrifugal is pro to pick-up edge of the blades.

In the course of development of our o pump, Fred Fleischer devised the arrang Fig. 12. In this case, the pump takes s center of the shaft outward, as in a centri therefore eliminates centrifugal force as entry. The ports into the teeth are also fi ciable restriction, so that the internal los siderably reduced. Fig. 13 shows a co losses in a spur-gear pump of normal internal-ported design. At 2700 rpm, a 1½ pump has an internal loss at full flow of approximately 10 in. of mercury; the internal-ported design at corresponding conditions, 6 in.; a good centrifugal pump 4 in.

With this type of design, we have the possibility of approaching the low internal losses of the centrifugal pump while avoiding the serious weakness of the latter on aerated oil, and inability to handle cold oil.

With this pump no booster would be required, and the necessity for two drives would therefore be avoided.



■ Fig. 13 – Variation of normal delivery with inlet pressure for 1.389 x 1.285-in. P. L. internal-ported pump and a typical spur-gear pump – 2600-rpm pump speed, 100-psi discharge pressure maintained, no air admitted, Gulfpride SAE 60 oil at 185 F (120 SUS viscosity at 210 F)

■ Air Separation

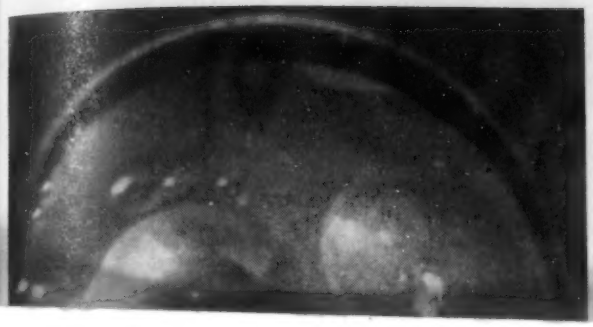
Tank design can have a very marked effect on air retention, and return to the pressure pump. Four factors are involved in air separation:

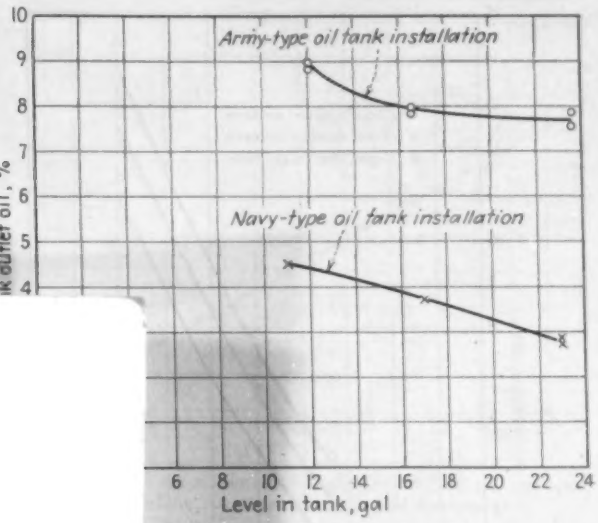
1. Short travel, for escape of bubbles.
2. Large surface area, for escape of bubbles.
3. Low viscosity, to increase speed of rise.
4. Time for rise to occur.

In addition to the above basic factors, there are three other secondary factors dependent on design:

1. Avoidance of free fall of oil into tank, to avoid driving bubbles down.
2. Avoidance of counterflow conditions in air and oil, to prevent remixing or obstruction of bubble rise.
3. Avoidance of short-circuit, low-resistance paths between point of air introduction and oil-offtake.

The scavenge pump normally returns from 40 to 70% of the mixture as air, and most of this bounces out very readily as large bubbles. The removal of 80% of the





ation of air in tank outlet oil with level in tank – SUS at 210 F) at 185 F, oil flow rate: 120 lb per d: 2700 rpm, scavenge pump discharge pressure: 15 psi

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r idea is very popular, for quick warm-up, in which the oil from the scavenge pump is he top of the hopper is a very poor air violates three of the basic principles and one design principles; namely, long bubble travel, area, short time; free fall of oil (usually), and if the warm-up bypass or the main return e bottom of the hopper, there is a short-circuit o the pressure pump.

ows one of these bad connections; and Fig. 15 s happening. It could hardly be worse.

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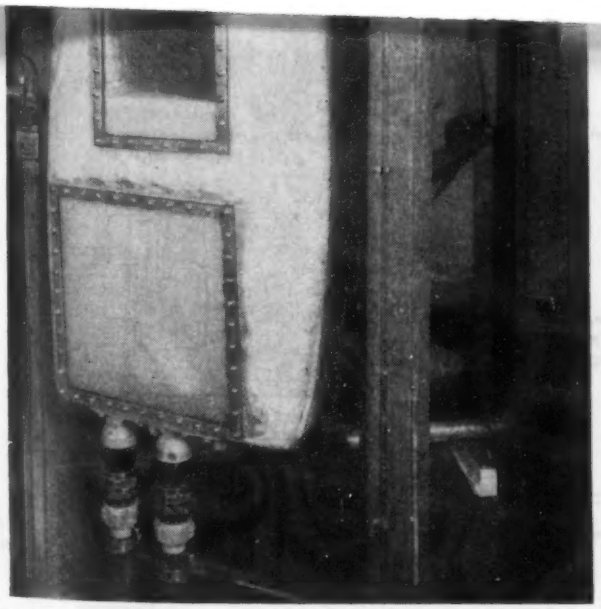
of oil in the tank affects the entrained air. vs the effect of oil level in two designs. The upper one is an Army-type tank, the lower a Navy-type tank.

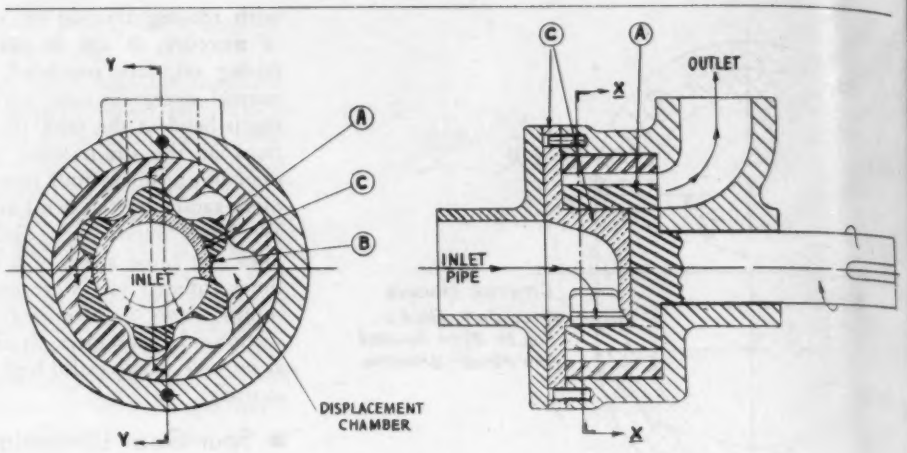
Pressure on the scavenge pump also affects air release. Fig. 18 shows the effect on air to pressure pump at three oil levels, for the tank shown in Fig. 16. The reason for this effect is that as pressure is increased, the pump compresses more of the bubbles to small size, increasing the amount of difficult bubbles to separate.

We have done some work on improvement of tanks, but not much of the material is in shape to present at this time. One design, with helical flow to increase area, decrease path for bubble rise, and prevent splash at any level, has been given preliminary tests. The results are shown in Fig. 19. Evidently some definite improvements are possible in tank air separation.

One arrangement with possible merit used in the Focke-Wulf is to place the coolers after the tank, with a booster pump, on the way to the pressure pump. This would give a tank temperature of, say, 250 F, instead of 180-185 F, and the consequent reduction of viscosity. This should allow bubbles of a given size to rise faster. However, as the tank flow may be relatively turbulent, the gain may be small, or nothing.



■ Fig. 16 – Airplane oil tank – front view



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But when the rotary begins to fail, less air is inducted, the centrifugal pump begins to develop pressure, and by the time the sump level has risen enough to submerge the inlet completely, most of the air is eliminated and the centrifugal does a full job. It, therefore, is a perfectly sound bet, provided the centrifugal is properly designed as to pick-up edge of the blades.

In the course of development of our own internal gear pump, Fred Fleischer devised the arrangement shown in Fig. 12. In this case, the pump takes suction from the center of the shaft outward, as in a centrifugal pump, and therefore eliminates centrifugal force as a resistance to entry. The ports into the teeth are also free of any appreciable restriction, so that the internal losses are very considerably reduced. Fig. 13 shows a comparison of the losses in a spur-gear pump of normal design, and the internal-ported design. At 2700 rpm, a 1½ O.D. spur-gear pump has an internal loss at full flow of approximately 10 in. of mercury; the internal-ported design at corresponding conditions, 6 in.; a good centrifugal pump 4 in.

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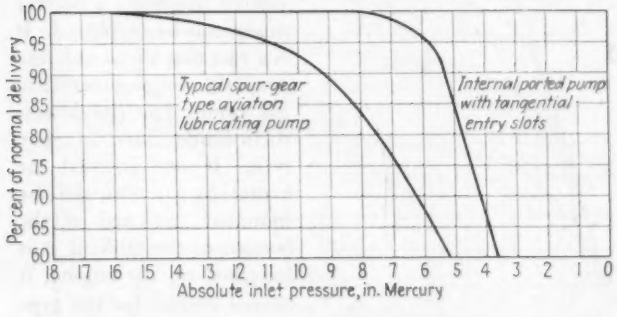

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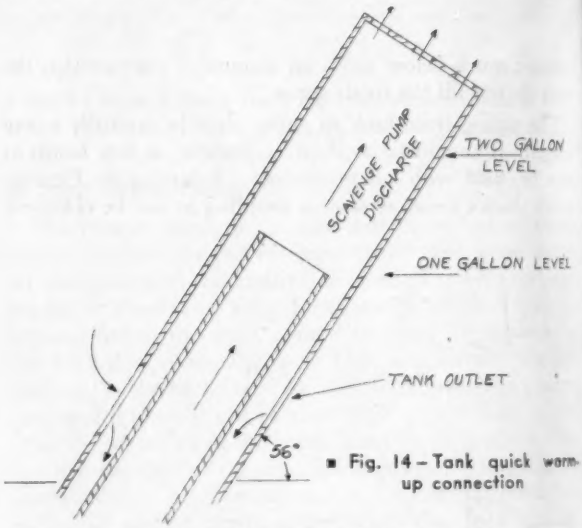

■ Fig. 14 – Tank quick warm-up connection

on aerated oil, and inability to handle cold oil.

With this pump no booster would be required, and the necessity for two drives would therefore be avoided.

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1. Short travel, for escape of bubbles.
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In addition to the above basic factors, there are three other secondary factors dependent on design:

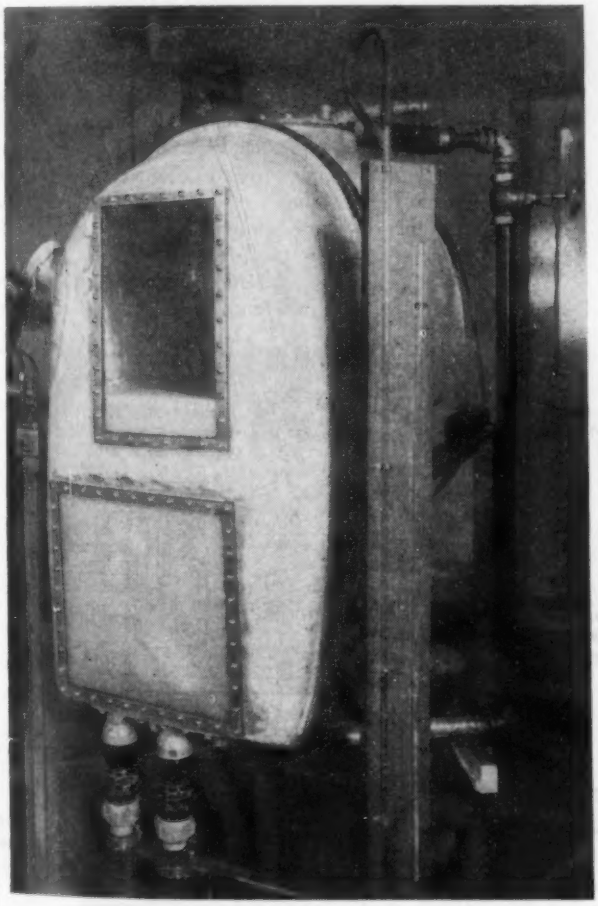
1. Avoidance of free fall of oil into tank, to avoid driving bubbles down.
2. Avoidance of counterflow conditions in air and oil, to prevent remixing or obstruction of bubble rise.
3. Avoidance of short-circuit, low-resistance paths between point of air introduction and oil-offtake.

The scavenge pump normally returns from 40 to 70% of the mixture as air, and most of this bounces out very readily as large bubbles. The removal of 80% of the

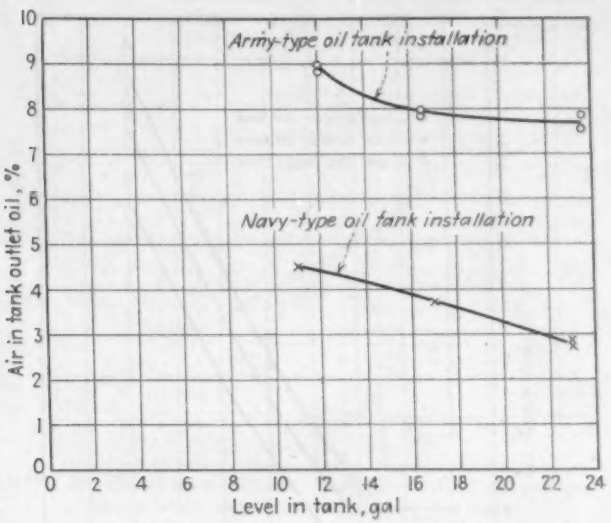
■ Fig. 15 – View looking into top of tank

returned air is easy; it is the remaining 20%, which may be in the form of *small* bubbles, that is hard to get out.

Examination of present tank designs meeting Army and Navy specifications shows some rather sour examples.



■ Fig. 16 – Airplane oil tank – front view



■ Fig. 17 – Variation of air in tank outlet oil with level in tank – SAE 60 oil (120 SUS at 210 F) at 185 F, oil flow rate: 120 lb per min, pump speed: 2700 rpm, scavenge pump discharge pressure: 15 psi

The hopper idea is very popular, for quick warm-up, but that type in which the oil from the scavenge pump is returned to the top of the hopper is a very poor air separator. It violates three of the basic principles and one or two of the design principles; namely, long bubble travel, small release area, short time; free fall of oil (usually), counterflow; and if the warm-up bypass or the main return opens into the bottom of the hopper, there is a short-circuit of air direct to the pressure pump.

Fig. 14 shows one of these bad connections; and Fig. 15 shows what is happening. It could hardly be worse.

Fig. 16 shows another type of tank with a free fall of oil; the pea soup at the bottom, where oil has access to the hopper, is approximately 6 or 7%.

The level of oil in the tank affects the entrained air. Fig. 17 shows the effect of oil level in two designs. The upper line is an Army-type tank, the lower a Navy-type tank.

Pressure on the scavenge pump also affects air release. Fig. 18 shows the effect on air to pressure pump at three oil levels, for the tank shown in Fig. 16. The reason for this effect is that as pressure is increased, the pump compresses more of the bubbles to small size, increasing the amount of difficult bubbles to separate.

We have done some work on improvement of tanks, but not much of the material is in shape to present at this time. One design, with helical flow to increase area, decrease path for bubble rise, and prevent splash at any level, has been given preliminary tests. The results are shown in Fig. 19. Evidently some definite improvements are possible in tank air separation.

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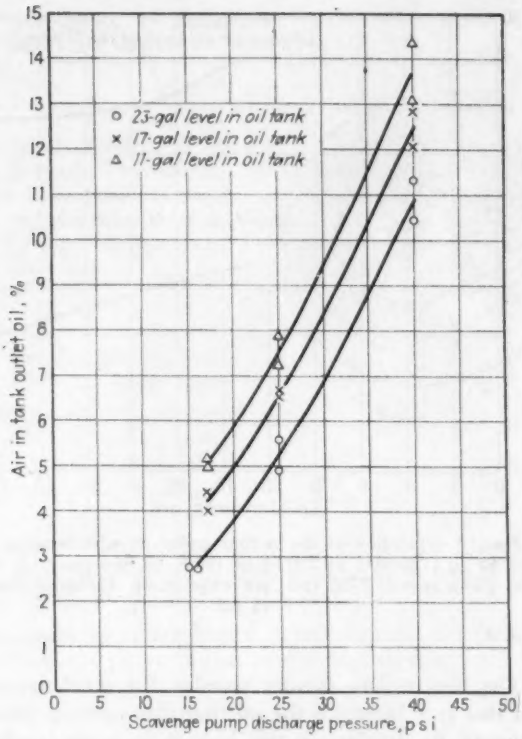

23-gal level in oil tank
17-gal level in oil tank
11-gal level in oil tank
Air in tank outlet oil, %
Scavenge pump discharge pressure, p s i

■ **Fig. 18 – Variation of air in tank outlet oil with scavenge pump discharge pressure – airplane oil tank with U. A. P. coolers Nos. U-6011 S-M1 and U-6011-M2, 100% oil flow into top of tank, SAE 60 oil at 185 F, oil flow rate: 120 lb per min, 2700-rpm pump speed, scavenge pump suction: 3 in. of mercury, connecting lines 1¼ in. O.D. x 0.047 in. tubing, back pressure applied to scavenge pump by clamping hose installed in scavenge pump discharge line at inlet to oil tank**

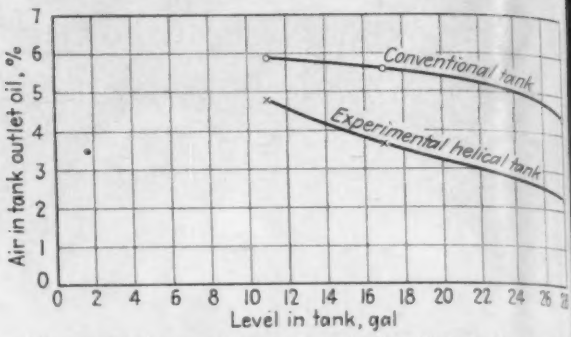
In viscous flow, Stokes' law applies, as discussed earlier in this paper.

Here viscosity is controlling. But in turbulent flow, the turbulent-flow formula applies.

It will be seen that viscosity does not appear in the turbulent-flow formula. If the flow is viscous, the change from 180 to 250 F. would improve air release: if in turbulent flow, no difference.

■ Other Separators

Other devices, separate from the tank, have been tried with varying success. Centrifugal devices such as the Sharples separator and the Heenan coil show a removal of 90 to 95% of air present under some circumstances. But the final answer to these devices is to test with the separator connected to the tank and see if there is any improvement. We have so far not found any, provided the tank was itself of good design. No doubt such devices will improve the performance of a bad tank, but why not make the tank right first, and then add the gadget if it can make *further* improvement?



■ **Fig. 19 – Tank comparison test – Gulfpride SAE 10 oil at 101 F, oil flow rate: 120 lb. per min, 20% of oil flow into bottom of tank, 80% into top of tank, 2700-rpm pump speed, barometer: 29.2 in. of mercury, absolute**

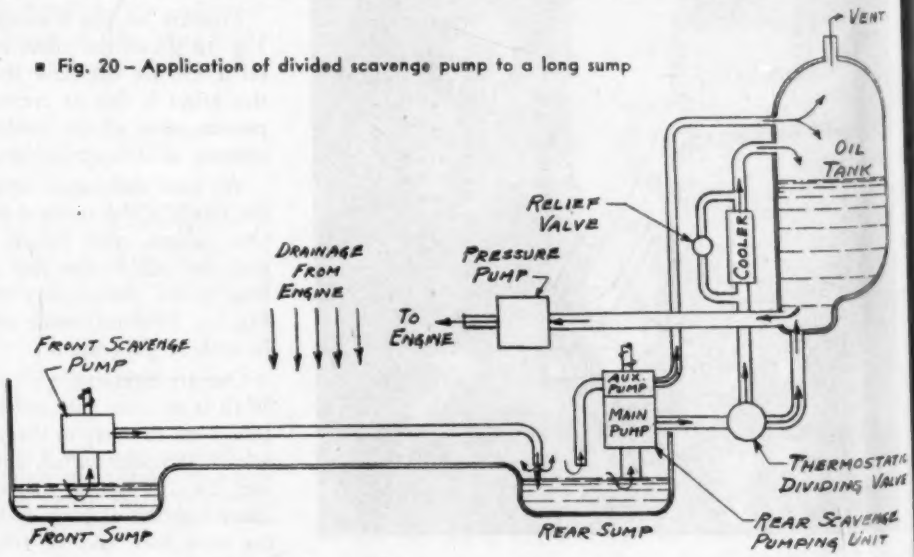
■ Divided Scavenge Pump

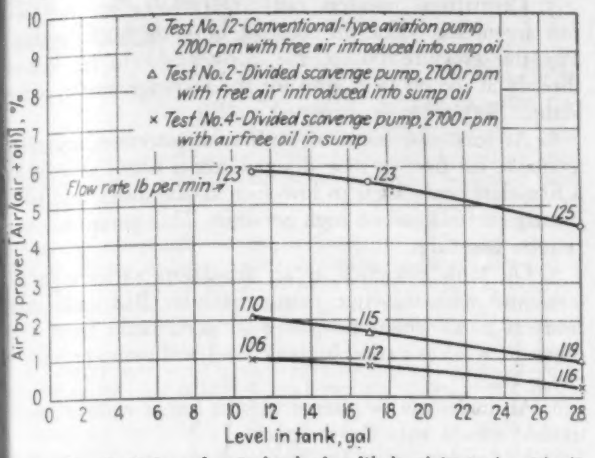
Another way of getting good air separation is not to put it in the oil – prevention rather than cure. B. R. Walsh came up with the following scheme: Divide the scavenge pump into two sections, totaling the normal capacity. One section is made the same capacity as the engine flow, and delivers as now, through the oil cooler. The remaining section (the excess capacity) as shown in Fig. 20, has its suction placed a little above the normal sump level, and ordinarily gets only air, which is delivered to the top of the tank by separate pipe, bypassing the cooler.

With this scheme, the only air going through the main scavenge pump and cooler will be that *entrained* in the sump oil, a matter of perhaps 6 or 8% with normal 240 F oil. The rest of the excess capacity of the scavenge pump is delivering unmixed air to the top of the tank, where it does no harm. Fig. 21 shows test results of this arrangement. The reduction in aeration to the pressure pump is of the order of from 6 to 2%.

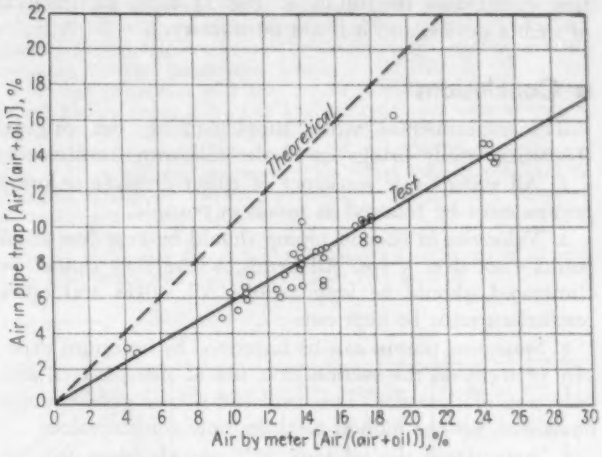
The immediate objection offered is: it adds a pipe to be shot up; for in-line engines with fore and aft scavenge.

■ **Fig. 20 – Application of divided scavenge pump to a long sump**




Test No. 12-Conventional-type aviation pump 2700 r.p.m. with free air introduced into sump oil
Test No. 2-Divided scavenge pump, 2700 r.p.m. with free air introduced into sump oil
Test No. 4-Divided scavenge pump, 2700 r.p.m. with air free oil in sump
Flow rate lb per min
123
123
125
110
115
119
106
112
116
Air by prover [Air/(air + oil)], %
Level in tank, gal

■ Fig. 21 – Variation of entrained air with level in tank, with divided scavenge pump – 20% oil flow in bottom of tank, 80% oil flow in top of tank, Gulfpride SAE 10 oil at 101 F, per cent air based on 29.2 in. of mercury, barometer

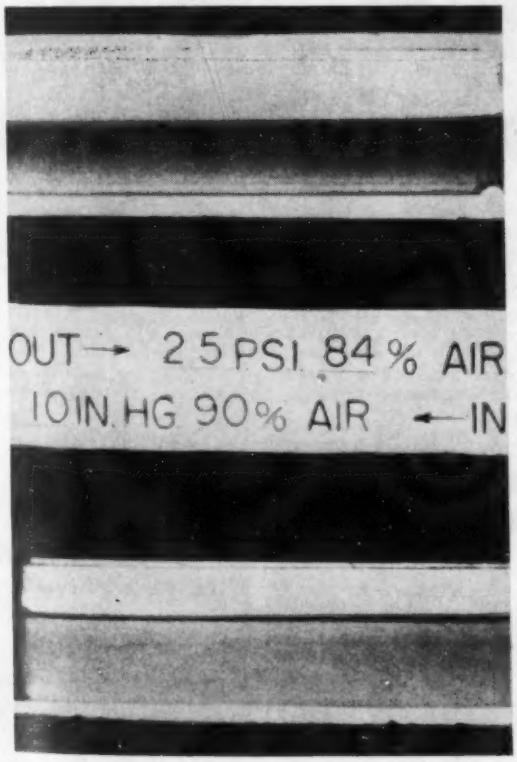


■ Fig. 23 – Air-oil measurements – Gulfpride SAE 10 oil, per cent air based on room temperature and barometer

two pipes. Would you rather avoid the risk of one extra pipe to four or five already there, or do you want to fly the plane higher? The second extra pipe objection for the in-line engine is not valid. The fore pump can discharge to the aft sump, single pipe as now, and the bulk of the air will free in this sump, when diving; then the divided scavenge pump is only employed at the rear and gets oil both in climb and dive.

■ Measuring Air in System

One thing needed to convince the designers and operators that they have been faced with an aeration problem



■ Fig. 22 – View showing the relative amount of air and oil in sections of the inlet and outlet pipe of a rotary gear pump

ahead of the pumps is some means of measuring the actual amount of air going to the pumps. We tried selenium cells for a measure of opacity; no good. Then we tried electrical condensers to measure change of dielectric strength; no good, upset by nearly everything from segregation to water. Finally, B. R. Walsh and G. S. Peterson concluded that the best thing to show how much air is present in a system is a pump. All our tests on actual airplane pumps were demonstrating that the delivery of any positive displacement pump is uniquely determined by the absolute inlet pressure and the percentage of air present in the suction. Therefore it is only necessary to draw a reliable sample of the mixture, pass it through such a pump, and read suction and discharge pressure, while operating at a known speed and delivering through a fixed resistance, such as a disc orifice. Change of viscosity such as occurs in normal operation has insignificant effect, and speed at all loads is held to a slight consistent variation merely by holding voltage constant. The actual unit has been flown successfully, is accurate to ±½ % air, and weighs 5½ lb.

There are really two problems in one; measuring the percentage of air, and getting a representative sample. The last part of this double problem is really a tough one. Many experimenters have tried sampling at points in the suction line; most of them got wrong results, because of the fact that when in a pipe, air segregates more and more as the mixture flows in the pipe, until it is finally largely traveling in a section of the pipe, mostly distinct from the oil, and at a considerably higher velocity. The reason is that any mixture of air in oil is unstable, and indeed it is very difficult to hold more than 10% distributed in 185 F oil for any length of time.

We have carried out many tests using transparent pipe, and Fig. 22 shows a typical condition. The lower pipe shows about 45% of the area flowing air, and of course a small percentage still in suspension in the oil below. The actual volume of air is 90% of the mixture when measured in static condition. The air is traveling about 2¼ times the speed of the oil, since it tends to adjust the airflow loss to the same value as the oil-flow loss; it can only adjust by shrinking the area for flow and raising the speed.

A pipe trap type of measuring device such as used by some investigators would show about 50 to 60% air, whereas a sample taken at the tank outlet before segrega-

tion would show the full 90%. Fig. 23 shows a calibration of such a device, and it is not satisfactory.

■ Conclusions

The experimental work supplementing our original theoretical study brings us to the following conclusions:

1. All suction-side resistance of either pressure or sump pumps must be reduced as much as possible.

2. Velocities in suction piping should be kept low at *all* points – not over 5 fps. All bends, where they cannot be eliminated, should be long radius. All valves and other restrictions must be kept out.

3. Spur-gear pumps can be improved by optimum exposure of teeth on the suction side, use of side pockets, and reduction of clearance. They will still not be good enough unassisted, for the highest altitudes now contemplated.

4. Pressurizing the oil tank will provide boost for the pressure pump.

5. Centrifugal boosters can largely solve the problem, but are of less value on cold oil, or with much aeration. For the pressure pump, the booster should be located directly at the oil tank, and for the scavenge pump, at the sump. Both must be drowned, no lift.

6. At least one form of positive-displacement rotary is available for fixes or new designs that is nearly as good as a first-class centrifugal in low suction-side losses, and satisfactory in thick oil or high aeration. This pump does not require boosting.

7. Oil tank efficiency as air separators varies with oil level and with scavenge pump pressure. Bad tanks vary from 16% air down to 3 or 4%; good tanks from 6% down to 2%; as varied by tank level and pump pressure – 185 F oil.

8. All tanks can be altered to best design without much trouble, except with the designers.

9. An instrument of satisfactory accuracy is available for proving how much air is in circulation.

DISCUSSION

Low Flow Resistance Needed in Pump Inlet

– C. H. van Hartesveldt

Ranger Aircraft Engines

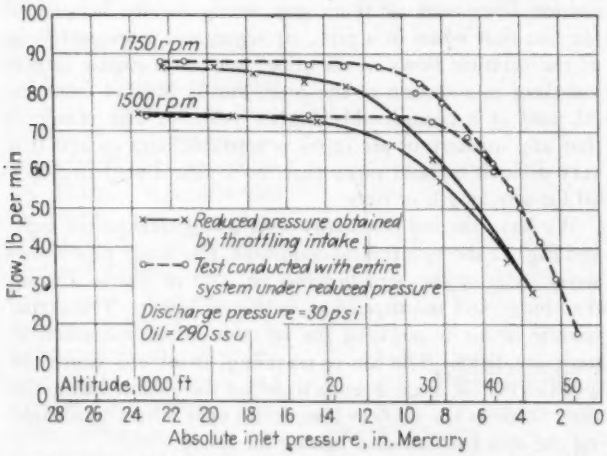
IN one instance our experience in testing oil pumps does not agree with Mr. Pigott's. We were unable to remove dissolved air sufficiently by the method of circulating oil with the pump inlet restricted in order to cause pump cavitation. In spite of taking all the precautions we could think of, a check on our data wherein we produced low pump inlet pressure by evacuating the whole system showed improved pump performance over that obtained by throttling the pump inlet with a valve. By the evacuation method we measured substantial increases in pump output as compared to the throttling method, particularly at the lower inlet pressures in which we were the most interested. These data are shown in Fig. A.

It may well be that our system, which employed a weigh-tank, provided greater opportunity for the oil to resaturate with air than in Mr. Pigott's system.

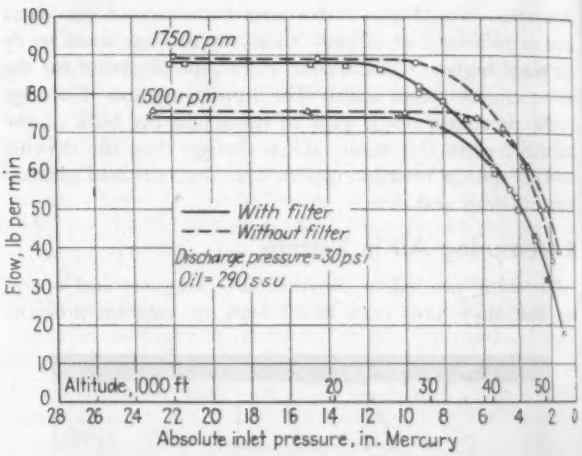
Our pump testing under simulated altitude conditions certainly has emphasized to us the requirement for pump inlets with low resistance to flow. On one of our scavenge pump assemblies we found that merely removing a screen at the inlet increased the flow to the pump by 30% at a simulated altitude of 45,000 ft. These results are shown in Fig. B.

I would also like to add what emphasis I can to Mr. Pigott's remarks on the need for side pockets in pumps to relieve trapping. At one time we had an epidemic of oil pump drive-shaft failures in service which was completely eliminated by milling in the proper pocket, an operation which was readily performed at service activities. At the time we put the change into effect we convinced all skeptics of its necessity by having them immerse one pump with and one without side pockets in a bucket of oil and turn them by hand.

Along with Mr. Pigott we have found engine oil-flow difficulties subject to all the flow relationships published for the most part in chemical engineering literature. This approach to our problems has proved a sound one.


1750 rpm
1500 rpm
Reduced pressure obtained by throttling intake
Test conducted with entire system under reduced pressure
Discharge pressure = 30 psi
Oil = 290 ssu
Flow, lb per min
Altitude, 1000 ft
Absolute inlet pressure, in. Mercury

■ Fig. A – Bench test of scavenge pump – comparison of performance obtained when low inlet pressure produced by the evacuation method and by throttling the intake



■ Fig. B – Bench test of scavenge pump showing the effect of removing a filter – test conducted with entire system under reduced pressure

Tests Needed to Show Effect of Diluted Oil

– E. A. Ryder

Pratt & Whitney Aircraft

ONE matter, not touched upon by Mr. Pigott, is the handling of diluted oil. The general practice of introducing gasoline in the oil before stopping the engine in cold weather, brings in some new problems. The disposal of the gasoline vapor when the engine is restarted may cause quite high velocities to exist in breather passages. This may need some attention from engine designers. In the meantime, it might be desirable for Mr. Pigott to make some additional tests, if he has not already done so, using diluted oil to determine the performance characteristics of pumps and oil systems during the transient conditions, while the gasoline is being disposed of.

Develops Test to Determine Foaming Tendencies of Oils

– H. L. Moir
The Pure Oil Co.

WE think that perhaps Mr. Pigott has bent over backwards a little in taking the onus from the oil men as regards foam trouble since I do know that, in spite of what would commonly be termed good engineering, oils will foam.

Recognizing this, our laboratory has developed a method for determining motor oil foaming tendencies in a gasoline engine. This test consists of incorporating a window in the crankcase of a Chevrolet engine where the actual foam formation can be observed during and after running. In order to test the foaming tendencies of a motor oil under actual operating conditions, the following procedure was developed.

A standard Chevrolet engine is equipped with a rectangular window on the port side of the oil pan between cylinders Nos. 4 and 5. The window is calibrated by placing one quart of oil at a time in the crankcase and marking the oil level on the side of the window. Through this window the oil level and foam level, if any, can readily be seen, as shown in Figs. C and D.

The engine is then run under the following conditions:

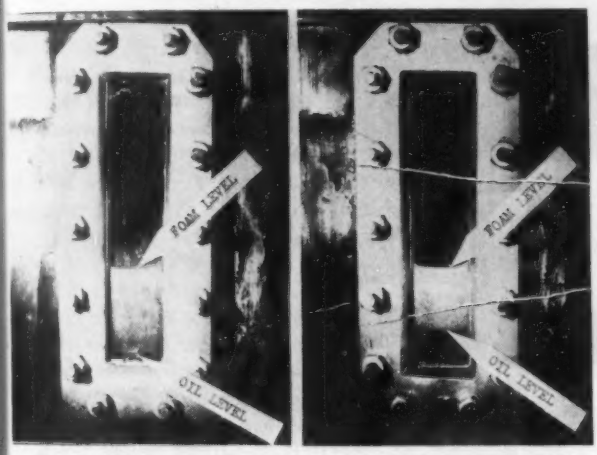
Idle at 500 rpm under no load for 1 hr.
Run at 1000 rpm with a 5-bhp load for 1 hr.
Run at 2000 rpm with a 20-bhp load for 1 hr.
Run at 3150 rpm with a 30-bhp load for 1 hr.
Idle at 500 rpm with no load for 5 min.

The oil temperatures are recorded, but not controlled. Jacket temperature is not allowed to exceed 200 F.

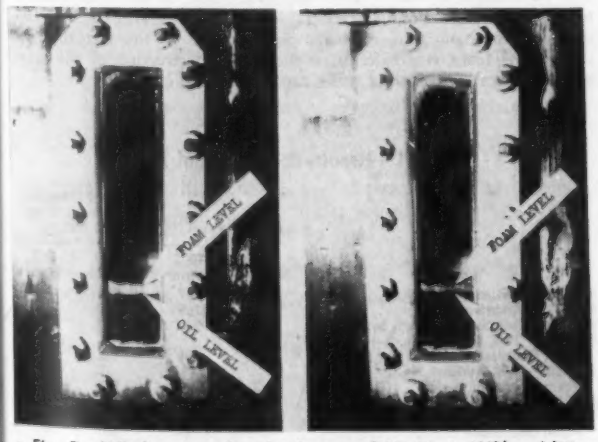
The engine is filled with 5 qt of the test oil prior to each test. The oil level and the foam level is observed and recorded every 15 min during the test conditions outlined above. During the portion of the test when the engine is operating at 3150 rpm, an 8 oz sample of oil is drawn from a sampling line tapped into the oil gallery. This sample can be used to determine the amount of foam circulating through the pressure system.

After the completion of a test, the oil is drained and the engine flushed out by running for one hour at 1500 rpm with a straight SAE 20W oil. This oil is then drained, the crankcase flushed with kerosene, and 2 qt of a light flushing oil circulated through the engine to clean out the kerosene. This rather complicated flushing procedure is deemed necessary to make certain that the effects of oil additives and foam depressants are nullified before proceeding with another test.

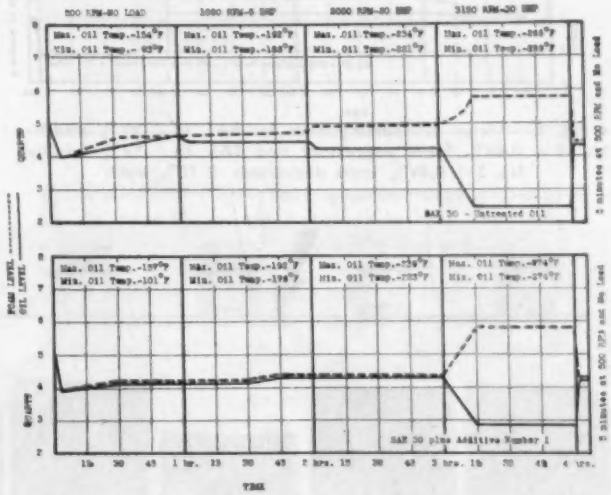
Curves are plotted for each test oil showing variations in oil and foam level throughout the test, as shown in Figs. E-H.



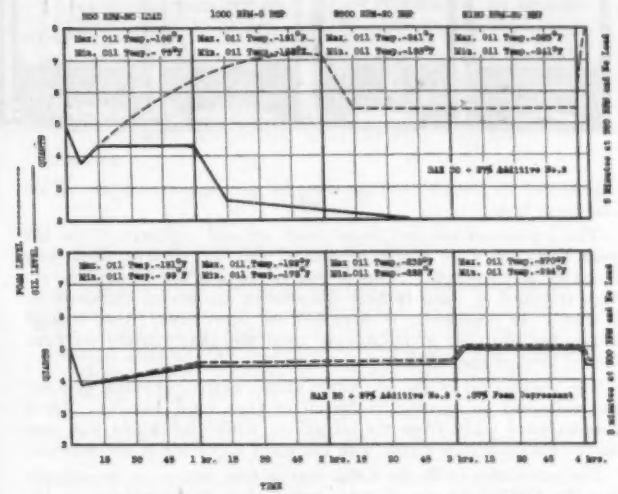
■ Fig. C – Window in side of oil pan – SAE 30 + 2V% additive No. 2 – left: running at 500 rpm, at end of test; right: after standing 15 min



■ Fig. D – Window in side of oil pan – SAE 30 + 2V% additive No. 2 + 0.5V% foam depressant – left: running at 500 rpm, at end of test; right: after standing 15 min

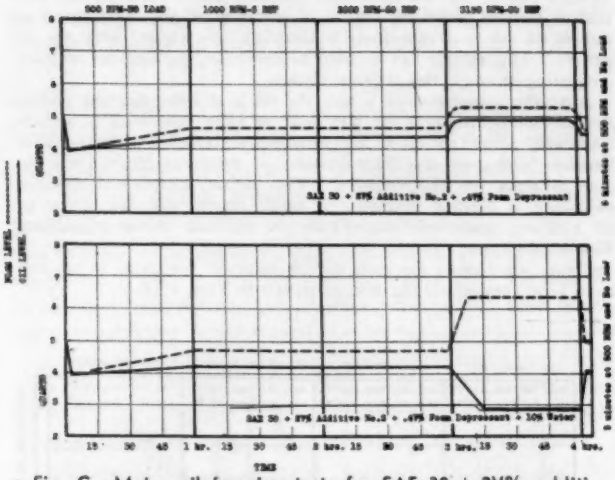


■ Fig. E – Motor oil foaming tests for SAE 30 untreated oil and SAE 30 + additive No. 1

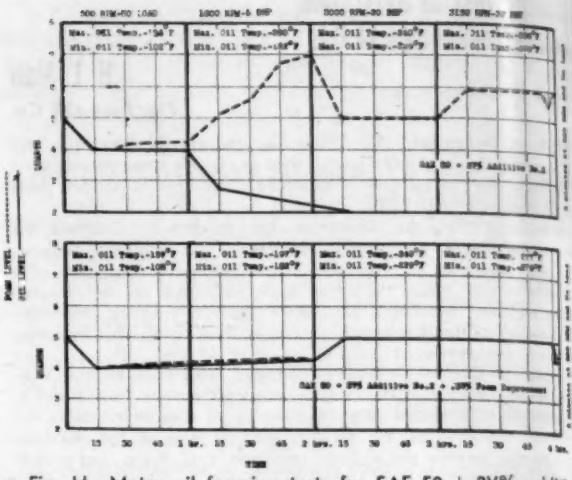


■ Fig. F – Motor oil foaming tests for SAE 30 + 2V% additive No. 2 and SAE 30 + 2V% additive No. 2 + 0.5V% foam depressant

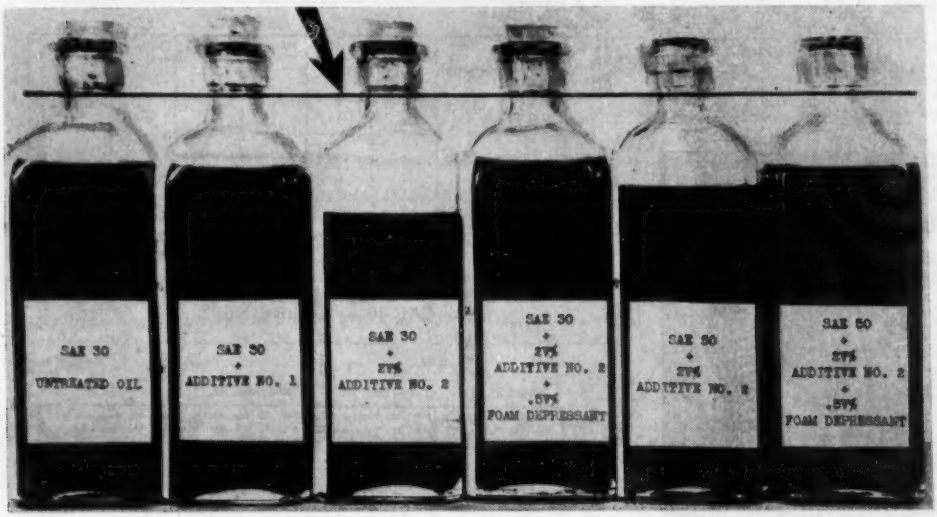
This test can also be used as a preface to the 36-hr Chevrolet test. The engine can be run for the foaming test and if the oil foams badly, there is no necessity of running the 36-hr oxidation test. However, if the oil passes the foaming test satisfactorily, it is only necessary to withdraw 1 qt of oil from the crankcase before proceeding with the regular 36-hr oxidation test. The results of engine tests and correlation of engine tests with the proposed CRC foam test are rather completely shown on the accompanying tables, curves, and photographs. The photograph of the window in the side of the oil



■ Fig. G – Motor oil foaming tests for SAE 30 + 2V% additive No. 2 + 0.4V% foam depressant and SAE 30 + 2V% additive No. 2 + 0.4V% foam depressant + 10% water



■ Fig. H – Motor oil foaming tests for SAE 50 + 2V% additive No. 2 and SAE 50 + 2V% additive No. 2 + 0.5V% foam depressant



■ Fig. I – Oil samples drawn at 3150 rpm and 30 bhp

Arrow points to original level of oil and foam

pan shows its general construction and the appearance of the oil and foam level.

The curves on oil and foam level are self explanatory for the most part. In some it will be noticed that the oil level apparently increases at 3150 rpm. This is due to oil aeration which has not been classified as foam because the relative amount of entrained air is small. Its appearance is more like oil than foam. Also, at high speeds the splashing of oil coming from the main bearing increases the oil level at the window.

The photograph of the six bottles shows relative amounts of foam and entrained air existing in the oil at 3150 rpm (Fig. 1). These samples were taken from the oil gallery while the engine was running. They correlate nicely with the foam appearing at the window.

The correlation with the CRC tests is fair, but seems to indicate that the CRC foam test is more severe than this particular engine test. Table 1 shows the results of the CFR test. For instance, the difference between the untreated oil and oil containing additive No. 1 is not very great in the engine test. In the CRC procedure, however, the difference is quite marked with 13% foam for the untreated oil and 241% for the oil with additive No. 1. Also, the CRC procedure shows SAE 50 + additive No. 2 to have a greater tendency to foam than SAE 30 + additive No. 2. The engine test, however, shows the SAE 30 a little worse than the SAE 50, but both oils were extremely bad.

We think that this foaming test can be nicely introduced as a valuable adjunct to the Chevrolet 36-hr oxidation test. It is interesting to note that, on the data presented, with two different additives, additive Nos. 1 and 2, the maximum foaming occurs at a different speed and load in each case. This may be the reason why foaming troubles on some additives have not been encountered in the 36-hr oxidation test since these data would indicate that at a particular speed the foaming tendencies of an oil may be more pronounced than at another.

I would like to mention that we run the engine 5 min at 500 rpm and no load immediately after the conclusion of the 4-hr foaming test. We have found that this, in some instances, will cause certain additives to foam tremendously, possibly due to the fact that the creation of foam is greater than the actual destruction of the foam by mechanical agitation or other means in the engine.

Table 1 – Results of CRC Foam Test

Lubricant	Average Foam, %		Foam Permanency after Standing 30 Min, %	
	Dry Sequence	10% Water	Dry Sequence	10% Water
SAE 30 Untreated Oil	13	17	Nil	Nil
SAE 30 + Additive No. 1	241	188	103	7
SAE 30 + Additive No. 2 (2% by Volume)	352	383	195	184
SAE 30 + 2V% Additive No. 2 + 0.5V% Foam Depressant	3	95	Nil	Nil
SAE 30 + 2V% Additive No. 2 + 0.4V% Foam Depressant	31	225	Nil	Nil
SAE 50 + 2V% Additive No. 2	408	395	185	122
SAE 50 + 2V% Additive No. 2 + 0.5V% Foam Depressant	2	28	Nil	Ni

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How Will the 100-Octane Aviation Gasoline Program Affect Post-War Motor Gasoline?

by BRUCE K. BROWN* and D. P. BARNARD**

HOW much 100 octane-number aviation gasoline will be available after the war?

How will it be used?

How will the availability of 100-octane gasoline and the equipment to make it affect the quality of motor gasoline?

In view of the intense interest in these very timely questions, the authors have undertaken to project as best they can, on the basis of such facts as are available for public use, estimates which will at least indicate the limits within which these answers must fall. Qualitatively, the best answers which can be given now are as follows:

While all of the high octane-number aviation gasoline facilities can be operated to produce components for this product as long as it is required, an appreciable fraction of the facilities employed must be shut down or diverted to other service as promptly as possible because of operating costs and crude utilizations, justifiable as part of the war effort, but excessive under normal conditions. As much as 100,000 bbl per day of wartime production will probably fall in this category.

Of the facilities continuing to produce high-octane components after the war, some will be operated under conditions which will make available relatively high octane-number stocks for use in motor fuels.

The effect of the use of these components on motor-fuel quality may be appreciable, but in all probability, the result in octane numbers will be far below the mystic value of 100.

In view of the natural interest in the probable effect which the wartime developments in refining technology will have on post-war motor fuels, we shall discuss, in so far as secrecy regulations permit, the following points:

1. How much 100 octane-number gasoline is being made and how much will be made?
2. How is it being made?
3. How much will be made under peacetime conditions?
4. How much will be required for aviation use after the war?
5. To what extent will other requirements compete for high octane-number gasoline components and for manufacturing facilities?
6. How much will be left for use in motor fuel?
7. To what extent will it be used in motor fuel?

[This paper was presented at a meeting of the Metropolitan Section of the SAE, New York City, Dec. 2, 1943.]

* Assistant deputy administrator, Petroleum Administration for War.
** Consultant, Petroleum Administration for War.

ALTHOUGH 100-octane gasoline may someday be feasible as an automobile fuel, that day will not arrive immediately after the war.

The authors of this paper expect that the average octane-number level of post-war gasoline may increase about three units over pre-war values, and thus the majority of marketers will be able to offer house-brand gasoline of about 85 octane and premium grade around 85 octane number.

The picture appears to these authors to be that immediately after the war, these improvements are definitely practical, with succeeding improvements possible but requiring equipment design, development, and construction. In other words, improvements much above these levels will be "evolutionary rather than revolutionary," and the evolution must be predicted with due regard to the fact that at these high octane-number levels already prevailing, each succeeding octane-number improvement becomes increasingly difficult to achieve.

■ ■ ■

THE AUTHORS: BRUCE K. BROWN, who was with Standard Oil Co. of Ind. for many years in the capacities of general patent attorney, manager of development and patents, and general manager of research and development, left to serve in the Office of the Petroleum Coordinator for War as assistant deputy petroleum administrator in charge of the Refining Division. Mr. Brown, who received his chemical engineering and chemical degrees from the University of Illinois, has been employed successively as a chemical engineer for Burgess Laboratories of Madison, Wis., research chemist and patent attorney for Commercial Solvents Corp., and has practiced law privately in New York City. He served in the first World War as a second lieutenant in a machine gun battalion. D. P. BARNARD (M '24), consultant for the Petroleum Administration for War, has long been prominent in technical work of the SAE, particularly that concerned with fuels and lubricants. In 1935 he was vice-president of the Society representing the Fuels & Lubricants Activity. After graduating from the University of Delaware in 1919, he did research work in applied chemistry at M.I.T., where he received his master's and doctor's degrees in 1921 and 1925, respectively. Dr. Barnard has been widely known in the industry as associate director of research of the Standard Oil Co. of Ind.

8. What increase in the octane number of motor fuel will result?

How Much Is Being Made and Will Be Made?

The topic is particularly difficult to discuss during the war when we don't know how long the war will last. The indicated ultimate military requirements have risen more than tenfold in the past 30 months. While it is improbable that there will be further significant increases in the requirements figures, it is not impossible, and hence, we do not know definitely just how much 100 octane-number gasoline will ultimately be needed nor how many facilities we are going to build to produce it.

It takes a year or more, even in peacetime, to build a complete new plant to make 100-octane gasoline and the tremendous expansion in production of combat fuel achieved during the first 18 months of war was attained largely by the use of expedients such as changes in tetraethyl lead content, elimination of plant bottlenecks, conversion of pre-war equipment to the manufacture of special ingredients, and a nationally administered "blending program."

The summer of 1943 marked a significant change in the production situation in that, commencing at about that time, the new facilities construction program began to produce bigger dividends. More and more new plants or operable parts of new plants swung into production to assist the pre-war 100-octane plants and the converted peacetime facilities. A list of 72 new major plants, finished or about to be finished, and 20 more scheduled for next year, was released by the Petroleum Administration in November, 1943.[1] It has been estimated that the new plants alone will cost about $900,000,000. In a recent appearance before a Committee of Congress, Deputy Petroleum Administrator Davies stated that production of 100 octane-number gasoline as of November, 1943, was at a rate more than four times production in the spring of 1942 and that it was scheduled to be doubled again. Whatever the ultimate figure is, it will be a very *large amount.*

How Is It Being Made?

The base stock used for aviation gasoline would be high octane-number motor gasoline if it were not otherwise employed. Base stock forms the greater proportion of 100 octane-number gasoline. The blending agents, alkylates, iso-octanes, and the like, which make up the rest of the aviation gasoline, are really largely byproducts of the cracking of oil because they are made largely from butylene and isobutylene (or similar olefins) which are produced in the cracking reaction, by combination either with each other or with isobutane.

Producing high octane-number aviation gasoline may be likened to a form of sifting process in which we concentrate in one part of the total gasoline the highest octane-number quality. Thus both the amount and quality of the gasoline remaining for automotive use are reduced as a result of the sifting operation. Of course, there are ways by which we can make a great deal of aviation gasoline out of the "4,000,000 plus" bbl of crude which the United States will refine each day and still avoid any substantial reduction in the amount of motor gasoline made. The most obvious way to do this is to crack more oil. This might require more cracking equipment, but, technically at least, it can be done easily enough. However, if we should undertake to make more *total* gasoline – aviation gasoline plus motor gasoline – than we now do we would have no option except to make *less* of the remaining petroleum products. As it is, those other petroleum products, such as fuel oil, must furnish the cracking stock to make the additional gasoline. Fuel oil, however, is necessary also, directly or indirectly, to the war program as well as aviation gasoline. It is therefore generally recognized that there is a definite limit to the proportion of crude that can be converted to gasoline and that, in fact, this limit has been approximated closely, if not reached. For completeness, reference must be made to the production of 100-octane gasoline, actual and potential, from natural gasoline raw materials. It can be done and it is being done. However, the total volume of raw material available from this source is relatively small, both as to total gasoline and potential yield of aviation gasoline.

The production of 100 octane-number gasoline must not be regarded as a procedure in which a stream of raw material is charged to a 100-octane gasoline plant from which the finished product emerges at the other end. As a matter of fact, this fuel is a rather complex mixture of highly specialized synthetic and carefully selected stocks. The better known of these are: alkylate, hydrocodimer iso-octane, isopentane, neohexane, catalytically cracked aviation naphtha, hydroformates, high octane-number straight-run naphthas from selected crude, and cumene. A brief description of each of these components and their relative importance is necessary as a part of our evaluation of the post-war fuel situation.

Alkylate

Alkylate is made by combining butylenes or other olefins and isobutane, such as are produced as byproducts of cracking processes, and the end product is essentially a mixture of the various isomeric iso-octanes. Butylene alkylate forms the backbone of present-day 100-octane aviation gasoline because, while it constitutes substantially less than 50% of the final product, its octane number (after lead has been added) is high enough to permit the incorporation of large percentages of other stocks, particularly catalytically cracked and straight-run naphthas which constitute the principal portion of the product. In a sense, the alkylation process may be regarded as a vapor pressure control procedure inasmuch as the butylenes and isobutane which serve as usual starting materials have octane numbers about as high as the finished product. Their very high vapor pressures, however, (30 to 40 psi) make it impossible to incorporate more than a very small fraction of 1% in the aviation fuel product. The vapor pressure of alkylate, on the other hand, is sufficiently low to facilitate the use of other stocks having vapor pressures higher than that permissible for the finished product. Alkylates can also be and are made in which other olefins are substituted for butylenes.

Hydrocodimer

Hydrocodimer, another product describable as "commercial iso-octane," is produced from the gaseous byproducts of cracking. In this instance, butylenes are polymerized and are subsequently hydrogenated. The end product, however, is similar to alkylate. An earlier development and not so economical from the standpoint either of raw material utilization or dollars, hydrocodimer is not manufactured as extensively as is alkylate. However, all of the other considerations stated above for alkylate apply equally to hydrocodimer, and its emergency wartime production has been of vital importance.

Isopentane

Isopentane may be regarded as a high vapor pressure companion to alkylate. It is obtained principally by the superfractionation of natural gasoline, although a certain amount is manufactured by the "isomerization" of the normal pentane fractions existing in light refinery naphthas. In a typical case, a mixture of about one part

[1] See *National Petroleum News*, Vol. 35, Nov. 3, 1943, Section 1, pp. 22, 24: "100-Octane Output Quadrupled Since 1942."

isopentane to four parts alkylate or hydrocodimer is employed as an alkylate blending agent. The vapor pressure of such a mixture is usually in the neighborhood of 7 psi.

Neohexane

Neohexane is another high-octane isoparaffin falling intermediate between isopentane and the iso-octanes in its volatility. It is manufactured principally by the catalytic process of "isomerization" which converts the normal or straight-chain hexane of about 60 octane numbers into one of the isomeric forms which has an octane number between 90 and 100 before the addition of lead. Its vapor pressure is in the neighborhood of 9 psi and its other volatility characteristics are such as to render it particularly attractive as a means for offsetting the rather high boiling points of some of the other components, particularly the heavy catalytically cracked naphthas and the heavy fractions from hydroforming. Neohexane, while valuable, is only a minor component in the present 100-octane gasoline program.

Catalytic Cracking

The catalytic processes for cracking the heavier crude fractions into light hydrocarbons suitable for incorporation in gasoline form the base of the 100-octane aviation gasoline program. These processes, the best known of which are the "Houdry," "Fluid," and "Thermofor" techniques, permit the refiner to convert light or heavy petroleum fractions, not otherwise absolutely essential to the war effort, into gasoline. The exact choice of charging stock and method of operating the process determine whether the product is suitable for use in 100 octane-number aviation gasoline. The various catalytic cracking processes embody certain inherent differences in their scheme of operation and turn out products also differing in their make-ups. In general, the catalytically cracked naphthas, as produced for use in 100 octane-number gasoline, consist of isoparaffins and aromatics boiling over almost the entire aviation gasoline range. While not always the case, the manufacture of 100 octane-number gasoline may be generally described as one in which naphthas, such as straight-run and catalytically cracked naphthas already quite high in octane number, are brought up to the required octane level by the addition of alkylate blending agent to reach 100 octane number.

As catalytic cracking processes are operated to make aviation gasoline components, their production capacities and the yields of charging stocks are substantially less than would prevail if they were operating to maximum advantage in the manufacture of motor fuel. In some instances, the operating costs, when operating for aviation purposes, are extraordinarily high and can be justified only by the emergency of war. The importance of catalytic cracking to the aviation gasoline program, however, cannot be overestimated because of its versatility in converting heavy fractions to high-octane aviation gasoline base stocks. Admittedly, certain other components, such as alkylate and isopentane, would produce very high-grade aviation gasoline if they could be made available in sufficient quantity. As was pointed out, however, the alkylation process operates upon the gaseous byproducts of cracking, and the starting material for the process, therefore, is available only to a limited extent. Catalytic cracking, on the other hand, suffers from no such stringent limitation and, in effect, provides a process which is not restricted by the necessity of close interlocking with other refinery processes or, as in the case of isomerization, to stocks which exist only in rather small percentages in the original crude. Approximately 60 "cat crackers," as the catalytic oil cracking units are commonly described, are either operating or will operate as a part of the war program. The input of cracking stock will be in the order of 760,000 bbl per day, but in the post-war period (when, as we shall explain, we doubt that they will be operated solely for maximum output of aviation fuel) the daily charge capacity will be nearer 1,000,000 bbl per day. This figure may be compared with the rated capacity of American refineries for old-style thermal cracking in 1940 – about 2,200,000 bbl per day.

Hydroforming

Hydroforming is a catalytic process for converting gasoline-range hydrocarbons of low octane number into components of higher octane number. To this extent it has much in common with the isomerization process. Currently it operates upon extremely narrow cuts obtained from selected crudes. When making motor gasoline components, however, it can be operated on a broader range cut to produce material suitable for fuel use at a considerably lower cost than is now being incurred. Virtually all of the hydroforming capacity in the country was installed for the purpose of making toluene for explosives. A so-called heavy hydroforming cut, however, consisting almost wholly of aromatics heavier than toluene, is produced and is suitable for inclusion in 100 octane-number gasoline. As the demand for explosive slackens after the war, the whole hydroformate could be used for aviation fuel. The total amount of such hydroformate which can be made available from equipment now built or being built is relatively small.

Straight-Run Base Stocks

Straight-run gasoline obtained from certain crudes, particularly those produced in some sections of Texas and California is also used as a base stock. The simple distillation processes whereby such gasolines are produced are well understood and will not be elaborated upon here. Suffice it to say that in one way or another such straight-run gasolines are required in the current program to the fullest extent of their availability.

Cumene

Cumene is an alkylation product of steel mill byproduct coking benzol. It is particularly useful in increasing the so-called rich-mixture performance of aviation gasoline and has aided materially in the earlier stages of the production program. The amount which can be made is directly limited by the amount of byproduct benzol which can be spared for the purpose.

Component Manufacture

We have previously referred to the conversion of peacetime refining equipment to the manufacture of ingredients, that is, "components" for 100-octane gasoline. Seventy-two refineries are participating in this program. These refineries, by and large, had no special 100 octane-number gasoline-making equipment. However, they converted prewar polymerization units (previously suitable only for motor gasoline manufacture) to produce codimer or cumene. The codimer is collected laboriously and moved by tank car to certain other refineries where, after further treatment, it is converted to hydrocodimer, a highly desirable ingredient for 100-octane gasoline.

The refiner producing codimer or cumene has had to give up his polymerization unit upon which he had been relying to produce a high octane-number ingredient for his motor gasoline with a correspondingly decreased expenditure for lead. Not only has he lost the economic advantage of this pre-war operation, the actual output in barrels of product has decreased, due to the necessity for operating the unit in a highly selective manner. A typical figure is one in which a polymerization unit which produced 800 bbl a day of polymer prior to the war, but now makes only 300 bbl a day or less of aviation-type codimer.

The war program for making 100 octane-number aviation gasoline has required resorting to every expedient to obtain maximum production from existing equipment and new equipment under construction without regard either to ordinary refining economics or the extravagant use of crude.

How Much Will Be Made After the War?

There is school of thought which has held that, as the demand for aviation gasoline is reduced with the cessation of hostilities, the unwanted part of the 100 octane-number gasoline to be used as such will be blended more or less directly into motor gasoline. Persons who look on the matter this way feel that, in the post-war period, refineries will simply, in effect, sell as much 100 octane-number gasoline as they can and pour the rest of it back into motor fuel.

There is another school of thought which believes that American ingenuity and technological advances made during the war will be translated very rapidly after the war into powerplants for civilian automobiles which will in-

corporate the features of airplane engines and which will demand 100-octane fuel.

We hold it most probable that in the immediate post-war period neither of those things will happen and that the actual result will be a compromise. It appears that since no new automobiles are being manufactured and since millions of cars will be available after the war which cannot use 100 octane-number fuel to any real advantage, the public simply will not be able to afford to pay the price for a superfuel. Further, it appears very certain that when passenger-car production is first resumed, it will be upon the basis of continuing the 1942 models, at least as far as mechanical features are concerned. In addition, in the event that substantial increases in economy are desired by the purchasing public, several other design avenues offer more promise than does the simple increasing of compression ratio with its attendant high octane-number fuel requirements. It, therefore, seems quite improbable that there will be any substantial early demand on the part of motorists for 100 octane-number gasoline. It may be that, some time long after the war, engineers will produce powerplants which will operate effectively on 100 octane-number gasoline; but even so, for an appreciable number of years to come, the automobile's diet will be determined by the digestion of the older model cars and the pocketbooks of their owners.

It appears quite definite that the production of 100 octane-number gasoline must take a sharp and early drop as soon as the peak military demand is passed. The emergency of war has necessitated the forcing of every possible available unit to its utmost capacity and the taking of stringent measures which, in many instances, are quite uneconomical. As soon as the Government ceases to be a willing buyer for 100 octane-number gasoline, regardless of cost, and as soon as diminishing demand brings again into play the old law of supply and demand and brings about a return of a "buyers' market," the reduced market for high octane-number gasoline will be supplied from low-cost production units.

It has been pointed out that manufacturers of codimer have been forced to produce ingredients for high octane-number gasoline without regard either to cost of the codimer or its effect upon the other refining operations and also that the operators of catalytic cracking units were forced to convert them to the manufacture of base stocks at cost and yields from crude which in all probability could not survive after the war. These are illustrations of the manner in which 100 octane-number production was achieved under the duress of war emergency.

As another illustration of the ephemeral character of much of our 100-octane gasoline produced during the early stage of the war, it should be appreciated that the petroleum industry in "lifting itself by its boot-straps" to produce greatly increased quantities of high-octane aviation gasoline without having completed its new manufacturing facilities, of necessity, had to operate almost all of its refineries as if they belonged to one organization. This was done under the guidance and authority of the Petroleum Administration. In this operation, components from East Chicago for example were sent by tank car to Philadelphia to be blended in 100 octane-number gasoline; components from Whiting, Ind., were sent to Texas, and isobutane available on the East Coast was shipped by pressure tank car to the West Coast. These operations, however, extravagant both in operating and transportation cost, practically saved the day for our Air Forces. Obviously, they could not be continued in peacetime if for no other reason than that the antitrust laws would prevent it. In order to be entirely clear on this point, it should be definitely stated that we think that a great deal of 100-octane production which this country was making available by the middle of 1943 was wartime emergency production which will continue only as long as the war and which will certainly stop immediately after the war.

Similar situations probably exist for many of the processes now being used to make aviation gasoline and its components. Perhaps the simplest illustration is the isomerization process now being used to make isobutane out of ordinary normal butane. This is a wartime necessity, but in peacetime, at least, the highest value that butane can be given in refining economics is its value as gasoline (or rather what its value would be if it were blended into motor gasoline). It is perfectly obvious that in the post-war world, isobutane made by a chemical process from normal butane will have to compete in price with such isobutane as is available from natural gasoline and from any and every other process. This does not mean that when the war is over we will entirely cease to isomerize butane. It does mean, however, that the less economical units will probably stop operating.

This same thing is true, in principle, of other processes. It may even be true of certain cracking processes which use high-quality naphtha as a raw material. These processes will, in effect, "make gasoline out of gasoline," or, at best, out of high-quality virgin stocks almost as valuable as gasoline. Such operations will be required to compete with other processes which actually create gasoline out of much lower grade petroleum fractions. As has already been pointed out, many of the operations currently producing 100-octane gasoline are quite extravagant in a number of ways, including their requirements for crude oil. The United States is not going to run out of crude for many years yet to come, but it is fairly evident to all concerned that our national crude oil position is rapidly changing from one in which we are producers of excess crude to one in which we will be required to import our marginal crude requirements. This change in position is apt to alter the fundamentals of our refining economics and will severely penalize those operations which do not use crude oil to maximum efficiency. The crude situation, therefore, will place much emphasis upon the desirability for shutting down as soon as possible after "V day" those aviation gasoline manufacturing operations which are wasteful of crude.

■ How Much Will Be Required For Aviation?

The real answer to the foregoing question, of course, involves an accurate estimate of the economic and political conditions after the war. Certainly the authors do not possess the clairvoyance necessary to forecast this picture with any appreciable degree of accuracy. However, certain estimates must be made if we are to arrive at any kind of a picture of the post-war motor gasoline situation.

Prior to the development of the present national emergency, all civilian aviation gasoline uses totaled less than 9000 bbl per day, with scheduled airline consumption accounting for just about half. If we assume that a fivefold increase in domestic airline fuel consumption will develop during the first five post-war years and that the use of

aviation gasoline by other civilian agencies (including engine builders) will increase in like amount, we conclude that the total post-war civilian aviation gasoline requirement may be in the neighborhood of 50,000 bbl per day. Of this, of course, the lion's share must be selected on the basis of economic considerations and, therefore, can embody only those components which can be produced at the lowest cost. It may be that the bulk of this business will continue to use the familiar more economical 91-octane grade currently used by the airlines. Post-war military demands are, of course, almost impossible to predict. Prior to the emergency, they accounted for scarcely 5000 bbl per day and just before Pearl Harbor had risen to something less than 30,000 bbl per day. If we assume that such demands will rapidly drop at the conclusion of the war to about 50,000 bbl per day, we would still be operating at about ten times the pre-emergency level. This fuel will, of course, fall in the highest octane-number category and in all probability will be better than the present 100 octane-number grade.

In general, such estimates as can be made at this time indicate a maximum probable early post-war demand for all aviation fuels of about 100,000 bbl per day, of which somewhat more than half will be of the high octane-number type, while the balance will be selected mainly on a cost basis. Our estimates may or may not be accurate within 50%, but in any event, it appears that any probable variation in the amount of aviation gasoline required in the early post-war period will have comparatively little effect upon the quality of post-war motor gasoline. Therefore, it is believed that the foregoing estimates should suffice for the purpose at hand.

■ Other Uses for High-Octane Gas Components

Discussion up to now has dealt primarily with refining economics and its effect in determining how much of the high-octane aviation gasoline facilities may be operated under peacetime conditions. However, there is another factor which must be appraised in determining the amount of high-octane gasoline components to be available for use in motor gasoline after the war. This is the possibility of a greatly enlarged opportunity to produce chemicals from petroleum. The war has already brought into being one important new industry which depends upon petroleum; namely, the synthetic rubber industry. The same butylene which can be and is used as an important raw material in the production of high octane-number blending agents can and is being used for making butadiene for buna rubber; and the same isobutylene which is also a valued ingredient for iso-octane blending agents is the raw material from which butyl rubber is made. Should there be any post-war expansion in the synthetic rubber business, either as the result of increased demand for rubber as such, or because technological improvements will permit synthetic rubber to be used to a greater extent and in wider fields than natural rubber was used, the logical source of the raw material for such an expansion would be the refining equipment now being used to produce 100 octane-number gasoline. The tremendous wartime output of butylenes for 100-octane gasoline and for rubber suggests that as military demand for aviation fuel drops off from the peak, butylenes may drop in price. The last increment of butylene forced from an "unwilling unit" usually occasions the greatest refining loss and the greatest maintenance expense.

Butylene, isobutylene, and isobutane are active materials from the chemical standpoint and it is very likely that there will be a demand for them by industries other than the synthetic rubber industry. Turning now to the base stock, that is, the high octane-number naphtha which forms the preponderant part of 100 octane-number gasoline, the present trend is such that these base stocks are being produced to contain maximum proportions of aromatic hydrocarbons. Should the chemical industry develop a sizable demand for benzene, toluene, and xylene, it would be most logical to obtain these materials from the naphtha that would otherwise be available for 100 octane-number gasoline.

■ How Much Will Be Left for Motor Fuel?

The foregoing discussions present two reasons for assuming lessened production: one based on refining economics, and the other based on possible advances in technology. While there can be no such thing as definite quantity of 100-octane aviation gasoline available after the war, certainly (if the demand continues) there is no reason to assume that we cannot continue to produce the amount necessary to satisfy that demand. However, as the demand is reduced (and certainly there will be a substantial reduction), the result definitely is *not* going to be that there will be a considerable excess of 100 octane-number gasoline available on a distress market. On the contrary, *that part of the 100 octane-number gasoline which is not required simply will not be produced.*

If we combine this conclusion with the obvious fact that there will be some requirement for high octane-number post-war aviation gasoline, it becomes fairly evident that, with some possibly minor exceptions, the real effect of the aviation gasoline war program on post-war motor gasoline quality will be to make available, after the war, certain facilities – notably for catalytic cracking – which, instead of manufacturing aviation gasoline base stock, will be operated under more economical conditions to make motor gasoline.

■ How Will Extent of Its Use Be Determined?

In line with the foregoing, it appears that the extent to which high octane-number aviation gasoline manufacturing facilities will contribute to post-war motor gasoline quality can be briefly summarized about as follows:

1. Most catalytic cracking capacity will be converted to the manufacture of high-octane motor gasoline as rapidly as possible after the conclusion of the war.

2. Alkylates and other high octane-number blending agents will find their way into motor gasoline only to a comparatively slight extent and then only as their use can be justified as means for vapor pressure control, that is, as means for incorporating in motor gasoline the high octane-number but very volatile butanes and the like which might otherwise be lost.

3. Polymerization units will no longer produce the expensive codimer required for aviation gasoline use, but rather will revert to their pre-war procedures of nonselective polymerization in which they occupy a position similar to that predicted above for alkylation; namely, to increase octane number and in the same sense as means for vapor pressure control.

4. The various isomerization processes, and also hydroforming, must occupy a marginal position if they are to figure in post-war motor gasoline manufacture at all. Es-

sentially these processes "make gasoline out of gasoline" which, in turn, means that they can be justified only as long as the cost of the operation as a quality-improving procedure can be passed on to the ultimate purchaser. It may well be that some of these processes will find a few important applications in the post-war picture, but it is fairly certain that they will not account for any substantial percentage of ultimate motor gasoline production. By and large, it appears that the effect of the aviation gasoline program on post-war motor gasoline quality can be judged on the basis of the effect of catalytic cracking facilities.

Within the limits of accuracy of the estimates which can be made, it would appear that catalytic cracked motor gasoline will be produced to the limit of capacity of the available equipment, even though this may involve the shutting down of a substantial portion of the older thermal cracking units. If this is done, possibly 25 to 30% of the post-war motor gasoline production in this country may consist of catalytic cracked naphthas as the principal contribution of the wartime aviation gasoline program to post-war motor gasoline quality.

■ What Increase in Octane Number Will Result?

Catalytic cracked naphthas as produced for motor gasoline manufacture will, in general, tend to run 10 to 12 octane numbers higher than the materials which they will replace. This, therefore, indicates that the net effect of the aviation gasoline program will be to permit an increase of from two to three octane numbers on the average over immediate pre-war levels. This estimate, of course, assumes that lead usage will not vary appreciably over that prevailing before the war. We do not mean to imply that immediately after the war the octane numbers of all gasolines marketed throughout the country will uniformly increase by two to three octane numbers. On the contrary, much of the catalytic cracking capacity to which reference is made is localized in a few areas of the country. It may well be, therefore, that certain manufacturers will offer products representing substantially greater improvements than the average indicated.

There is considerable opinion to the effect that the advantages of high octane-number catalytic cracked gasoline may appear, in considerable part, as markedly improved premium-type gasoline. Obviously, so long as premium gasoline accounts for a comparatively small percentage of the total, many manufacturers might find themselves able to market premium gasolines of very high knock rating, indeed. The best guess as to the extent to which the octane levels of such products may be increased can probably be based upon a consideration of automobile requirements.

There are many reasons why we may anticipate a civilian demand for gasoline of improved octane numbers. One of these, of course, is that the automobile companies will wish to start with a new slate in order to produce the best type of automobile to be used by our people. It appears to us that in the post-war period, taxes will inevitably be very high and motor fuel will have to carry a large share of this tax. Also, unless there is a marked change in the outlook for crude oil in this country, it appears that in the post-war period we will have every reason to economize as much as possible on the quantity of crude petroleum used for transportation. The automobile and petroleum industries have never been able to stray very far from the fundamental basis of the maximum number of car miles obtainable from a barrel of crude, at the same time complying with acceptable standards of performance. These factors indicate a real demand for an economy automobile as soon as possible after the war. Such an automobile might, of course, take the form of a small light car having a somewhat lighter and more powerful motor of the high compression type and requiring a very high octane-number fuel. It appears, however, that certain other avenues for improving economy appear more immediately attractive than that of mere compression ratio increase. We refer, of course, to the now well-recognized method of reducing engine speed and amount of mixture breathed per car mile, and the significant position of the automatic transmission in effecting such improvements. By such means, quite large economies can be effected without recourse to high compression ratios with their attendant high engine and fuel manufacturing costs.

It is our belief that perhaps too much significance has been attached to the expression "100-octane aviation gasoline." Actually, as is well-known, the octane number of aviation gasoline, while highly important, is only one of several equally important qualities. In particular, aviation gasolines must be made to much narrower boiling ranges than are required for automobile use which, in turn, profoundly influences yields obtainable from a given quantity of crude. However, the fact should not be overlooked that 80-octane gasoline is still very good gasoline.

By and large, it appears that about 85 octane number, as expressed by the ASTM procedure, represents the highest knock rating which can be used effectively by automobiles to be available in the immediate post-war period. By the expression "used effectively," we mean automobiles which, by virtue of design, construction, *and maintenance*, are able to deliver the performance and fuel economy necessary to justify the costs associated with producing fuel of that quality. It has been emphasized by the foremost authorities in the automobile industry that far-reaching fundamental engine design changes will be required to make possible the effective utilization of gasolines much over 85 octane number. The writers, therefore, hazard the guess that in the first post-war year or so motor gasoline will rapidly approach octane levels of about 85 for premium and around 80 for the regular grade. If there are still "octanes to spare," they will probably be used in the deluxe (premium) grade of fuel. It appears, further, that a third grade of lower octane number will still be available at a lower price. These fuels will adequately take care of the highest requirements of the cars producible in the immediate post-war period, as well as the great bulk of pre-war designs which, of course, will constitute the preponderant share of cars in use at that time.

If these values seem discouragingly below the glowing predictions which have occasionally appeared in the press and which have been bandied about so blithely, we should not overlook the fact that 85 or even 80 octane-number gasoline is still very good gasoline. Further, we should not overlook the fact that practically none of the cars currently on the road can take full advantage of motor gasolines as they were supplied just prior to the war. It, therefore, appears that the war aviation gasoline program will contribute to post-war gasoline a quality increase which at first sight will seem rather modest but which, on closer scrutiny, will be adequate to support the development of cars of the improved economy characteristics certain to be in greatest demand.

Conclusion

To recapitulate the foregoing discussion:

1. 100 octane-number aviation gasoline is now being made to the maximum capacity of the industry without regard to the economics of operating cost, transportation, or crude utilization.

2. Under peacetime conditions, the amount of such gasoline to be required must, of course, be but a fraction of the wartime requirements.

3. Under such conditions, the amounts which can be made within the acceptable limits of economics of crude utilization and using the equipment of the war emergency program are probably not over half of the present schedule.

4. The net result is that immediately after the war a substantial portion of our emergency aviation gasoline manufacturing facilities will be discharged from war duty, and those which can be economically operated will be turned to the manufacture of motor gasoline for which they will run under conditions chosen for economy rather than maximum octane number.

5. These and the other factors concerned in this discussion seem to indicate that immediately after the war we can expect that the average octane-number level of motor gasoline may increase as much as three units over immediate pre-war values and that the majority of the marketers will be able to offer house-brand products around 80 and premium products of, say, 85 octane number.

6. It seems fairly probable that values higher than these would result in extravagances in crude usage which will not be tolerable in the post-war period.

In conclusion, it is not intended that the foregoing be interpreted as indicating that 100 octane-number motor gasoline will *never* be feasible as an automobile fuel. Rather, the picture appears to be that immediately after the war the foregoing improvements are definitely practical with succeeding improvements possible but requiring equipment design, development, and construction. In other words, improvements much above the indicated levels will be "evolutionary rather than revolutionary," and the evolution must be predicted with due regard to the fact that at these high octane-number levels already prevailing each succeeding octane-number improvement becomes increasingly difficult to achieve.

DISCUSSION of PIGOTT Paper

cont. from p. 86

Stress Usefulness Of Foam-Suppressors

– W. L. Weeks and E. V. Farrar

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WHILE appreciating Mr. Pigott's reticence in discussing foam-suppressing agents in this paper, it is felt that some mention should be made of the help which such an agent can lend in at least some cases of aircraft-engine powerplants.

We realize that broad distribution of a foam-suppressor is a special problem, but it should be noted that the agent may come to have a definite place in the general problem of suppressing foaming. Such experience as we have so far gained indicates that in an oil system which held entrained an objectionable quantity of air, the use of a foam-suppressing agent in the oil reduced the entrainment by the ratio of at least 3 to 1. No alterations in tank design were made. Endurance tests which we have made on at least one foam-suppressing agent now available have to date shown no bad effect on engine parts or performance.

In speaking of the pressurization of oil tanks as the "fix" for performance of engine pressure pumps which have been confronted with altitude demands beyond their capacity, it is well to note that the idea has had considerable practical application. The British have for some time found it worthwhile to boost the pumps in some fighter airplanes by this method and did so in the face of the objection which, as Mr. Pigott says, is commonly brought up.

On the subject of centrifugal pumps as boosters for engine pressure pumps, Mr. Pigott's explanation of the problem might be expanded to say that the application of that type of pump calls for special design considerations. Its action is not required at ground level. It should nevertheless not present too great a flow restriction to the rotary pump on the engine because, when the oil is cold, pump priming will be delayed. As a supplement to the booster, in order to provide low resistance to flow when it is not running, a generous, low resistance bypass valve has been considered. Special types of centrifugal pumps having low resistance to flow when not running are available.

Mr. Pigott has said earlier in his paper that "what we often need is a 'fix' to take care of present production and engines already operating." This is, of course, applicable to oil tanks and it is felt that Mr. Pigott's treatment of accessory devices for air (gas) separation somewhat slights the "fix" possibility. He says we "have so far not found any overall improvement in separation by the accessory and tank combinations . . ." provided the tank was itself a good design.

Fair enough statement, but it naturally raises the question – Are most existing tanks actually of good design?

It can be said that there are tanks now in production and service which do not separate efficiently and for them the "fix" by accessory means is well worth some thought. This thought may, we believe, be carried over into the paper's numbered conclusions.

Number 7 classes bad tanks as those which leave from 16% air, down to 3 or 4%, entrained and delivered to the engine pressure pump. Then, in conclusion Number 8, it is stated that – "All tanks can be altered to best design without much trouble, except with the designers."

Aside from being pretty rough on designers, this practically assigns a post-war status to the business of correcting airplane oil systems which have bad separating characteristics.

Actually some tanks in wide military use and current production could and should be improved if their construction permitted internal modification.

It is safe to say that 90% of them cannot be reworked. In those cases an accessory separator would do a wartime job in the maintenance of normal engine oil pressure.

On the remaining 10% group of bad tanks, which let's say can be taken apart without using a can-opener and without working ruin on the rubber lining, things can be done in the way of, for instance, lowering return oil velocity, eliminating cascading and so forth. Admittedly, such modifications involve some intelligently done bench testing. But, as Mr. Pigott knows from his company's association with ours, over the past year and a half on the subject problem, most of the leading aircraft concerns are equipped in material and personnel for such tests.

A little way back we charged Mr. Pigott with being rough on the designers of oil tanks. We are sure he will agree that, under the stress and pressure of war production, new concepts which involve radical change in established specifications, do not thrive readily. This has been clear to us ever since the attention of several companies was drawn to the fallacy of the centrifugal separation theory which has always been bound up with the conventional hopper for oil tanks.

It is true that both the aircraft engineers and the engine people were late in recognizing the importance of air separation. Also, in the absence of a practicable air/oil indicator, effective evidence for the indictment of some hoppers in question was hard to get. In respect to instrumentation for spotting entrainment trouble, the situation is presently improved and with evidence of trouble established we confidently expect some action.

Many tank designers are now not only entrainment-conscious, but now having a tool in their hands, are determined to force the issue of correction.

Comparison of HYDRAULI an ACCESSORY SYSTEM in

COMPARISONS of hydraulic and electrical accessory systems in the past have been for the most part confined to problems surrounding a specific installation on a particular airplane.

The conclusions drawn were limited to the particular problem at hand and it was difficult to fathom out any general comparisons over a wide range of sizes. In addition, the results were not always directly comparable, thereby adding further complications to the problem of making a fair and general comparison of the merits of the two systems.

The authors of this paper have made a general study of the weight and efficiency of the two systems and have resolved them into factors which are directly comparable and from which definite conclusions can be drawn.

The comparisons have been made on the two following bases:

1. A comparison of the individual components of each system on the basis of weight in pounds per horsepower at various horsepower ratings and at efficiencies which are commonly used in aircraft practice.

2. A comparison of typical hydraulic and electrical accessory power transmission systems designed to perform an identical job on a modern airplane.

A study of the comparison of the individual components will reveal specific advantages for one of the two systems and show to a great extent the influence of the component parts on the complete system.

Extreme care was taken to compare only those units which are in active use today and which represent the best performance and weight characteristics consistent with the meeting of Army and Navy requirements.

This eliminated the excessively heavy units of bygone days and also a few extremely light-weight units with erratic performance and relatively short life. In addition, this method yielded smooth comparison curves from which trends can easily be established.

All of the components compared either met the existing weight and performance specifications of the Army and Navy, or were capable of meeting them.

Recent developments in both the hydraulic and electric fields have introduced accessory equipment which is intended to show considerable improvement over the existing equipment.

Due to the fact that very little of this equipment is in use and the fact that it is still in the experimental stage in so far as usage is concerned, the authors have confined their comparison strictly to 28-v, d-c electrical systems and 1500-psi hydraulic systems.

The comparisons of component items which were made are listed as follows:

1. Power generating devices.
2. Power transmission systems.
3. Directional controls.
4. Power application devices.

The range of sizes compared covered the present-day aircraft field.

The first item of comparison is the device which converts airplane-engine power into a usable accessory power. In an electrical system, this device is called the generator while the equivalent in a hydraulic system is the hydraulic pump.

The weights of these units are not directly comparable as such since both of these devices operate continuously in flight and therefore the weight of the fuel consumed by these items must also be considered since this fuel performs no useful purpose in improving the range or performance of the airplane.

For purposes of comparison, an average flight time of 5 hr was used. In addition, voltage and pressure controls are necessary to utilize and control the energy output properly so that the weight of the voltage regulator and reverse current relay must be added to the weight of the generator; and the weight of the pressure regulator and check valve added to the weight of the hydraulic pump.

Since hydraulic systems are adversely affected by low temperatures, comparisons were made at ambient air temperatures of 60 F and —60 F.

Fig. 1 shows first the weight in pounds per horsepower of a hydraulic pump.

This is the lowest curve on this graph. Slightly above this we have the weight of a hydraulic pump and pressure regulator. The last curve of the lower group shows the weight in pounds per horsepower of a hydraulic pump plus the pressure regulator plus the weight of the fuel consumed in continuous rotation of the hydraulic pump for an average flight of 5-hr duration.

This added fuel weight also includes the fuel required to circulate the fluid through the pressure regulator, and through the circulatory portion of the hydraulic system at an ambient air temperature of —60 F.

Since this is the lowest air temperature at which such equipment is expected to operate, intermediate temperatures were not chosen as the latter two curves are quite close together.

A bit of explanation regarding the function and fuel consumed of a hydraulic pressure regulator may be in

[This paper was presented at the SAE National Aircraft Engineering and Production Meeting, Los Angeles, Calif., Oct. 1, 1943.]

and ELECTRICAL in AIRCRAFT

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HYDRAULIC systems have the advantage of lighter weight, particularly in the higher horsepowers, when compared with electrical systems, although both systems should maintain their rightful places in aircraft, to be used where they can contribute most to the efficiency of the airplane as a whole.

These conclusions have been reached by the authors of this paper, after having made comparisons on the following bases:

1. A comparison of the individual components of each system on the basis of weight in pounds per horsepower at various horsepower ratings and at efficiencies that are commonly used in aircraft practice.

2. A comparison of typical hydraulic and electrical accessory power transmission systems designed to perform an identical job on a modern airplane.

This study has been confined to 28-v, d-c electrical systems and 1500-psi hydraulic systems.

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THE AUTHORS: W. C. TRAUTMAN, chief engineer of Bendix Aviation, Ltd., has been directing the design and development of Bendix hydraulic equipment since 1938. He has written several articles on aircraft hydraulic equipment, including "Aircraft Lends Itself to Hydraulic-Electric Units," which he delivered before the SAE last year. He is a member of SAE Subcommittee A-6 on Aircraft Hydraulic Standardization. R. E. MIDDLETON is chief engineer of Aircraft Accessories Corp. After his graduation from the University of Michigan in 1929, where he received a B.S. degree in aeronautical engineering, Mr. Middleton was employed by the U. S. Army Air Forces, Materiel Division, Wright Field, for 11 years, and later joined Curtiss-Wright Corp. as landing gear staff engineer.

order, as a popular misconception regarding their function seems to exist.

A hydraulic pressure regulator is connected in the outlet line of a hydraulic pump. One of the ports of the regulator leads to the pressure accumulator while the third and remaining port leads back to the reservoir. It is the function of the pressure regulator to charge the accumulator to a predetermined pressure and then divert the fluid output of the pump through an internal bypass back to the reservoir.

This bypassing of fluid back to the reservoir at a very low circulatory pressure continues until a fluid demand is made upon the accumulator, whereupon the drop in pressure of the accumulator causes the regulator to cease bypassing and recharge the accumulator automatically.

This cycle continues indefinitely and is analogous to the action of the voltage regulator with respect to a generator.

Knowing that fluid pressure is needed mainly at the beginning and end of a flight, plus a few times during a flight, it is obvious that the pump is bypassing through the regulator back to the reservoir for the majority of the time, and is therefore drawing very little power from the engine.

Hence, the very close relationship between the curve of hydraulic pump plus pressure regulator and the curve of hydraulic pump plus pressure regulator plus fuel consumed.

The curve of generator weight can be compared directly to the weight of a hydraulic pump. However, since a generator requires a voltage regulator when used with a storage battery and a reverse current relay to prevent the battery from draining back through the generator when the generator is not operating the weight of these items must of necessity be charged directly to the generator.

The upper-most curve of Fig. 1 represents the weight of typical generators with their attendant voltage regulators and reverse current relays. This upper-most curve of the generators is comparable to the middle curve of the hydraulic pumps.

Unfortunately no data are available on the fuel consumed by a generator when running under no load and a direct comparison with the hydraulic pump, regulator, and its fuel consumed is therefore not possible.

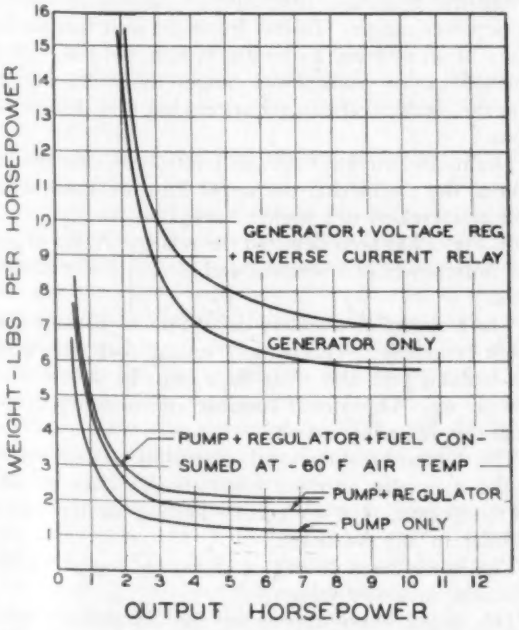

WEIGHT - LBS PER HORSEPOWER
OUTPUT HORSEPOWER
GENERATOR + VOLTAGE REG + REVERSE CURRENT RELAY
GENERATOR ONLY
PUMP + REGULATOR + FUEL CONSUMED AT - 60° F AIR TEMP
PUMP + REGULATOR
PUMP ONLY

Fig. 1 – Comparison of electrical generators and hydraulic pumps

However, it is known that the fuel consumed by a generator is slightly greater than that of a hydraulic pump of equal horsepower.

Also, since a generator may be operating a greater percentage of the flight time than a hydraulic pump because of the drain on it occasioned by radio equipment, lights, and so forth, it would be very difficult to apportion accurately the fuel consumed by the generator during the operation of radio and lighting equipment.

All of the data on pumps and generators are based on a speed range of 2500 to 4000 rpm. It can be assumed that the advantages of higher speed and the disadvantages of lower speed will be the same for both the hydraulic pump and electrical generator.

All of the curves shown have been extended out sufficiently far so that they have assumed a position almost asymptotic to the horizontal.

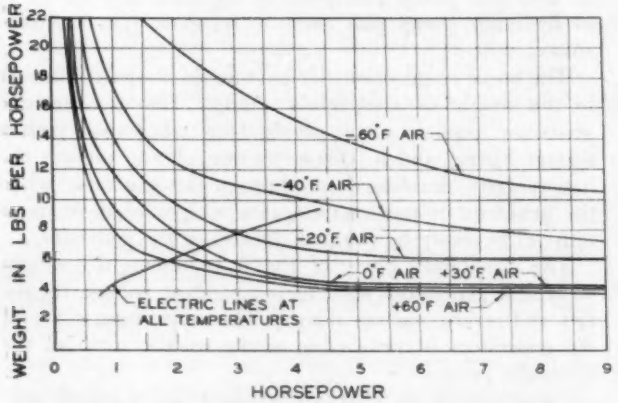
The trend in higher horsepower ratings is, therefore, definitely established and the limitations of the two systems, if any, are readily ascertained.

Fig. 2 shows the weight of electrical and hydraulic power transmission systems. This included in the electrical system the wire, terminals, junction boxes, terminal strips, and connector screws in the same proportions as a typical aircraft installation. The hydraulic system included tubing, oil, tube fittings, and disconnect couplings in the same proportions as a similar aircraft installation would have. The line length used was 50 ft and the efficiency of both types of conductors was maintained at 90%.

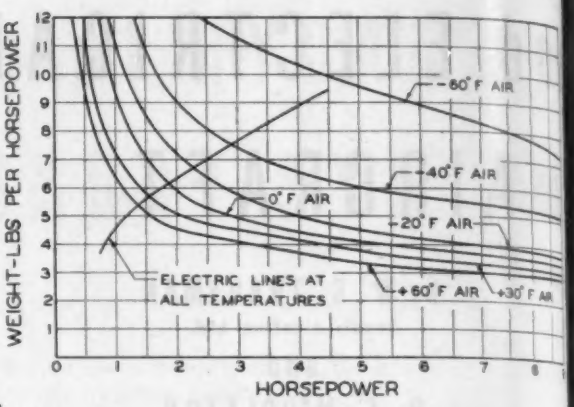
A severe handicap is imposed on the hydraulic transmission system, as 100 ft of tubing are necessary to transmit power 50 ft because a return conductor is necessary. The electrical system was based on using the metal airplane structure as a return conductor.

The effect of subzero air temperature is readily noticed on the hydraulic system. Due attention must be given to the slope of the electric and hydraulic curves as they are exactly opposite. It is quite evident that in the lower horsepower ranges the electrical conductor is much lighter than the hydraulic conductor when considering the full temperature range.

However, when studying the slope of the curves and their directional tendencies, it is readily evident that in the higher horsepower ranges the hydraulic conductor may become more efficient than the electrical conductor even when considering the wide temperature range. Unfortunately data on large electrical transmission systems are not available and a true comparison in this region cannot be made.



■ Fig. 2 – Comparison of hydraulic and electric transmission line systems at various air temperatures (efficiency of both conductors: 90%)



■ Fig. 3 – Comparison of hydraulic and electric transmission line systems at various air temperatures (efficiency of hydraulic conductor: 80%; of electrical conductor: 90%)

Fig. 3 shows the identical relationship as Fig. 2, except that the efficiency of the hydraulic conductor has been dropped to 80%, a value which is much closer to actual aircraft practice than 90%. The electrical conductor has been left at an efficiency of 90% as this is very close to good aircraft practice.

Here we see a very marked improvement in the weight of the hydraulic conductor. In fact the reduction in weight is much greater than the reduction in efficiency. This simply means that the optimum weight is achieved at a lower efficiency than that of the electrical conductor.

We also see that the weight of the hydraulic conductor is equal to the weight of an electrical conductor at a much lower horsepower than in Fig. 2.

One can conclude from these curves that the electrical conductor is lightest in the very low horsepower ranges while the hydraulic conductor is lightest in the higher horsepower ranges. In fact it can be seen that the limitations of a 1500-psi hydraulic system have not yet been reached in the horsepower ranges shown in Fig. 2 but that the electrical conductor system has very definite limitations.

Higher electrical voltages will, of course, alter this limitation to the extent that the upper limit of economic usage will be increased to a higher horsepower rating.

In Fig. 4 has been depicted the relative weight in pounds per horsepower of hydraulic and electrical directional controls.

The majority of accessory devices on an airplane must be made reversible. This simply means that such items as the landing gear and wing flaps must be driven down as well as up. Directional controls are necessary to accomplish this reversible action.

The hydraulic directional control is a four-way valve of the manually operated type. Such valves are usually of the poppet type to ensure locking of the hydraulic cylinder in any position.

The lower curve in Fig. 4 shows the weight of typical hydraulic four-way valves.

The upper curve represents the composite weight of two power relays, and a small low-current type toggle switch operates the coil of either of the power relays or

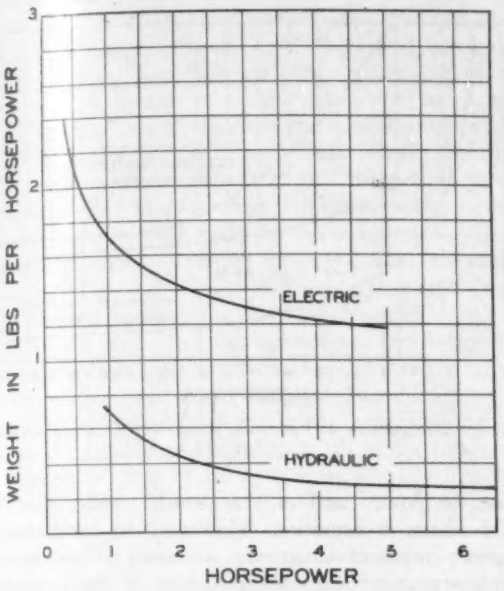


Fig. 4 – Comparison of hydraulic and electric directional-control devices

...nding upon the position of the toggle switch.

The function performed by this combination on an elec-...ic motor of the reversible type is exactly identical to that ...erformed by a hydraulic four-way valve on a hydraulic ...ctuating cylinder.

Unfortunately relays are rather inefficient devices with ...espect to weight particularly because of the copper wire ...nd heavy iron core that is required.

Hydraulic four-way valves are usually made of alumi-...um alloy, plastic, and steel.

This explains the wide gap between the two curves of ...ig. 4.

However, since neither power relays nor hydraulic four-...ay valves occupy a great percentage of the total weight of ...ther system, too much importance need not be attached ...o the wide gap shown on Fig. 4.

The final criterion will be the weight of the complete ...ystem, and the comparison of the components only shows ...general influence.

We have discussed three of the fundamental units of ...he accessory power system; namely, power generating ...nits, the line transmission system, and directional con-...ols. We propose now to continue this comparison and ...iscussion with power application devices as the subject.

The most commonly used power application device in ...he aircraft accessory system is that which translates ac-...essory power whether it be electrical, hydraulic, or some ...ther medium, into linear motion for the operation of ...anding gear retracting mechanisms, the opening and clos-...ng of bomb doors, the operation of flaps, and the like. ...his is the most common problem that is met in the opera-...on of accessories on modern airplanes.

In making this comparison, it is assumed that the same ...undamental mechanism is used in either case. In other ...ords, it would be possible to remove a hydraulic actuat-...ng cylinder and substitute directly an electrically driven ...ctuator. In considering the most efficient device to be ...sed for this application, it is necessary first of all to de-...rmine the performance requirements which must be met. The first requirement will be the peak load which the device must overcome. Next we must consider the maximum time allowed to complete the function or the desired operation. We must also consider the range of operating conditions which will include operating temperatures and such factors as structural attachments and loads and stresses in the device itself. For this reason, we will find a considerable variation in the overall weight of the actuator, depending upon the type of installation that is required. For instance, some hydraulic actuating cylinders are required to meet very difficult structural attachment conditions.

In many cases long extensions of end attachment fittings are needed; sometimes the piston rod is much longer than is necessary because of the installation requirements. Sometimes the actuating cylinder must carry auxiliary devices which are linked in with the function of the mechanism being operated. These things often add weight to the cylinder assembly which is not fundamentally necessary for the performance of the simple duty of the cylinder to transmit power as linear motion.

In making this comparison, we have selected and gathered together cylinders of all types from which information was available. These data represent a wide variety of applications on airplanes, ranging from the heaviest present-day production bombing planes and four-engine transports down to single-seat fighters and training airplanes. The variety of applications represents about the normal spread between the heaviest and the lightest requirements. Since a great many of these applications have been made in the systems which operate at approximately 1000 psi, the maximum system pressure has been indicated in Fig. 5 with two curves – one curve represents the weight of a group of cylinders which were designed to operate at 1500-psi maximum system pressure. The other represents the weight of a group of cylinders designed to operate at 1000-psi maximum system pressure. Both curves represent approximately the maximum weights of either group.

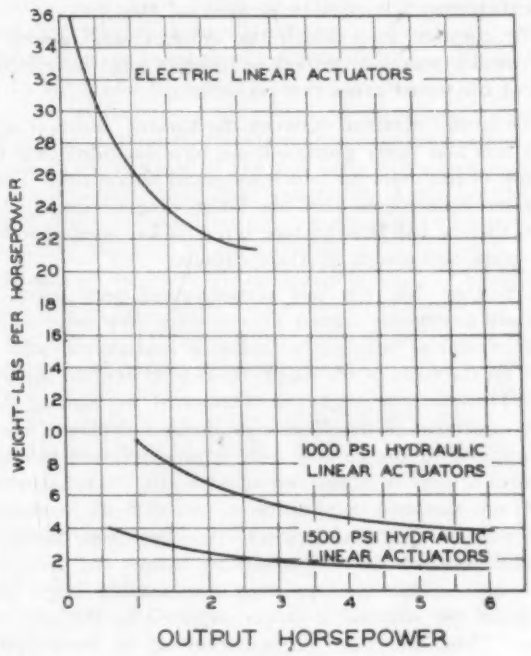


Fig. 5 – Comparison of hydraulic and electric linear-motion actuators

We have then compared these cylinders with a group of electrically operated screw-jack actuators. This comparison was made as follows:

For the electrical screw-jack actuator, we have taken in each case actuators designed to operate at a normal system voltage of 28 v. All high-voltage, high-frequency, alternating-current actuators have been left out of the discussion.

We have used in each case either actual time of actuation or the time of actuation as quoted by the manufacturer. It was assumed that the actuator was operating at a constant load equal to the normal operating load. The means by which this load was determined will be explained shortly.

The horsepower output used was then calculated on the basis of the above time, constant load, and the known stroke.

In making a similar calculation to obtain the horsepower output of the hydraulic actuating cylinders, it was necessary, in many cases, to assume a constant rate of flow into the cylinder in order to obtain the time of actuation. This was necessary because, in many cases, the actual time of actuation was not obtainable, and furthermore, in many cases, the load varied so widely over the stroke of the cylinder that it would have been impossible to have obtained a comparable time with constant load acting on the cylinder. Therefore, the time of actuation was determined as follows:

It was assumed that the rate of flow through a ⅜-in. line was constant at 3.5 gpm; through a ½-in. line, it was 6.0 gpm; and through a ⅝-in. line, it was 10.5 gpm. These represent average values in most present-day systems.

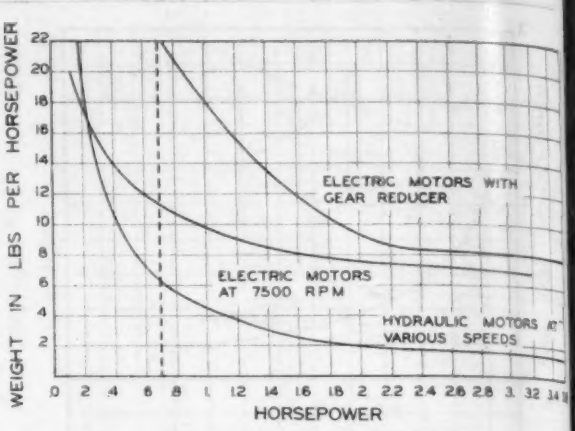
In the case of the electrical actuators, the operating loads were assumed constant at 1/3 of the maximum design load.

In the case of the hydraulic cylinders, the pressure at the cylinder was assumed to be 750 psi in a system having a maximum pressure of 1000 psi; and 1100 psi was used for systems having a maximum pressure of 1500 psi.

The constant load which the cylinder was capable of overcoming was determined by multiplying the effective area of the piston times this pressure.

The spread obtained between the various electrical actuators was also fairly great, but we have included only the lightest of the units for which we could obtain data. These units we believe contained the latest weight-reducing features, such as ball-bearing type screw jacks, integral reduction gears, and torque overload cutouts.

In neither case has any consideration been given to auxiliary emergency means of actuation. We believe that a comparison of emergency means of actuation should be made on the basis of the merits of each individual installation. However, it might be mentioned in passing that many cases exist in the present production airplanes where the lowering of the landing gear is accomplished through the force of gravity in the case of hydraulic cylinder installation, whereas such application is very difficult to accomplish with the electrical actuator. In other cases, auxiliary hydraulic systems or compressed air means are provided which are capable of supplying a reasonably large percentage of the amount of power supplied by the original system. Thus, the time of actuation is, in some cases, almost equivalent to the normal time, whereas most electrical actuators require emergency operation by means of a hand crank which in a system requires considerable



■ Fig. 6 – Comparison of hydraulic and electric rotary actuator

amount of power and is very much smaller than the normal means of actuation. However, the installation of emergency means of actuation is necessary with either type of accessory and the merits or faults of the emergency system depend largely upon the design of the mechanism being operated and upon the installation conditions; therefore, it is not practicable to make a general comparison.

The next series of power application devices to be considered are those which transmit rotary motion. We thus come to a comparison between electric motors and hydraulic motors. This represents a very simple comparison, since in both cases we can compare directly the power output based on the output torque at a given rpm.

The upper two curves of Fig. 6 show the weight of 28-v electric motors with and without reduction gears. It should be noted that in almost all practical applications, gears are a necessity, since electric motors must operate at extremely high speeds in order to bring the weight within reasonable limits.

The lowest curve of Fig. 6 shows hydraulic motors without reduction gears. A heavy vertical dashed line will be noted at 0.7 hp output. At horsepowers to the left of this line, the curve is to be compared with the higher of the two electric motor curves, since the hydraulic motor weight used is such that sufficient torque is available to eliminate the need for a reduction gear regardless of speed. At horsepowers to the right of the line, the curve is to be compared with the lower of the two electric motor curves since this represents horsepower output at higher speeds. This comparison favors electric motors since for many applications gear reduction is not necessary with hydraulic motors because of their relatively high torque output at low speeds and their wide speed range. In both cases continuous-duty ratings were used.

We have now discussed the comparative features of the various essential parts of the aircraft accessory power transmitting system. We have discussed each main subdivision of this system strictly from the standpoint of its power transmitting ability compared to its weight. This comparison is the one which weighs most heavily in designing a system to be used in a given airplane. Next, in order to prove out our comparison and to show how such a system is selected and compared for a complete airplane, we have chosen a medium-sized twin-engine attack-bomber type airplane which is now and has been in production for some time. This airplane has, at the present

ime, a hydraulic system which operates at a maximum system pressure of 1000 psi and which is used to operate two main landing gears, one nose wheel gear, bomb doors, and the brakes. The flaps on this airplane are operated electrically by means of a rotary actuator consisting of motor and gear box which drive the flaps through a system of torque shafts and small gear boxes. Since the brakes have been purposely left out of this discussion, we have eliminated from consideration all parts of the hydraulic system which are used solely for brake operation. In turn, we have added a hydraulic motor to drive the flaps in place of the electric motor; also connecting lines and fittings and the hydraulic selector valve required. We have analyzed the loads and the horsepower requirements for each part of the system based on the known times of extension and retraction of the gear, and the known time of opening and closing of the bomb doors and the operation of the flaps. We have included all of the fundamental and necessary parts of the system as actually installed in this airplane. We have then included a comparative electric system which will perform the same duty. Each component power generating, power transmission, power control, or power application device has been located in the same relative position in the airplane. Table 1 lists the weight of all the items included in the comparison.

Table 1 – Weight of Typical Hydraulic System for Medium Twin-Engine Bomber

Unit	Weight with Oil, lb	Number Required per Airplane	Weight per Airplane, lb
Main Landing Gear Cylinder	11.200	2	22.400
Nose Wheel Cylinder	9.150	1	9.150
Bomb Door Cylinder-Forward	7.300	2	14.600
Bomb Door Cylinder-Aft	7.080	1	7.080
Flap Motor	4.25	1	4.25
Pesco Pump	3.901	2	7.802
Regulator	2.741	1	2.741
Accumulator	13.590	1	13.590
Reservoir	31.91	1	31.910
Filter	2.082	1	2.082
Bomb Door Selector Valve	1.351	1	1.351
Landing Gear Selector Valve	2.021	1	2.021
Flap Selector Valve	1.20	1	1.20
Relief Valve	1.000	1	1.000
Relief Valve (Thermal)	0.125	2	0.250
Miscellaneous Fittings and Line Clamps	9.500	—	9.500
Lines and Fittings	—	—	40.19
		Total Weight	171.12

Table 2 lists the comparative electric system. It will be noted that in the case of the electric system, the battery has been eliminated. This was done because it is necessary to have electric energy to operate lights, radio, and so forth. Therefore, only fundamental requirements of the electric power generating system that would be necessary to perform the functions noted above have been included.

In the case of the hydraulic system, the weight of the hydraulic accumulator has been included; however, it is to be noted that this is not an absolutely necessary part of the fundamental hydraulic system shown since it is required only for the storage of energy necessary to operate the brakes on the ground while the engines are in operation. It, therefore, performs the function similar to that of the electric battery; namely, energy storage. It has been included, however, as a part of the hydraulic system to forestall any question that we have not considered all of the vital parts of the hydraulic system.

Even with all of these considerations, the total weight of the hydraulic system required to operate landing gear, nose wheel, bomb bay doors and flaps is 171 lb, whereas the 28-v electric system required to perform the identical function is 209 lb.

Table 2 – Weight of Typical Electrical System for Medium Twin-Engine Bomber

Unit	Weight, lb	Number Required per Airplane	Weight per Airplane, lb
Generator	26	2	52.
Main Landing Gear Motor	16.25	2	32.5
Nose Landing Gear Motor	15.5	1	15.5
Bomb Door Motor	24		24.0
Wing Flap Motor	16.75	1	16.75
Main Landing Gear Screw Jack	1.88	2	3.76
Nose Landing Gear Screw Jack	2.1	1	2.10
Bomb Door Motor Screw Jack	4	1	4.00
Reversible Generator Relay	2.5	2	5.0
Voltage Regulator	2.6	2	5.2
Toggle Switch	0.08	7	0.56
Main Nose Motor Relay	1.4	3	4.20
Bomb Door Relay	4.28	1	4.28
Wing Flap Relay	2.58	1	2.58
Wiring:			
Generator Wire	0.288	30 ft	8.64
Main Gear Wire	0.065	6 ft	0.39
Bomb Door Wire	0.368	10 ft	3.68
Wing Flap Wire	0.156	12 ft	1.88
Nose Gear Wire	0.065	10 ft	0.65
Generator Switch Wire	0.012	180 ft	2.16
Main Gear Switch Wire	0.009	72 ft	0.65
Nose Gear Switch Wire	0.009	10 ft	0.09
Bomb Door Switch Wire	0.009	32 ft	0.29
Wing Flap Switch Wire	0.009	36 ft	0.32
Generator Shunts	1.5	2	3.00
Junction Boxes (Firewall)	0.5	2	1.00
Junction Boxes (Motor)	0.5	5	2.50
Junction Boxes (Wing)	1.	2	2.00
Junction Boxes (Cockpit)	0.5	2	1.00
Circuit Breakers	0.4	4	1.6
Circuit Breakers	0.6	1	0.6
Limit Switches	0.15	10	1.5
Terminals	—	—	2.5
		Total Weight	208.88

No attempt has been made in this discussion to analyze such factors as service reliability, vulnerability to gunfire, or cost. Much has been said about these things in previous articles and discussions. They are so controversial in nature, so difficult to pin down to an accurate and fair analytical comparison, that they have been strictly avoided here.

We believe that these factors depend largely on the details of each individual installation, and that either system can be made to operate reliably provided that proved sound principles are followed throughout.

It should again be emphasized that no consideration whatever has been given to modern developments that have not yet been thoroughly proved in service. Many statements have been heard concerning high voltages, high frequencies, and high speeds in electric equipment. Much can also be said concerning high pressures and improved systems in hydraulics; for example, a new system is known which, performing all of the functions of the one discussed here excepting bomb door operation, in an airplane of almost the same gross weight, weighs approximately 60 lb complete, operating at only 1500-psi system pressure.

However, the only fair comparison that can be made is one based on facts proved by service experience.

It is conceded that electrical energy is required for lighting, radio, and other purposes. It should also be conceded that hydraulic power is much superior for such functions as brake control or other operations requiring delicate, smooth, and accurate control.

It should also be noted from the various figures presented herewith, that the weight advantage of hydraulic accessories becomes greater as the horsepower increases. This indicates that the greatest field of usefulness for the hydraulic system lies in applications requiring high horsepowers, such as are likely to be encountered in the larger airplanes of the future.

It seems self-evident that both systems should maintain their rightful place in aircraft, to be used where they can contribute most to the efficiency of the airplane as a whole.

PHYSICAL CHARACTERISTICS of INTERNAL-COMBUSTION EN

THIS paper consists principally of an experimental study of so-called "engine roughness" as it manifested itself in two widely different engines. The work was originally undertaken to test experimentally and, if possible, to correlate the several published theories, statements, and observations of this phenomenon. Although events not within the control of the writers terminated the work before it was completed, experiments had progressed far enough to be of some interest to automotive engineers. And it now seems that they will become increasingly interested in the "roughness" problem in the future because:

1. Recent advances in the technique of producing high-octane fuels make possible, without danger of knocking, much higher pressures than are handled in present automotive engines.

2. The developing demand for better fuel economy can probably be met best with light cars fitted with small engines of high specific output.

An examination of the automotive literature gives rather a confused impression of the meaning of the term "engine roughness." To some degree, the term has meant what the user intended it to mean on any particular occasion. In the present paper, the term refers to the objectionable increase in engine noise and vibration as the throttle is suddenly opened. It is recognized that this definition is far from perfect; for example, it should be specified perhaps that the engine be tuned as well as possible, that it be firing regularly in all of its cylinders, and that it be entirely free from knocking combustion. Moreover, it is conceivable that a given engine may be quite rough under part-throttle conditions and, therefore, that the noise and vibration may not increase perceptibly as the throttle is opened farther. Despite these possible deficiencies, it is believed that this definition of engine roughness is sufficiently broad to give the correct impression of the phenomenon that is being studied.

In general, the published information about engine roughness indicates that it may be controlled in two different ways:

1. By controlling the exciting force through proper combustion-chamber design.

2. By developing a basic engine structure which will withstand the exciting forces. There seems to be general agreement among automotive engineers that the pressure development during combustion is the exciting force and that the exciting force causes certain engine parts to vibrate and thus to generate sound. But, it is not always clear just what parts are vibrating or how these several vibrations are coupled together.

Consider now the characteristics of the exciting force or pressure development and its relation to engine roughness. Such factors as the maximum rate of pressure rise, maximum acceleration of the pressure rise, or combinations of these factors have been advanced as criteria of roughness. For example, Ricardo[1] at one time found the threshold of roughness to be 30 psi pressure rise per crankshaft deg. Taub in one of his most recent papers[2] has set up a roughness factor R^2V, where R is the ratio of the maximum rate of pressure rise during combustion to that found during compression and V is the percentage of the total pressure rise during combustion that the pressure has attained at the instant that the maximum rate of pressure rise is observed. Janeway[3, 4] computes a roughness shock factor which is the per cent increase of restoring forces in the mechanical system over the applied forces. The final decision about the relative merits of these criteria must await some sort of a fundamental study of the relationship between pressure development and the behavior of some particular part or parts of the engines which are responsible for the sensation of roughness.

The idea that a stiff engine structure contributes toward the smoothness of an engine has already been emphasized by the leading authorities on the relationship between engine roughness and combustion-chamber design. Ricardo[5] states that roughness "is dependent, on the one hand, on the beam stiffness of the engine as a whole and of the crankshaft in particular, and, on the other hand, on the rate of pressure rise." Janeway[6] also remarks, "While it is possible to obtain a fundamental freedom from shock by correct combustion-chamber design, the more sound the engine structure is, the greater will be the possible compression ratio without objectionable effect." These views should be very interesting to those who contemplate a redesign of their basic engine structures after the war, particularly so, because building a light engine of high specific output requires the maximum achievement with the minimum of material.

Additional evidence of a close relationship between

[This paper was presented at a meeting of the Detroit Section of the SAE, Detroit, Mich., Dec. 6, 1943.]

[1] See *The Automobile Engineer*, Vol. 19, July, August, 1929, pp. 257-261, 284-286: "Cylinder-Head Design," by Harry R. Ricardo.

[2] See SAE Transactions, Vol. 34, May, 1939, pp. 201-209: "What about the Engine?" by Alex Taub.

[3] See *SAE Journal*, Vol. 24, May, 1929, pp. 498-512: "Combustion Control by Cylinder-Head Design," by Robert N. Janeway.

[4] See *SAE Journal*, Vol. 27, October, 1930, pp. 428-437: "Combustion-Chamber Progress Correlated – Part 3, Janeway's Contribution Appraised," by Alex Taub.

[5] See *The Automobile Engineer*, Vol. 17, April, 1927, pp. 149-152: "Some Notes on Petrol-Engine Development," by Harry R. Ricardo.

[6] See *SAE Journal*, Vol. 25, October, 1929, pp. 384-387: Discussion of reference 3. Comment by Mr. Janeway on p. 385.

# of ROUGHNESS in ENGINES

by LLOYD WITHROW and ARTHUR S. FRY

Research Laboratories Division, General Motors Corp.

EXPERIMENTS have been run to find relationships between the primary causes and the outward effects of roughness as produced by a single-cylinder engine and by an 8-in-line engine.

This was done by recording simultaneously on a multi-element oscillograph oscillograms of the following phenomena:

1. Sound pressure near the engine.
2. Vibrations of the crankcase structure.
3. Lateral vibrations of the flywheel.
4. Lateral vibrations of the crankshaft.
5. Pressure development in the combustion chamber.

A study of the oscillograms reveals that the sensation of roughness in both of these engines is closely related to a shock type of excitation that is developed in the crankshaft-flywheel system.

In the single-cylinder engine this phenomenon was affected by any change in engine conditions which altered the development of the combustion pressures. Moreover, the sensation of roughness was greatly diminished by using a so-called "flexi-disc flywheel."

In the multicylinder engine, the shock type of excitation of the crankshaft-flywheel system as well as the sensation of roughness could be affected in two different ways:

1. By changing the characteristics of the pressure development.

2. By changing piston weight and thus altering inertial forces acting on the crankshaft.

engine roughness and the basic engine structure is supplied by the following three examples: Heldt[7] maintains that roughness consists of synchronous vibrations of the crankcase which arise from combustion-pressure forces and inertial forces of reciprocating engine parts. He recommends "beefing up" the crankcase to prevent the undesirable vibration. Still another method makes a well-known and highly successful V-8 engine operate smoothly. It consists in the use of a "flexi-disc flywheel," which alters favorably the vibrations of the engine. Finally, the manufacturers of one of the better known straight-8 engines found in 1941 that this engine became rough when cast-iron pistons were substituted for lighter aluminum ones.

These widely different approaches, each successful to a certain degree in the control of roughness in different engines, demonstrate that the problem has many ramifications. Indeed, they emphasize the need for experimental studies of the basic phenomena which relate to the apparent inconsistent outward manifestations of roughness and which will, it is hoped, furnish links between its causes and effects in the engine.

Present Experimental Approach

The object of the present experiments has been to reveal by a series of experimental steps the relation between the primary causes and the outward effects of roughness in two different engines. These were a single-cylinder test engine and a commercial 8-cyl in-line automobile engine. Each was rough under certain conditions; however, the changes of conditions leading to rough operation in the two engines were entirely different. The single-cylinder engine became rough when the location of the ignition spark was shifted from the side to the center of the combustion chamber. The 8-cyl engine, on the other hand, became rough when its usual aluminum pistons were replaced with cast-iron ones.

In this particular study of the primary causes and the outward effects of engine roughness, a special investigation has been made of the lateral vibrations of certain rotating engine parts, namely, the flywheel and crankshaft. Records of these vibrations were photographed during 1940-1941 with instrumentation that had previously been developed at the Research Laboratories Division, General Motors Corp. The instrumentation consisted of a condenser-type indicator and associated electrical equipment which was described by Martin, Grinstead and Frawley[8] for indicating

[7] See SAE Transactions, Vol. 38, February, 1936, pp. 47-54: "Engine Roughness – Its Cause and Cure," by P. M. Heldt.

[8] See AIEE Transactions, Vol. 60, 1941, pp. 513-523: "An Electrical Engine Indicator for Measuring Static and Dynamic Pressures," by E. J. Martin, C. E. Grinstead, and R. N. Frawley.

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and recording combustion-chamber pressures. In the present experiments, however, specially constructed condenser plates were substituted for the usual pickup unit. These insulated plates were installed in close proximity to the particular rotating part to be investigated.

Changes of the capacity of the condenser thus formed by the fixed plate and the moving engine part indicated in an oscillatory circuit any motion of the part in respect to the fixed condenser plate. The indicated vibration could then be viewed on a cathode ray screen or recorded permanently by means of a four-element mechanical oscillograph which was driven in synchronism with the engine.

By using several elements of the mechanical oscillograph, the lateral vibrations of the rotating-engine parts could be compared and accurately correlated in time with other dynamic properties of the engine performance. For example, these vibrations were recorded simultaneously with the pressure development in the combustion chamber and sound pressure in air at various points near the engine. The sound pressure was detected with a microphone, amplified and fed into one of the oscillograph elements. The first stage of amplification was accomplished in the General Radio noise level meter, Type 759A, which also served for measuring the general noise levels. Finally a few oscillograms were recorded to correlate these various events with vibrations in the crankcase. The latter vibrations were detected with a crystal pickup which was supplied us by Brush Development Co.

The remainder of the paper consists principally of a description of a series of oscillograms which correlate the events just mentioned and thereby tell part of the story of engine roughness. The data obtained with the single-cylinder engine will be considered first because these results are much simpler than those from the multicylinder engine. The reasons for this will become apparent in the following brief description of the engine.

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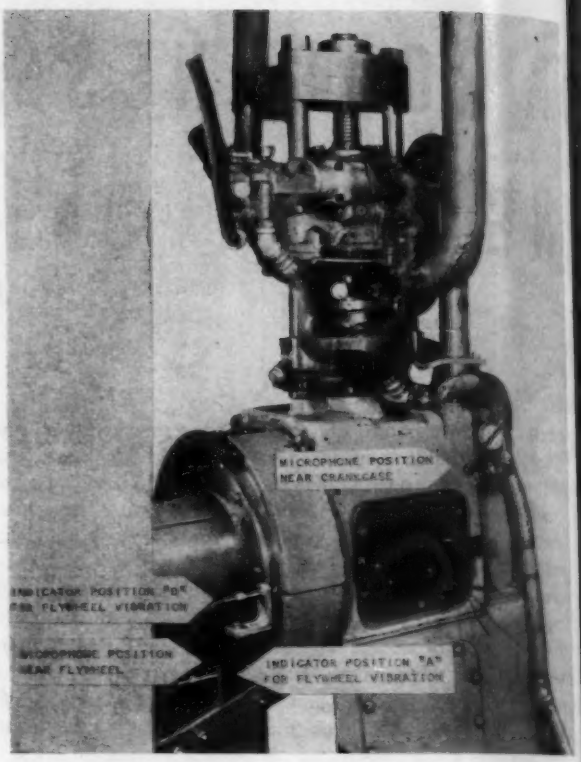
1. The engine is noticeably rough when operated at compression ratios of 7 to 1 or higher and with the spark near the center of the cylindrical combustion chamber.

2. Engine roughness is considerably decreased merely by changing the point of ignition from the center to the side of the combustion space.

3. The intermittent character of the sound emitted by this engine when running rough makes it ideal for determining the time in the cycle when the sounds characteristic of roughness are generated.

Preliminary experiments with the General Radio microphone and sound level meter showed that the general engine noise was loudest at two different places with respect to the single-cylinder engine. These two positions are indicated in Fig. 1. One of them was near the oil gage and the other was near the flywheel. To find the time in the engine cycle when the characteristic "thumping sound" of roughness occurred, a number of oscillograms were photographed to compare the pressure-time card with the variations in total sound pressure.



■ **Fig. 1 – Single-cylinder engine showing microphone positions for sound records and indicator positions for flywheel vibration records**

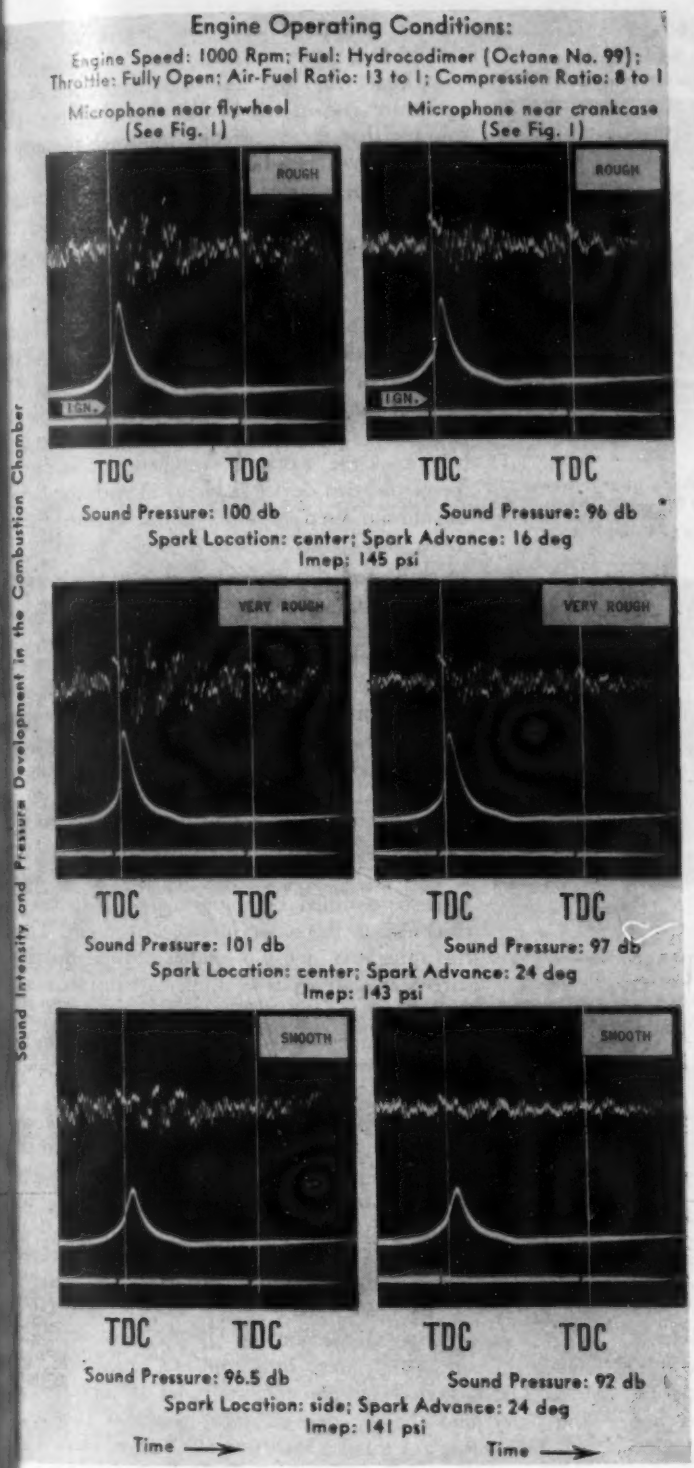
b. The Time of Occurrence of Sound Due to Roughness in the Single-Cylinder Engine – The data in Fig. 2 identify the "thump" which is characteristic of roughness in this engine with the increase in sound intensity that occurs at about the time of maximum pressure. The microphone was located near the flywheel for the three sets of records on the left side of the figure and near the crankcase for the three sets on the right. It will also be observed that the respective records differ from each other in respect to spark-plug location and spark advance. When the spark-plug position was changed from the center of the chamber to a point near the side wall, the intensity of the thumping sound was greatly reduced despite the fact that the spark advance had to be increased from 16 to 24 deg to obtain maximum power.

The effects of rough and smooth operation on the respective pressure-time cards and sound records can readily be discerned. In interpreting these data, allowance must be made for the effects of the ignition circuit on the resulting sound record. When this is done, the records in Fig. 2 may be summarized as follows:

1. The average sound pressure increases about 4 db during the transition from smooth to rough operation.

2. Just before ignition occurs, the sound intensities are approximately the same during rough and smooth operation.

3. During rough operation, the sound intensity increases sharply at about the time maximum combustion pressure develops.


Engine Operating Conditions:
Engine Speed: 1000 Rpm; Fuel: Hydrocodimer (Octane No. 99);
Throttle: Fully Open; Air-Fuel Ratio: 13 to 1; Compression Ratio: 8 to 1
Microphone near flywheel (See Fig. 1)
Microphone near crankcase (See Fig. 1)
ROUGH
IGN.
TDC
Sound Pressure: 100 db
Sound Pressure: 96 db
Spark Location: center; Spark Advance: 16 deg
Imep: 145 psi
VERY ROUGH
Sound Pressure: 101 db
Sound Pressure: 97 db
Spark Location: center; Spark Advance: 24 deg
Imep: 143 psi
SMOOTH
Sound Pressure: 96.5 db
Sound Pressure: 92 db
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Imep: 141 psi
Time

Fig. 2 – Correlation of sound emission and pressure development during rough and smooth engine operation

4. The total sound pressure near the flywheel exceeds that near the crankcase by about 4 db.

On the basis of these observations, the increase in sound pressure during the transition from smooth to rough operation appears to be due to the sharp increase in sound intensity which occurs shortly after ignition and which damps out during the remainder of the engine cycle. Since this phenomenon is repeated during each cycle, it is believed that it is the "thumping" noise that is heard during rough engine operation.

Since the "thumping" sound appears to be most intense near the flywheel, and since also the production of a sound is usually associated with a vibrating surface, it is of interest now to compare the total sound with the vibrations of the flywheel.

c. Relationship between Flywheel Vibrations and the Characteristic Sound of Roughness – Vibration of the flywheel was observed by means of an insulated condenser plate mounted close to its surface. Two of these pickup units are pictured in Fig. 1. Both were mounted on a rigid stand which was anchored solidly to the engine base. The condenser plate surface was about 7/16 in. in diameter and was set about 0.040 in. from the surface of the flywheel rim. Any variation in the gap separating the two surfaces produced an electrical response similar to that obtained with the condenser-type indicator. Calibration of displacement amplitudes was accomplished by attaching a metal strip of known thickness to the surface of the flywheel.

The data in Fig. 3 indicate that at least part of the sound of roughness (upper trace) arises from flywheel vibrations (lower trace). These vibrations were detected by means of the insulated condenser plate at position "A" in Fig. 1. The microphone was located within 2 in. of the condenser plate.

A detailed study of the data in Fig. 3, discounting the distortion of the sound record caused by the ignition circuit, reveals the following:

1. When the microphone is moved from a point 8 ft from the engine to the position near the flywheel, the sound pressure fluctuations become more pronounced.

2. When the microphone is near the flywheel, the sound pressure fluctuations increase sharply at the time the flywheel begins to vibrate. In fact, a general relationship can be observed between the low-frequency components of the sound record and the flywheel vibrations.

3. When the thumping noise is eliminated by changing the point of ignition from the center to the side of the combustion chamber, the maximum amplitude of the flywheel vibration decreases from 0.009 to 0.004 in.

Records similar to those in Fig. 3 were observed at sufficient points about the flywheel periphery to establish its mode of vibration. It was found that this vibration consists principally of a damped transient oscillation about a horizontal diameter; thus, the greatest displacements are observed at a point either directly above the centerline of the crankshaft or at a point directly below the centerline. The condenser plate was located at the latter point while recording the flywheel displacements in Fig. 3.

The oscillatory vibration of the flywheel has been found to be related to all of the engine variables which affect the roughness of operation. These include compression ratio, air-fuel ratio, spark advance, and location of the ignition

and recording combustion-chamber pressures. In the present experiments, however, specially constructed condenser plates were substituted for the usual pickup unit. These insulated plates were installed in close proximity to the particular rotating part to be investigated.

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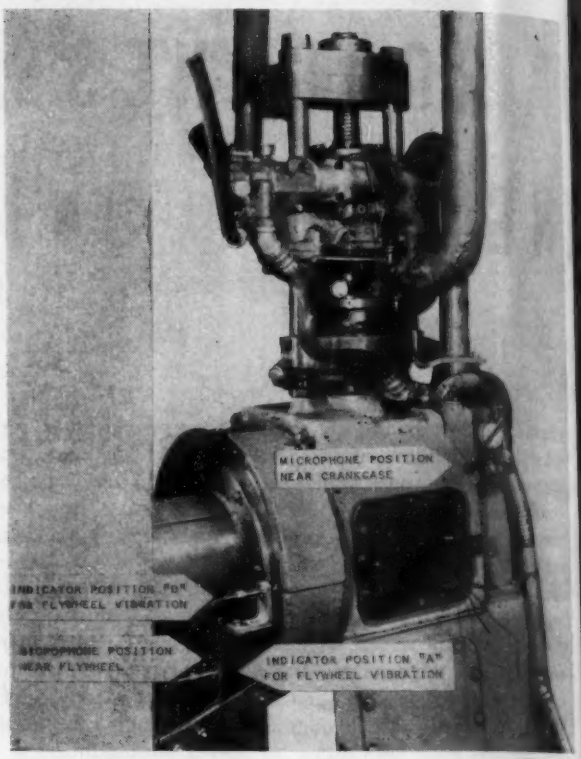

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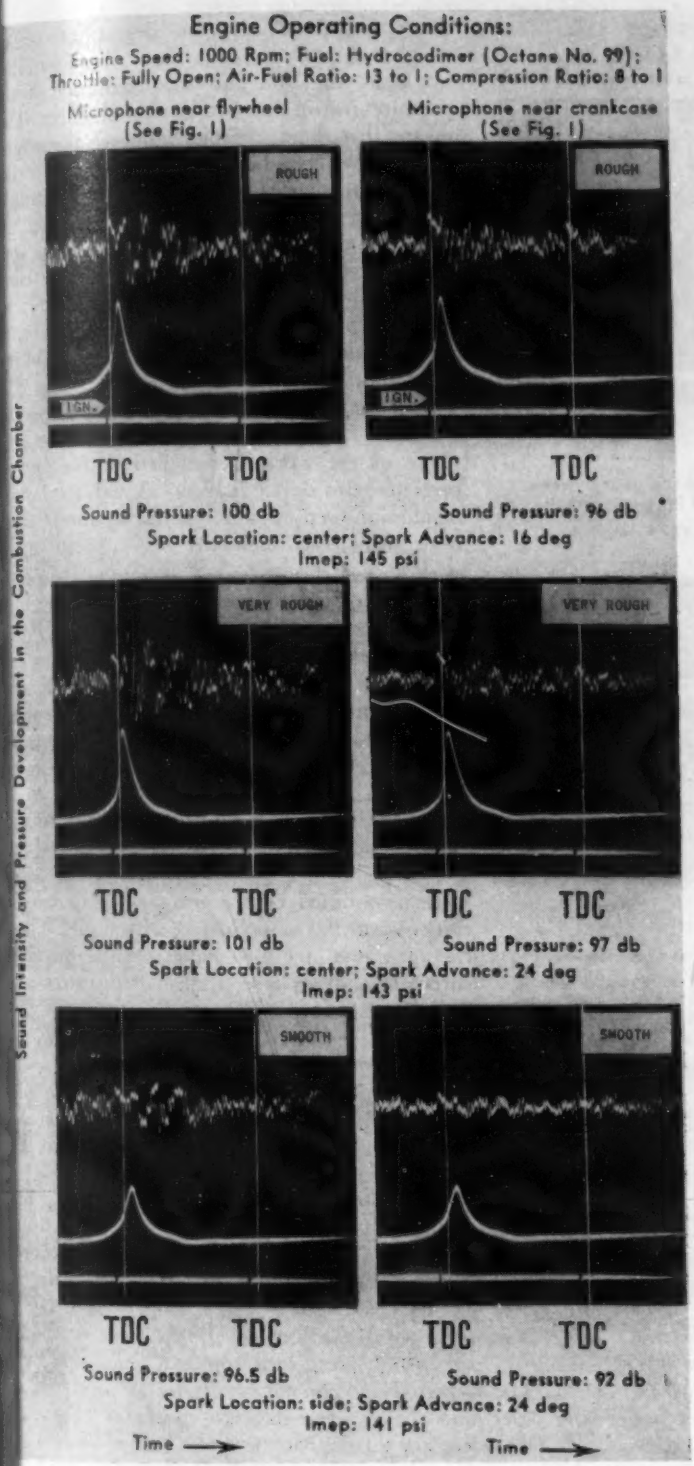
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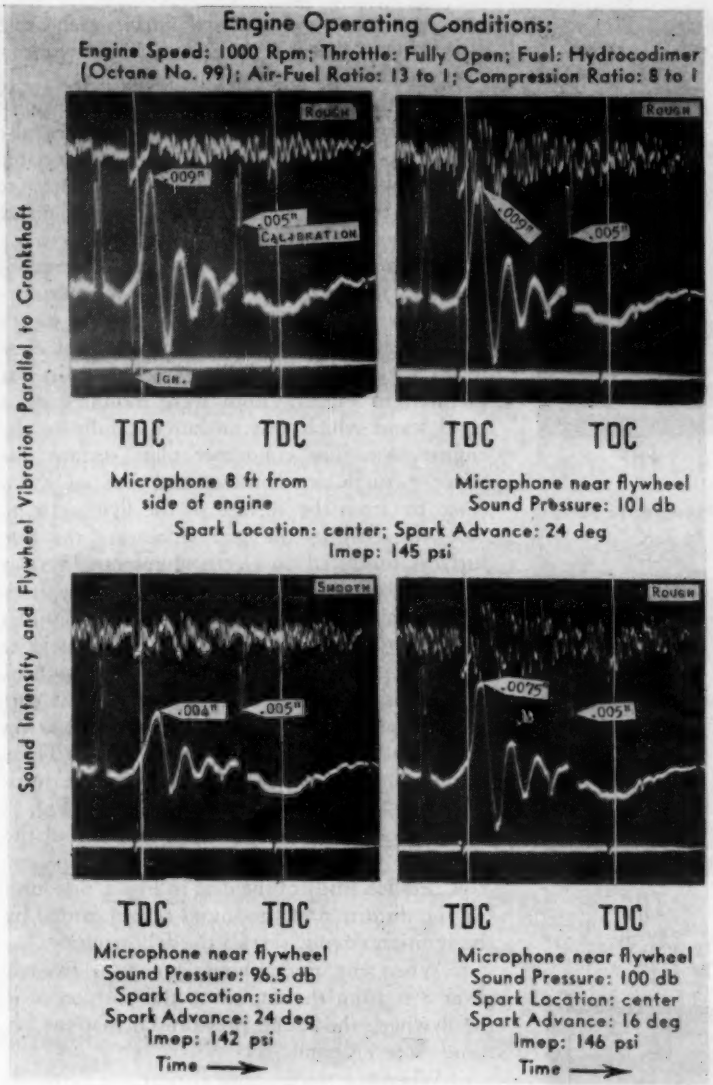


■ **Fig. 3 – Correlation of sound emission and flywheel vibration during rough and smooth engine operation**
Flywheel vibration observed at indicator position "A" at the bottom of the flywheel (See Fig. 1)

spark in the combustion chamber. Each change of an engine variable which resulted in increased roughness also increased the displacement amplitude of the flywheel vibration.

Maximum displacement of the flywheel is reached some time after maximum pressure develops. In other words, there is an appreciable phase difference between the applied force and the resultant displacement. This phase difference like the displacement amplitude increases with the degree of engine roughness at a given engine speed; also, it increases with engine speed when measured in seconds.

Changing the degree of engine roughness did not affect appreciably the frequency of the flywheel vibrations. In other words, the flywheel oscillation resembles a free vibration with damping. These characteristics of the transient vibration together with the fact that it damps out completely during the period between explosions indicate that the system is subjected to shock excitation during combustion; thereafter, it vibrates freely until the damping absorbs the vibrational energy.

The different engine variables which have been found to affect simultaneously the degree of engine roughness and the flywheel vibrations alter markedly the rate and the magnitude of the pressure rise in the combustion chamber. This means that these variables influence the magnitudes as well as the rates of application of the forces to the surface of the piston. It is natural then to speculate about the manner in which the forces on the piston affect successively the connecting rod and crankshaft as they are communicated to the flywheel. Hence, the next logical step in tracing the roughness disturbance is to examine the transverse displacements of the crankshaft.

d. Bending of the Crankshaft in the Single-Cylinder Engine – The apparatus used in this phase of the work is sketched in Fig. 4. In principle, this device is identical with the instrumentation used to observe the flywheel vibrations. It consists essentially of two parts:

1. A steel disc mounted on one of the crank cheeks in such a manner that its center coincides with the center of the crankshaft.

2. A bakelite sector mounted rigidly to the crankcase and carrying seven insulated condenser plates. The thickness of the semiannular space between the bakelite sector and the steel disc was adjusted to 0.035 in. with the engine at rest. Any disturbance in the engine which alters the distance from the steel disc to the condenser plate generates a signal in the oscillographs. The magnitudes of the disturbances were determined by soldering strips of known thickness to the outer surface of the disc. Thus, the apparatus in Fig. 4 makes possible the measurement of transverse crankshaft displacements in any one of seven different directions.

The flanges built on the steel disc were designed to throw the engine lubricating oil away from the annular space between the disc and the condenser units. With this construction the presence of oil presented no limitation to the performance of the indicators. Thus, the engine could be operated in a normal manner while observing the transverse vibrations of the rotating crankshaft.

The general nature of the crankshaft bending as revealed by detailed studies with the present indicators may be characterized as follows:

1. The principal deflection of the crankshaft occurs when the combustion pressure is applied.

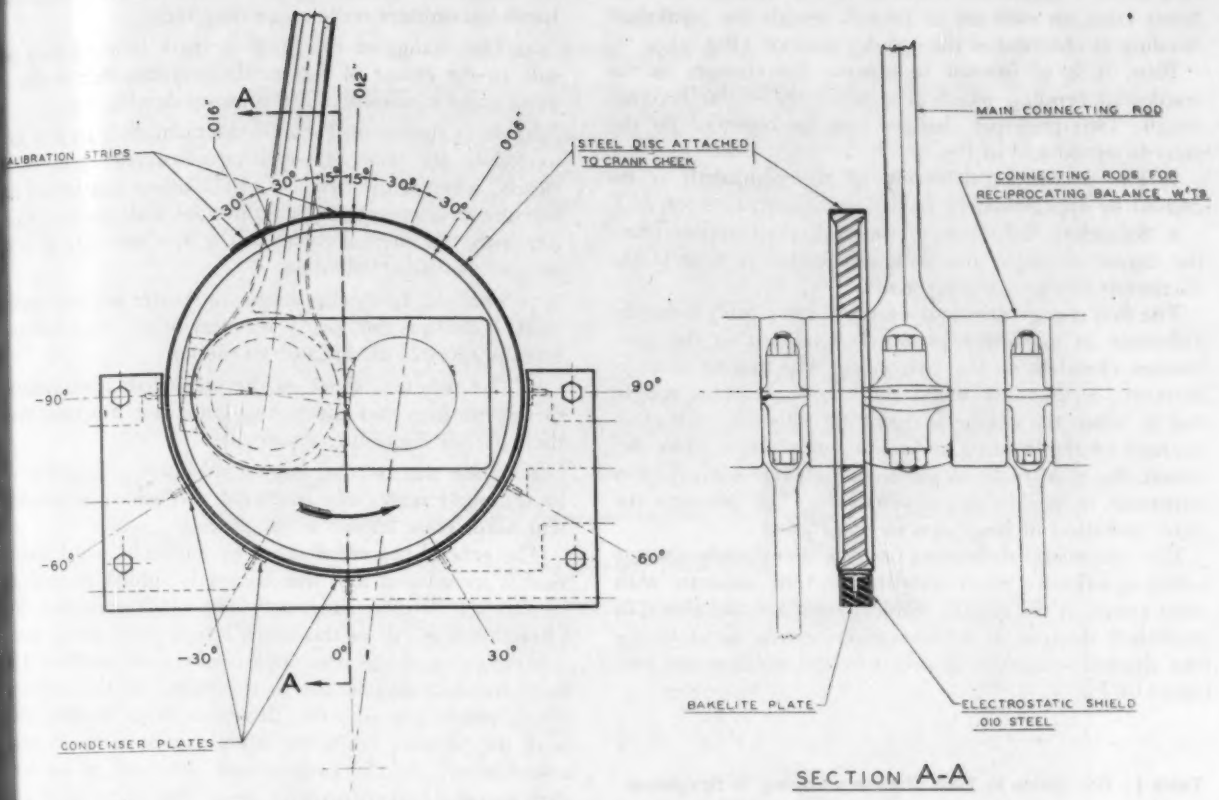
2. The largest displacement amplitudes are observed at the 0-deg and the −30-deg indicator positions (See Fig. 4). The reason for this is that the crankshaft is affected by the lateral thrust as well as the vertical forces.

3. The observed deflection under smooth engine operating conditions is approximately proportional to the applied load.

4. When the engine is rough, the nature of the bending is altered so that it resembles in certain respects the oscillations of the flywheel.

It is the latter characteristic that is of particular interest in connection with the problem of engine roughness.

e. Correlation of Crankshaft Bending, Crankcase Vibration, Combustion Pressure, and Flywheel Vibration – Fig. 5 correlates the crankshaft bending with the crankcase vibra-

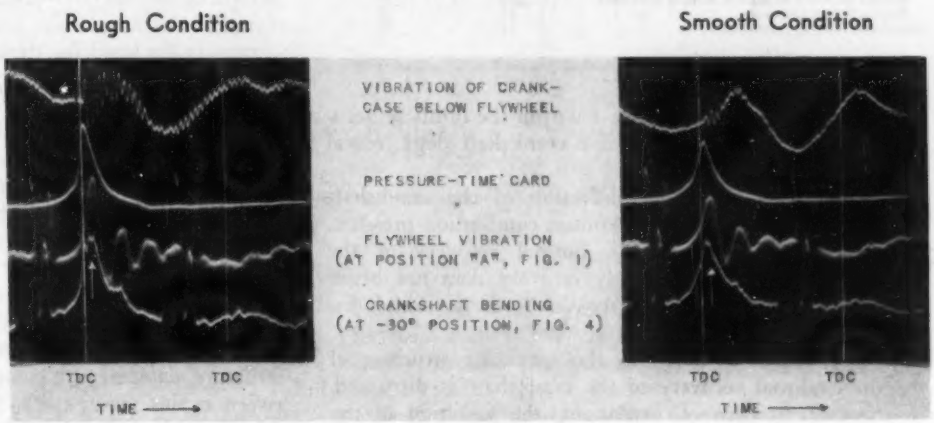


■ **Fig. 4 – Diagram of crankshaft bending indicator assembly showing condenser plate positions**

When an oscillator-detector unit is connected electrically to any one of the seven insulated condenser plates, any deflection of the crankshaft that alters the distance between the given condenser plate and the steel disc may be observed and recorded on an oscillograph. Thin metal strips soldered to the face of the disc provide a calibration of the deflection

ion, combustion pressure, nd flywheel vibration. All f these events were reorded simultaneously with he four-element oscillograph with rough and mooth engine operation. The engine operating conitions are tabulated at the ottom of the figure.

Consider now the arngements of the oscillorams on each set of recrds. The top trace is a cord from the Brush iezo-electric vibration ickup and represents the isplacement of the crankse below the flywheel uring 720 deg of revolution. The second trace ows the combustion pres-re development as indited by a condenser-type ressure unit which was stalled in the combustion hamber. The third trace cords the flywheel vibra-



Rough Operating Conditions	
Engine Speed, rpm	1000
Compression Ratio	8 to 1
Throttle	Fully open
Fuel	Hydrocodimer (Octane No. – 99)
Air-Fuel Ratio	13 to 1
Spark Advance, deg	24
Spark Location	Center
Imep, psi	142

Smooth Operating Conditions	
Engine Speed, rpm	1000
Compression Ratio	8 to 1
Throttle	Fully open
Fuel	Hydrocodimer (Octane No. – 99)
Air-Fuel Ratio	13 to 1
Spark Advance, deg	24
Spark Location	Side
Imep, psi	140

■ **Fig. 5 – Excitation of crankcase vibration correlated with pressure development flywheel vibration and crankshaft bending**

tion about a horizontal diameter of the flywheel, and the lower trace on each set of records reveals the crankshaft bending as observed at the −30-deg position (Fig. 4).

First, it is of interest to observe the changes in the crankshaft bending which arise when the engine becomes rough. Two principal changes may be observed on the records reproduced in Fig. 5:

1. The maximum deflection of the crankshaft is increased by approximately 20% (from 0.005 to 0.006 in.).

2. Secondary deflections of the crankshaft appear when the engine is rough, and these are related in time to the maximum flywheel displacements.

The first change noted appears to result simply from the difference in maximum pressures developed in the combustion chamber in the two cases. The second change, however, occurs only when the engine becomes rough; that is, when the engine is operating smoothly, this characteristic of the bending curve is entirely absent. For this reason, the appearance of the second change seems highly important in establishing a connecting link between the cause and effect of roughness in this engine.

The secondary deflections of the crankshaft assume added significance when compared in time sequence with other events in the engine. Table 1 compares the times, in crankshaft degrees, at which various events occur in the four dynamic processes recorded by the oscillograms presented by Fig. 5.

Table 1 – Correlation in Time of Events Leading to Roughness

Event Observed	Time of Occurrence During Rough Operation, deg ATDC	Time of Occurrence During Smooth Operation, deg ATDC
Initiation of Crankcase Vibration	15	25
Maximum Combustion Pressure	10	20
Crankshaft Bending Curve:		
Initial Maximum Deflection	10	23
Trough between Initial and Second Maximum	15	28
Begins Steady Recovery	31	32
Maximum Flywheel Deflection	32	35

The data presented in Table 1 within the limits of accuracy of these measurements (±1 crankshaft deg), reveal the following important facts:

1. The initial maximum deflection of the crankshaft coincides in time with the maximum combustion pressure.

2. Under rough conditions, normal recovery of the crankshaft is disrupted. Steady recovery does not begin until the flywheel begins to recover from its initial displacement.

3. Vibration is initiated in the crankcase structure at the time normal recovery of the crankshaft is disrupted. In the interval between explosions, the vibration of the crankcase structure has the characteristics of a free vibration with damping; consequently the oscillograms suggest that this vibration results from a shock type of excitation.

It will be observed in the oscillograms that under the smooth engine conditions there is also some evidence of slight disturbances in the crankshaft bending and in the crankcase. Evidently there was still a slight trace of roughness remaining even though, to the ear, the engine seemed to be operating smoothly.

f. Summary of Data from Single-Cylinder Engine – In summarizing the experimental evidence, the sensation of roughness in the single-cylinder test engine appears to result from the following chain of events beginning with the change of ignition spark location and resulting in the harsh intermittent noise called roughness.

1. The change of the ignition spark location from the side to the center of the combustion chamber results in more rapid combustion and pressure development.

2. As is shown in Table 1, the crankshaft itself is able to follow the more rapid pressure development rather closely, whereas the flywheel displacement lags behind the pressure development by several crankshaft degrees. Consequently, the inertial forces of the flywheel cause it to be set into oscillatory vibration.

3. Since the flywheel is unable to recover with the crankshaft, it disrupts the normal recovery of the crankshaft and actually reverses its direction of motion.

4. The resultant shock to the crankshaft is transmitted to the bearings and supporting crankcase structure; thus these parts are set into vibration.

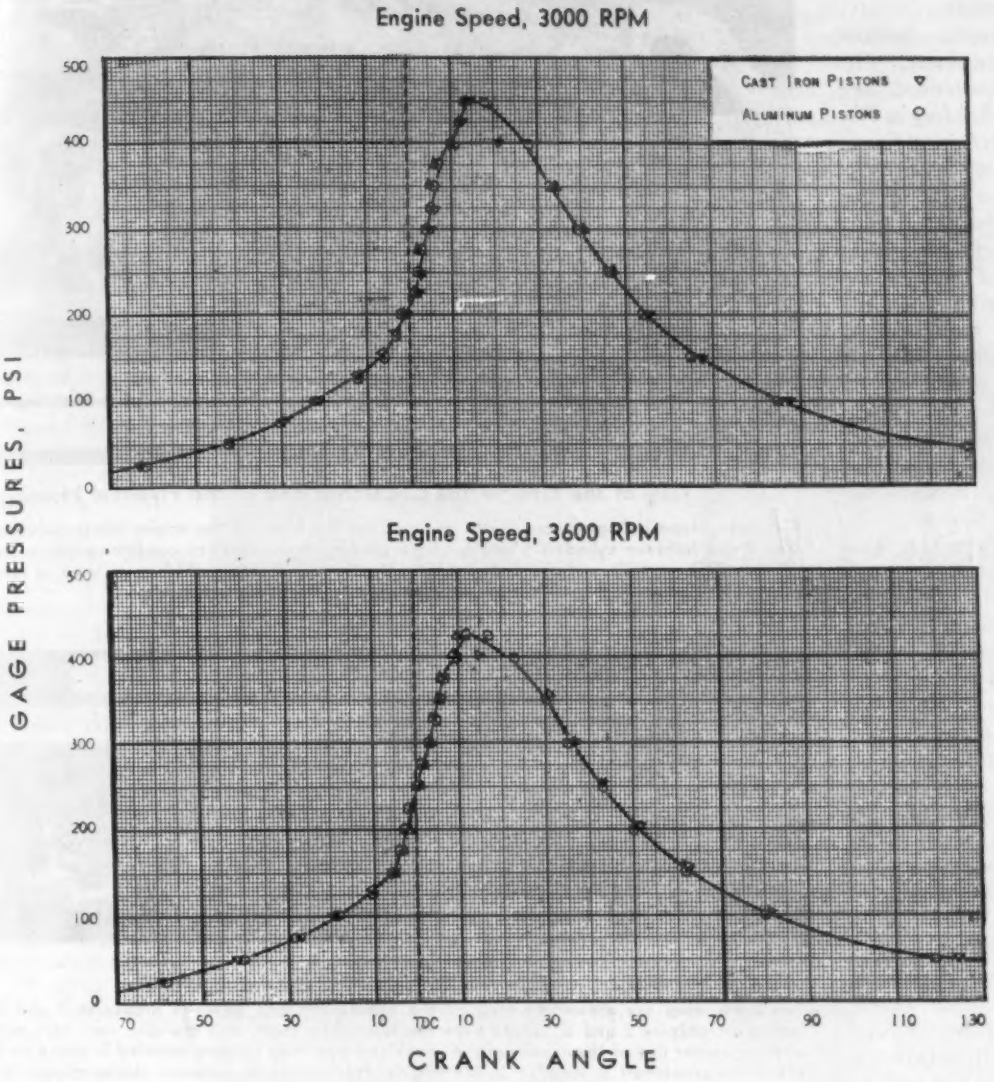
5. When the external engine structure is caused to vibrate, sound waves are generated and sensed as intermittent harsh noise known as roughness.

The general process of shock excitation by which roughness is introduced into this engine is similar in principle to that set forth by Janeway.[3] His conclusions were that "Roughness is due to the shock which gives rise to excessive deflection of the pressure-loaded engine members. The excessive deflection is due to the inertia of the deflecting mass, which prevents the deflection from keeping pace with the pressure when the latter rises at a rapidly accelerated rate." In the present case, however, it has been demonstrated experimentally how this takes place and which particular engine members are excessively deflected. With such specific information it is possible to prescribe mechanical alterations to alleviate the roughness. For example, in the present engine the results indicate that changes should be introduced to alter the vibration characteristics of the flywheel; moreover, a flexible type of flywheel did reduce the roughness under the engine conditions maintained for these experiments.

It should be pointed out that vibration was observed in other external engine parts as well as in the crankcase. For instance, vibration of the head structure was observed on top of the vertical threaded stud which, as may be seen in Fig. 1, supports the overhead camshaft mechanism and cylinder head. This vibration, however, differed in certain respects from that of the crankcase. The initial deflection of this structure was a direct reaction to the pressure development in contrast to the crankcase vibrations which were initiated by the reactionary forces of the flywheel on the crankshaft. The resultant transitory vibration of the head structure damped out completely during the interval between explosions in the same manner as did the flywheel vibration. But, since the sound emission in the vicinity of the head was of low intensity in comparison with that near the crankcase, the head structure vibration apparently did not contribute appreciably to the sensation of roughness.

■ Roughness in 8-Cyl, In-line Engine

The experiments to be described in this section of the discussion were performed on an 8-cyl automobile engine. As has been previously stated, this engine became rough when its usual aluminum pistons were replaced by heavy cast-iron pistons. The resulting sensation of roughness consisted of excessive noise and vibration when the throttle was suddenly opened at car speeds over 65 mph.



■ Fig. 6 – Effect of transition from aluminum to cast-iron pistons upon full-throttle pressure-time cards

In the light of current theories concerning the origin of roughness, there were two possible explanations for the introduction of roughness into this engine:

1. It was conceivable that the change in piston composition altered the pressure development so as to produce "rough combustion."

2. The increase in the inertial forces due to the heavier pistons could result in an objectionable increase of vibration.

Experiments were, accordingly, carried out to find the causes for the increase in rough operation.

a. Effect of Piston Composition on Combustion Pressures – One of the first important experiments run to find the relationship between piston composition and engine roughness was the determination of the effects on pressure development of changing from aluminum to cast-iron pistons. A marked effect was conceivable because of the differences in the heat conductivities of the two metals as well as because of a widely held belief that engine roughness and high rates of pressure rise are synonymous.

Fig. 6 reveals the results of this experiment. These quantitative pressure measurements were made with a balanced-pressure instrument on a 6-cyl engine having the same compression ratio and general combustion chamber arrangement as the 8-cyl engine in question. The data were taken while operating the engine at full-load at 3000 rpm and 3600 rpm. The latter speed is in the range in which roughness appears while using cast-iron pistons in the 8-cyl engine. These data demonstrate that changing the piston composition has no appreciable effect on the pressure development if all other conditions are held constant. Thus, it seemed highly probable that the rough operation of the 8-cyl engine was related in some way to the 26% increase in maximum mechanical forces which act on the crank-

shaft when the aluminum pistons are replaced with cast-iron pistons.

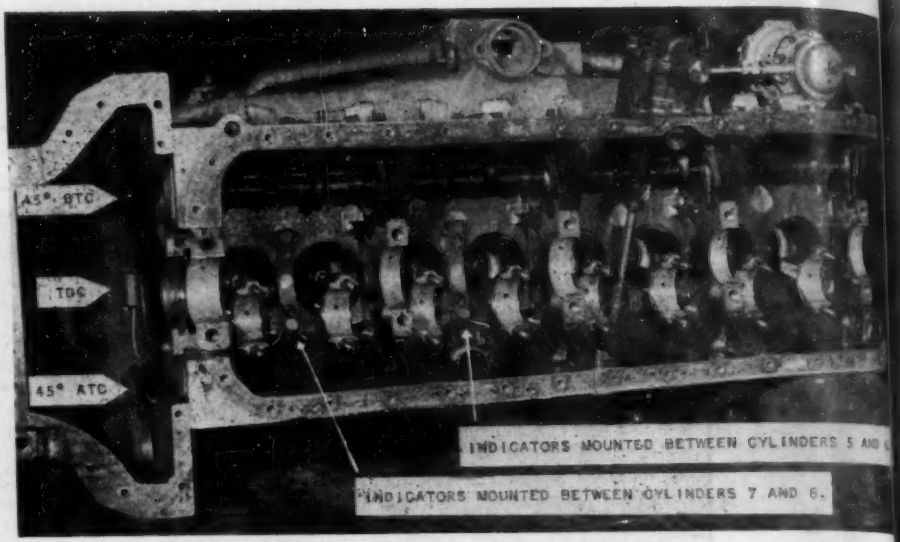
b. Relationship between Engine Roughness, Flywheel Vibrations, and Crankshaft Bending as Produced by Mechanical Forces and Gas Loads in the 8-Cyl Engine – The data in this section of the paper show what happens to the crankshaft-flywheel system in the 8-in-line engine when the degree of roughness is altered either by changing the piston weight or by changing the spark-plug position. These data consist of records of pressure development, flywheel vibration, and lateral crankshaft bending which, like the records in Fig. 5, were photographed simultaneously. The instrumentation used for detecting the lateral crankshaft bending and flywheel vibrations will now be described.

Fig. 7 pictures the special vibration indicators that were used for this phase of the work. They were patterned after those employed on the single-cylinder engine. As may be seen in the upper part of Fig. 7, condenser plates were provided to detect crankshaft bending in three different directions between cylinders 5 and 6 as well as between cylinders 7 and 8. The three directions were:

1. Vertical (TDC location, so-called).
2. 30 deg in advance of vertical in respect to crankshaft rotation (30 deg BTC location).
3. 30 deg in retard of vertical (30 deg ATC location).

The lower part of Fig. 7 shows the two discs mounted on the crankshaft. These discs were installed observing every precaution to avoid changing the original properties of the crankshaft. Their indicating surfaces were machined finally with the crankshaft installed in the engine to ensure satisfactory base lines for measuring the deflections.

Flywheel vibration could also be observed in three different directions. At the upper left of Fig. 7, three arrows point to the respective condenser plates in the flywheel housing. Note that they are secured rigidly to the engine block as also are the crankshaft bending indicators. Motion of the moving engine parts is thus observed in respect to the engine block which is considered as a reference base.



View of the Roof of the Crankcase and of the Flywheel Housing

Condenser-type indicators are shown mounted on the base of the engine block between cylinders and 8 and between cylinders 5 and 6. Each set has three separate condensers which are located 30 deg BTC, at TCD and at 30 deg ATC. The three indicators which were used to record flywh vibration may be seen in the flywheel housing.

View of the Crankshaft and the Flywheel

Two brass discs are shown mounted on the crankshaft; one between crankpins 7 and 8, the oth between crankpins 5 and 6. These were brazed to the shaft, and the disc rims were machined tru with the center line of the crankshaft. A machined ring may be seen sweated in place on the flywhe When the crankshaft is installed in the engine, the clearances between the condenser indicators o the base of the engine block and the discs on the crankshaft are about 0.040 in.

■ Fig. 7 – Photographs showing the indicators used to record crankshaft vibration and flywhe vibration

■ Typical Oscillograms

Fig. 8 presents four oscillograms which illustrate th general nature of the crankshaft and flywheel vibration as well as their interrelationships. The oscillograms on th left were recorded under motoring conditions and the tw on the right while operating at full load. Each oscillogra contains four different records. These are numbered alon the right-hand edges and are identified in the "Key t oscillograph records."

Note that upward trends in oscillograph records 1 and indicate upward motion of the crankshaft, and that up ward trends in oscillograph record 3 indicate that the fr of the flywheel is tilting toward the end of the cylinde block. All of these records of crankshaft and flywhe vibrations were taken with the indicators at the so-calle TDC locations. It was found that the deflections observ

these locations were the greatest and consequently were of primary interest.

The broken lines superimposed on the records of crankshaft and flywheel vibration form the base lines of zero displacement. The base-line positions were determined from records taken while motoring the engine at low speeds with the spark plugs removed; hence, the base lines are the records obtained when no appreciable forces were applied to the system. The principal deviations from the base lines have been marked with letters in Fig. 8. The significance of each event denoted by a letter is indicated in the "Key to the events shown by the records" at the bottom of the figure. Each event is also discussed briefly in the following paragraphs.

Event "A" – The deflections marked "A" are most prominent on the records taken at the higher engine speeds – 3500 rpm. Since the gas loads were negligible when motoring with the spark plugs removed, the deflections marked "A" are considered to be due to mechanical forces acting in the crankshaft.

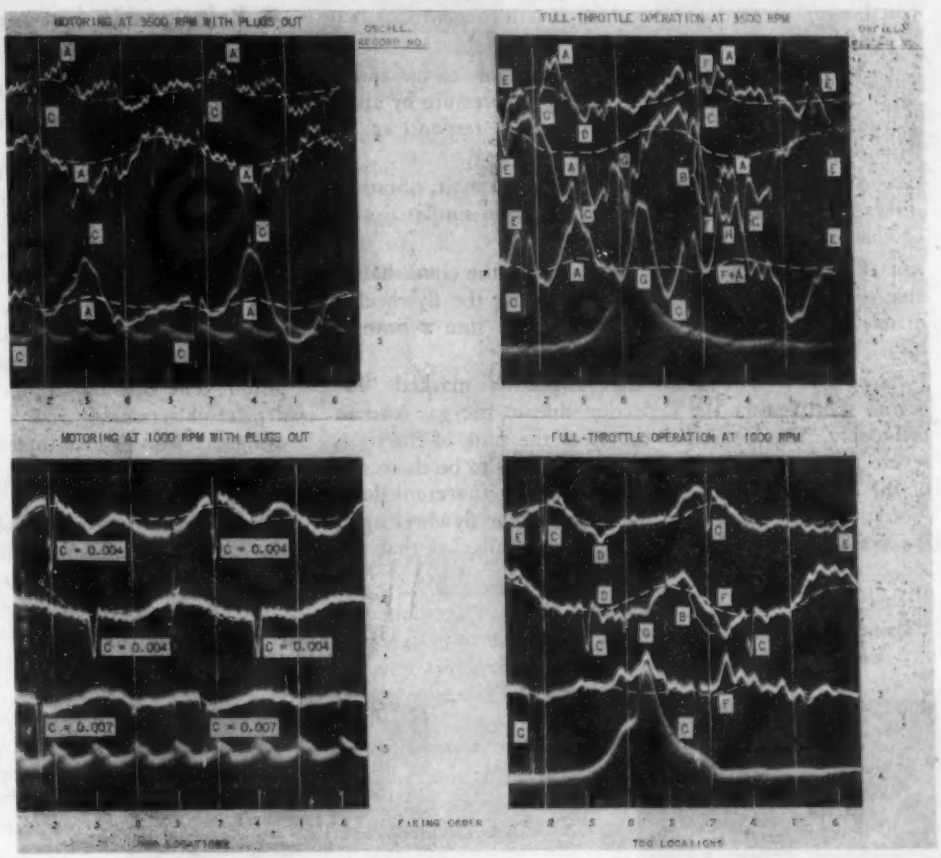
Event "B" – Symbol "B" marks the time in the engine cycle when the crankshaft was deflected laterally between cylinders 7 and 8 with maximum velocity. As will be shown later, changes in the degree of engine roughness alter the magnitude of this event.

Event "C" – The letter "C" labels the calibration marks which were produced by means of calibration grooves cut into the indicating surfaces of the crankshaft discs and flywheel ring. These were of known depths. Their respective magnitudes are given in the lower left oscillogram of Fig. 8.

Event "D" – Symbol "D" denotes the crankshaft deflections produced by the gas load in cylinder 5. As would be expected, this deflection is most pronounced in oscillograph record 1 under firing conditions. However, it is important to observe also that at 1000 rpm, the gas load in cylinder 5 bends the crankshaft downward between cylinders 7 and 8.

Event "E" – The letter "E" marks the deflections that were produced by gas loads in cylinder 6. At 3500 rpm, these gas loads deflected the crankshaft very prominently and in the same direction both between cylinders 7 and 8 and cylinders 5 and 6; in addition, the flywheel deflected inward. These observations are most interesting because in crankshaft development the main bearings are very frequently considered to be nodal points.

Event "F" – "F" labels the crankshaft and flywheel deflections resulting from the gas loads in cylinder 7. These deflections appear most prominently on oscillograph records 2 and 3, though record number 1 taken at 3500 rpm indi-



Key to the Oscillograph Records

1. Vertical crankshaft vibration observed between cylinders 5 and 6.
2. Vertical crankshaft vibration observed between cylinders 7 and 8.
3. Horizontal flywheel vibration observed at the TDC-position.
4. Pressure development in cylinder No. 8.
5. Time of ignition for each cylinder.

Key to Events Shown by the Records

A. Principal deflections resulting from mechanical forces alone.

B. Time of maximum bending velocity of the crankshaft.

C. Calibration marks.

D. Deflections resulting from explosion pressures in cylinder 5.

E. Deflections resulting from explosion pressures in cylinder 6.

F. Deflections resulting from explosion pressures in cylinder 7.

G. Deflections resulting from explosion pressures in cylinder 8.

H. Deflection resulting from recoil action of the flywheel.

■ Fig. 8 – Typical oscillograms that illustrate the general nature of the crankshaft vibrations and their relationship to flywheel vibration – 1941 production engine using cast-iron pistons

cates that the full-throttle gas loads in cylinder 7 also bend the crankshaft downward between cylinders 5 and 6. Note that the flywheel deflection "F" is superimposed upon a deflection due to mechanical forces "A." This resultant large deflection is labeled "F + A." The consequences of this combination of deflections, from two different causes, will be shown later to have a direct bearing on the roughness of operation of this engine.

Event "G" – The crankshaft and flywheel vibrations (labeled "G") that were excited by the gas load in cylinder 8 appear distinctly on oscillograph records 2 and 3 under firing conditions. When these vibrations are compared with the record of pressure development in cylinder 8 (oscillograph record 4) several important observations may be made:

1. At 3500 rpm, the maximum deflection of the crankshaft lags the development of maximum pressure by about 3 crankshaft deg; whereas, the maximum response of the flywheel lags by about 34 deg.

2. At 1000 rpm, the record of the crankshaft vibrations shows little or no evidence of free vibration similar to that on the record of flywheel vibration.

3. The above observations indicate that the crankshaft is able to follow the applied force, but that the flywheel is unable to follow and consequently is set into a state of transient vibration.

Event "H" – The crankshaft deflection marked "H" occurs shortly after the deflection due to the gas load in cylinder 7. This deflection occurs at the time of the most severe flywheel deflection and thus appears to be the result of the recoil action of the flywheel upon the crankshaft. Note that this evidence of interaction of the flywheel upon the crankshaft in this engine is very similar to that observed in the single-cylinder test engine under rough conditions.

Other Events – The few remaining deviations from the base lines have not been related to major engine phenomena. Some of these unexplained deviations may result from the clearances allowed in the main crankshaft bearings; others are produced by high-frequency oscillations that are caused by extraneous electrical interference. In any case, all of the events not marked with letters have been disregarded in the following observations of the relationship between engine roughness and (1) piston weight and (2) combustion pressure.

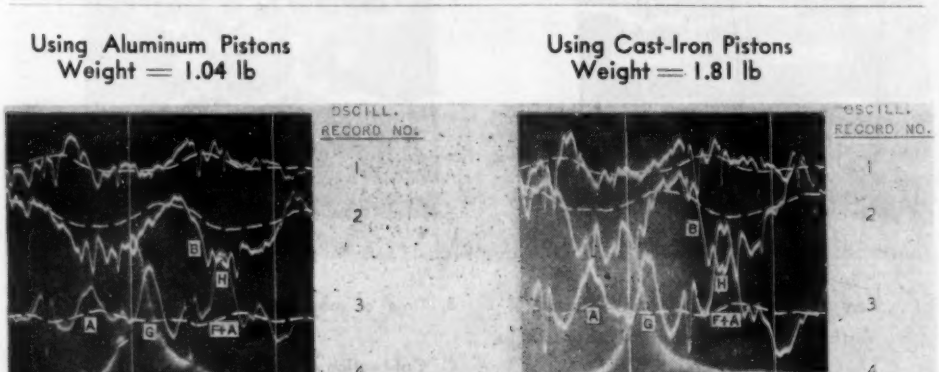
■ Relationship between Piston Weight and Engine Roughness

The relationship between piston weight and roughness was investigated by comparing the vibrations under engine conditions which, except for piston composition, were similar in all respects. The increase in piston weight affected the vibrations most markedly in the speed range where the engine was rough, namely, at speeds of 3500 rpm and above. The nature of these effects will now be pointed out.

The oscillogram on the left side of Fig. 9 depicts acceptably smooth operation while running the engine at 3500 rpm and using aluminum pistons that weighed 1.04 lb per piston. The oscillogram on the right side of this figure depicts objectionably rough operation at 3500 rpm while using cast-iron pistons that weighed 1.81 lb per piston. To facilitate the examination and intercomparison of the respective records, some of the events have been lettered following the convention adopted for Fig. 8. It will be observed that the degree of roughness affects the events on records 2 and 3 more than those on record 1. Apparently then, the crankshaft vibration between cylinders 5 and 6 (oscillograph record 1) was not affected appreciably by the change in piston weight.

Compare now oscillograph records 2 and 3 in Fig. 9. It will be observed that increasing the degree of roughness affected markedly the events labeled "A." These deflections, it will be recalled, are due to mechanical forces acting on the crankshaft. The transition from aluminum to the heavier cast-iron pistons increased the amplitudes of these deflections by approximately 50%.

In contrast with this effect, the amplitude of the flywheel deflection resulting from the gas load in cylinder 8 (event G) was not changed appreciably by increasing the piston weight. Such a result might



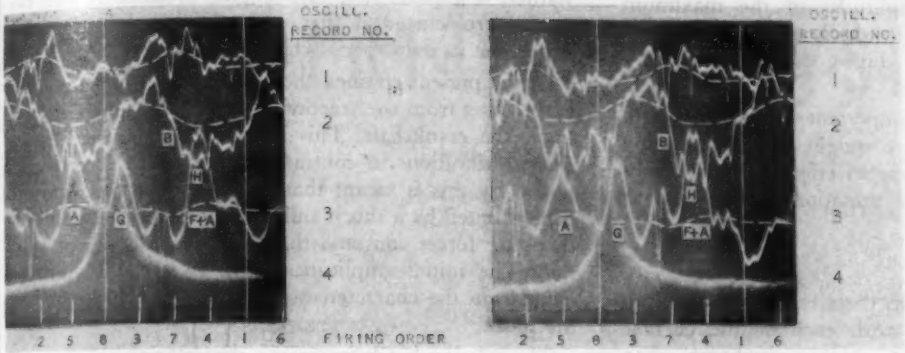
Key to the Oscillograph Records

1. Vertical crankshaft vibration observed between cylinders 5 and 6.
2. Vertical crankshaft vibration observed between cylinders 7 and 8.
3. Horizontal flywheel vibration observed at the TDC-position.
4. Pressure development in cylinder No. 8.

■ Fig. 9 – Effects of piston weight on crankshaft and flywheel vibrations – 1941 Olds 8 engine operating at full-throttle at 3500 rpm

Smooth Head
Plugs over Center of Intake Valves

Production Head
Plugs between Exhaust and Intake Valves



TDC Locations

TDC Locations

Key to the Oscillograph Records

1. Vertical crankshaft vibration observed between cylinders 5 and 6.
2. Vertical crankshaft vibration observed between cylinders 7 and 8.
3. Horizontal flywheel vibration observed at the TDC-position.
4. Pressure development in cylinder No. 8.

▪ Fig. 10 – Effects of spark-plug location on crankshaft and flywheel vibrations – 1941 Olds 8 engine operating at full-throttle at 3500 rpm, using cast-iron pistons

be expected, of course, on the basis of the results of the investigation of the relationship between piston composition and combustion pressures.

Increasing the piston weight increased by 30% the amplitude of the flywheel deflection (F + A) which results from the combined effects of the mechanical forces and the gas load in cylinder 7. Inasmuch as the gas load is independent of piston composition, the 30% increase in deflection (F + A) is caused by the increase in component A from the mechanical forces. In connection with this event, it is interesting to observe that at speeds below 3500 rpm where the roughness was not so objectionable, events F and A were not in phase. In other words, the phase lag between the application of the gas load in cylinder 7 and the maximum response of the flywheel was such that the flywheel responded to this excitation before it reacted to the mechanical forces.

Intercomparison of events H on record 2 of the respective oscillograms reveals the effect of the severe flywheel deflection (F + A) upon the bending of the crankshaft between cylinders 7 and 8. This secondary deflection, caused by the flywheel recoil action, increased markedly in amplitude when the piston weight was increased.

Another outstanding effect of the shock excitation of the crankshaft is shown by event B on record 2. A comparison of this event on the two oscillograms reveals that the heavier pistons increased the bending velocity of the crankshaft by approximately 20%. In summarizing, the processes by which roughness results from the transition from aluminum to cast-iron pistons may be stated as follows:

1. Increased inertial forces due to the heavier pistons cause greater deflections of the crankshaft-flywheel system.
2. Excessive deflections of the flywheel occur when the deflection caused by the gas load in cylinder 7 combines with the increased deflection due to the mechanical forces.
3. On account of the excessive deflection of the flywheel arising out of this set of circumstances, normal recovery of the system is upset and shock H is applied to the crankshaft.

The final result of these processes – shock to the crankshaft system – is very similar to the roughness phenomena in the single-cylinder test engine. The steps leading to the shock are, however, different in that the activating forces arise from somewhat different sources. The conclusion then appears to be that any process which leads to disruption of normal recovery of the dynamic system from deflections may cause an engine to become rough.

■ Effect of Altering Pressure Development

To study the relationship between engine roughness and pressure development, the combustion process was changed by moving the spark plug from a point midway between the valves to a point directly above the intake valve. This change in spark-plug location altered the sensation of roughness so that the engine was considered to be passably smooth even when operating with cast-iron pistons. Maximum power, however, was reduced by 2%. No other undesirable operating features were observed with the so-called smooth head while making these experiments.

The changes in vibration resulting from the shift in spark-plug location are revealed by the oscillograms in Fig. 10. While photographing these records the engine was fitted with cast-iron pistons and run at 3500 rpm. The combination of circumstances resulting in decreased roughness with the "smooth head" becomes apparent when the oscillograms are compared. The flywheel deflection G due to the gas load in cylinder 8 was reduced by about 10% by the shift of plug location. At the same time, deflection A resulting from mechanical forces remained unchanged. Hence the excessive deflection F + A and the shock H to the crankshaft were reduced, in the "smooth head" case, by a reduction of the component F resulting from the combustion pressures in cylinder 7.

■ Summary: 8-In-Line Engine

The physical characteristics of roughness in the present 8-cyl engine revealed by these experiments may be stated as follows:

1. Rough operation is intimately associated with the maximum displacement and velocity amplitudes of the

crankshaft deflections between cylinders 7 and 8. In the present experiments, the increase in the amplitudes of these deflections ranged from 25 to 50% during the transition from smooth to rough operation.

2. The increase in the amplitudes of the maximum flywheel deflections resulting in shock to the crankshaft amounted to from 10 to 30% during the transition from smooth to rough operation.

3. Finally – and this is most important – inducing roughness by altering either the piston weight or the spark-plug location produced the same type of effects on the crankshaft-flywheel system and on the sensation of roughness.

■ General Discussion

The experimental observations described herein are consistent, within certain limits, with each of the current theories and statements about engine roughness. For example, the data confirm the general observation that roughness can be altered considerably by varying the character of the combustion process. At the same time it is clear that the degree of roughness that is associated with a particular combustion-chamber design depends to a large extent on the mechanical properties of the engine structure. A thorough understanding of the behavior of the latter when subjected to shock excitation is especially important in high-compression engines because the small clearance volumes limit the possible alterations of combustion-chamber design.

For the study of roughness, the instrumentation described in the present paper affords a considerable advantage because it can be used under actual operating conditions without hampering the normal functions of the engine parts. On the other hand it is always possible to question whether the stationary pickup plate or the rotating engine part moves in space when interpreting the records of the lateral vibrations. In case of the flywheel vibrations, the use of several pickup units shows that the observed effects are not due to end play of the crankshaft and that they are actually due to a tilting action of the flywheel.

Detecting the lateral vibrations of the crankshaft is somewhat more difficult than picking up the tilting action of the flywheel. This is particularly true for the multicylinder engine because care must be used in choosing positions for the stationary condenser plates as well as in mounting the discs on the crankshaft. In case of the discs, it is essential that their centers coincide with the centerline of the crankshaft regardless of the nature of the lateral vibrations. And, as well can be imagined, it is not easy to mount a disc securely to a limber crankshaft without altering its mechanical characteristics. For this reason, it may be desirable, in many cases, to supplement the types of records described in the present paper with others from a suitable type of strain gage. The latter instrument, if applied under actual operating conditions, should be especially valuable for studying local bending of the crankshaft at various points and in different directions as it rotates in the engine.

■ Need for Further Tests

The need for further experimentation is apparent from the fact that the physical conditions giving rise to roughness in the two engines were quite different even though the net result – shock excitation – is the same in both cases. On this account, it is unwise to attempt to draw any general conclusions which apply to all types of engines. Rather, it appears that each engine presents its own particular problems and that each must be investigated to determine its own special characteristics.

In the present engines, the roughness disturbance apparently arises from the reaction of the flywheel-inertial forces upon the crankshaft. This phenomenon produces shock-excited vibrations in contrast to resonant or forced vibrations. By this is meant that certain engine parts are suddenly excited by a shock and left to vibrate freely until the damping forces consume the respective energies of vibration. The initial amplitudes of these transient vibrations depend on the characteristics of the force which produces the shock.

The relationship between the characteristics of the exciting forces and the response of the mechanical system has not been developed, even for the single-cylinder engine, to the point where it is possible to reach a definite decision about the relative merits of the different criteria of roughness which have been advanced by previous workers.[1, 2, 3, 4] In this connection some question has arisen as to how a so-called "shock factor," as defined by Janeway, is to be determined from corresponding flywheel and pressure records, for example, from records similar to those in Fig. 6. The difficulty arises because of the phase lag between the time of maximum pressure (maximum impressed force) and the time of maximum deflection of the flywheel (maximum restoring force).

On the basis of present information, the choice between the different criteria which have been suggested previously for denoting roughness in the multicylinder engine is even more confusing than for the single-cylinder engine. Here, the response of the crankshaft-flywheel system to mechanical forces appears to be just as important as to the gas loads. As yet, neither one of these sets of exciting forces have been analyzed to the complete satisfaction of the writers even though considerable time has been given to both of these problems. However, it has been found possible from a given pressure card (impressed force curve) together with the mechanical properties of the crankshaft flywheel system, to deduce analytically some of the motions of the flywheel and crankshaft. This procedure which, among other things, predicts the phase lag of the flywheel will be presented in another paper on this general subject.

■ Acknowledgments

The writers take this opportunity to express their appreciation to H. T. Youngren, J. F. Wolfram and their associates of the Oldsmobile engineering staff for use of their laboratories and other facilities while investigating roughness in the 8-cyl engine.

Special thanks are due Larry Becker and George Miskey for assistance rendered while taking records on the multicylinder engines, to R. V. Hutchinson for help in solving problems incident to mounting the discs on the 8-cyl crankshaft, and to V. C. Smith for help with the single-cylinder engine at the Research Laboratories Division.

Finally, the writers desire to express their appreciation to C. F. Kettering and T. A. Boyd for continued interest and encouragement during the progress of this work.

Use of Generalized Coordinates in FLUTTER ANALYSIS

by SAMUEL J. LORING

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Part I

I. Introduction

THE present author has outlined a general approach to the flutter problem in a previous paper.[1] The calculations and results presented here will serve to show how the methods of the earlier paper are applied to a particular problem in flutter. Calculations are described for a cantilever wing which was flutter tested at the NACA laboratories and the test results are compared with the calculated results. Reference 2 gives the test results.

II. Basic Data

The test for which the calculations were made is run 20-15 of wing 2A; the test data are given in Table I of the Appendix of Reference 2. The wing has a rectangular planform and was rigidly supported as a cantilever with the tip vertically below the support. The data necessary for the flutter computations are:

Span length $= s = 81$ in.
Semichord length $= b = 6$ in.
Flexural axis location $= 30\%$ chord aft of leading edge a (notation of Reference 2) $= -0.40$
Weight CG location $= 42.3\%$ chord aft of leading edge x_a (notation of Reference 2) $= 0.25$
Radius of gyration of wing weight $= 27.95\%$ chord $r^2{}_a$ (notation of Reference 2) $= 0.3125$
Total wing weight $= 36.50$ lb
Wing weight per in. $= 0.451$ lb ($m = 1.168 \times 10^{-3}$ lb sec^2 per in.^{2})
Natural frequencies:
First bending $= 1.29$ cps
Second bending $= 7.7$ cps
Third bending $= 21.0$ cps
First torsion $= 18.1$ cps
Mass density of wind tunnel air $= \rho = 2.154 \times 10^{-3}$ lb sec^2 per ft^4

THIS paper presents some further developments of the method of generalized coordinates in flutter analysis, a method which has been described by the author in a previous paper.[1]

The paper is composed of three distinct parts:

Part I shows in detail the calculations by which this method was applied to a flutter model; the method of application of generalized coordinates to a simple flutter problem, and the check with experimental results is shown.

Part II is a manual for the carrying out of more complicated flutter problems by this method; this manual has been in actual use for some time and has been very successful.

Part III is a discussion of the most important and difficult part of the generalized coordinates method, namely, the choice of the generalized coordinates.

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THE AUTHOR: S. J. LORING has been connected with Chance Vought Division of United Aircraft Corp. since his graduation from M.I.T. in 1936. Before assuming his present position of research engineer in charge of analytical research, structures and vibration testing, he had been structural design engineer.

[This paper was presented at the SAE National Aircraft Engineering & Production Meeting, Los Angeles, Calif., Oct. 2, 1943.]

[1] See SAE Transactions, Vol. 49, August, 1941, pp. 345-356: "General Approach to the Flutter Problem," by S. J. Loring.

[2] See NACA Technical Report No. 685 (Restricted), 1940: "Mechanism of Flutter, A Theoretical and Experimental Investigation of the Flutter Problem," by Theodore Theodorsen and I. E. Garrick.

Table 1

	ϕ_{h1}	ϕ_{h2}	ϕ_{a3}	ϕ'_{h1}	ϕ'_{h2}	ϕ'_{a3}	ϕ''_{h1}	ϕ''_{h2}
0	0	0	0	0	0	1.5708	−7.0320	44.060
0.1	−0.0336	0.1856	0.1564	−0.6551	3.3497	1.5515	−6.0637	23.050
0.2	−0.1278	0.5978	0.3091	−1.2130	4.6425	1.4938	−5.1010	2.996
0.3	−0.2731	1.0516	0.4540	−1.6756	4.0651	1.3996	−4.1549	−13.954
0.4	−0.4598	1.3658	0.5857	−2.0452	2.0203	1.2708	−3.2424	−25.947
0.5	−0.6790	1.4281	0.7071	−2.3259	−0.9060	1.1107	−2.3874	−31.417
0.6	−0.9220	1.1778	0.8090	−2.5252	−4.0360	0.9233	−1.6167	−30.089
0.7	−1.1817	0.6334	0.8910	−2.6529	−6.7360	0.7131	−0.9602	−23.167
0.8	−1.4508	−0.1360	0.9510	−2.7217	−8.5686	0.4855	−0.4493	−13.169
0.9	−1.7246	−1.0463	0.9877	−2.7489	−9.4126	0.2458	−0.1181	−4.089
1.0	−2.0000	−2.0000	1.0000	−2.7528	−9.5506	0	0	0

The foregoing values of frequencies are the coupled values measured with the wing supported vertically, the tip being below the support. From the natural frequency data, the following values of flexural rigidity and torsional rigidity were computed:

$$EI = 23.6 \times 10^4 \text{ psi}$$
$$GJ = 35.5 \times 10^4 \text{ psi}$$

These values were computed from the frequencies by an inverse of the type of calculation described in Section VI, in which the stiffnesses were determined from the given natural frequencies. In Section VI it will be shown that these values are consistent with the weight data and the observed natural frequencies of the wing. In actual flutter computations, the rigidities are available from calculations before natural frequency data can be obtained. The point of view taken from here on is that the rigidities are given and not the frequencies; the latter are computed from the former in Section VI for purposes of checking.

III. Choice of Generalized Coordinates

Since flutter of this wing results from a coupling of bending and torsional motions, an inspection of the natural frequencies suggests that at least the first and second bending modes and the first torsional mode of the wing will have to be considered in order to describe the flutter. To make the calculations as simple as possible, three generalized displacements and coordinates which are just sufficient to describe these three modes will be used. Two generalized coordinates will be taken in the form of the first two normal modes of a uniform vertical[3] cantilever in bending and one in the form of the first normal mode of a uniform cantilever in torsion. The bending and twisting displacements, h and α, are then given by:

$$\begin{bmatrix} h \\ \alpha \end{bmatrix} = \begin{bmatrix} \phi_{h1} b & \phi_{h2} b & 0 \\ 0 & 0 & \phi_{\alpha_3} \end{bmatrix} \begin{bmatrix} q_1 \\ q_2 \\ q_3 \end{bmatrix} = \begin{bmatrix} \phi_{h1} b\ q_1 + \phi_{h2} b\ q_2 \\ \phi_{\alpha_3}\ q_3 \end{bmatrix} \quad (1)$$

The semichord, b is introduced to make the coordinates, φ_{h1} and φ_{h2} dimensionless. The values of φ_{h1}, φ_{h2}, and φ_{α_3}, and their first and second derivatives with respect to a dimensionless span coordinate, $\xi = x/s$, are given in Table 1.

The values which appear in Table 1 are the values for a cantilever supported in a horizontal position. Calculations have shown that the bending wave forms for a horizontal and vertical cantilever are the same to four significant figures.

IV. Formation of Inertia Matrix

The inertia matrix expresses the kinetic energy of the system; this matrix is obtained by the methods of Reference 1, Section IV, and for the displacements of a uniform cantilever given by (1) is expressed by:

$$I_r a = I_r \begin{bmatrix} \int_0^1 \phi_{h1}^2\, d\xi & \int_0^1 \phi_{h1}\phi_{h2}\, d\xi & x_a \int_0^1 \phi_{h1}\phi_{\alpha_3}\, d\xi \\ \int_0^1 \phi_{h1}\phi_{h2}\, d\xi & \int_0^1 \phi_{h2}^2\, d\xi & x_a \int_0^1 \phi_{h2}\phi_{\alpha_3}\, d\xi \\ x_a \int_0^1 \phi_{h1}\phi_{\alpha_3}\, d\xi & x_a \int_0^1 \phi_{h2}\phi_{\alpha_3}\, d\xi & r_a^2 \int_0^1 \phi_{\alpha_3}^2\, d\xi \end{bmatrix} \quad (2)$$

where $I_r = msb^2$

Carrying out the integrations and substituting the values of x_a and r^2_a:

$$a = \begin{bmatrix} 1 & 0 & -0.16945 \\ 0 & 1 & 0.04840 \\ -0.16945 & 0.04840 & 0.15625 \end{bmatrix} \text{ and } I_r = 3.40 \text{ lb in. sec}^2$$

The zeros appearing in the above matrix result from the choice of orthogonal bending modes for the generalized coordinates, φ_{h1} and φ_{h2}.

V. Formation of Elastic Matrix

The elastic matrix expresses the potential energy of the system. This matrix is obtained by the method of Reference 1, Section V, and for the displacements of a uniform cantilever given by (1) is expressed by:

$$K_r f = K_r \begin{bmatrix} \int_0^1 \phi''_{h1}{}^2\, d\xi + \gamma \int_0^1 \int_0^\xi \phi'_{h1}{}^2\, d\xi\, d\xi & ; & \int_0^1 \phi''_{h2}\phi''_{h1}\, d\xi + \gamma \int_0^1 \int_0^\xi \phi'_{h2}\phi'_{h1}\, d\xi\, d\xi\ ; & 0 \\ \int_0^1 \phi''_{h1}\phi''_{h2}\, d\xi + \gamma \int_0^1 \int_0^\xi \phi'_{h1}\phi'_{h2}\, d\xi\, d\xi\ ; & & \int_0^1 \phi''_{h2}{}^2\, d\xi + \gamma \int_0^1 \int_0^\xi \phi'_{h2}{}^2\, d\xi\, d\xi & ; & 0 \\ 0 & ; & 0 & ; & \nu \int_0^1 \phi'_{\alpha_3}{}^2 d\xi \end{bmatrix} \quad (3)$$

where $K_r = \dfrac{EI}{s}\left(\dfrac{b}{s}\right)^2$

$$\nu = \frac{GJ}{EI}\left(\frac{s}{b}\right)^2$$

$$\gamma = \frac{gms^3}{EI}$$

g = acceleration of gravity

The terms in the matrix elements in γ take into account the potential energy due to the action of gravity on the vertical wing. The quantity,

$$\int_0^\xi \phi'_{hi}\, \phi'_{hj}\, d\xi$$

represents the displacement of the wing toward the support,[4] and the quantity

$$\gamma \int_0^1 \int_0^\xi \phi_{hi}\, \phi_{hj}\, d\xi d\xi$$

represents the total potential energy due to the vertical displacement against the action of gravity.

Carrying out the integrations, the elastic matrix becomes:

$$f = \begin{bmatrix} 13.953 & 0 & 0 \\ 0 & 494.33 & 0 \\ 0 & 0 & 1.2338\ \nu \end{bmatrix}$$

$K_r = 15.98$ lb in. $\quad \nu = 274$

[3] Effect of gravity on bending frequencies is appreciable and will be included; however, as mentioned in a later paragraph, the effect of gravity on the shape of the deflection forms is very small and will be neglected.

[4] See "Theory of Elastic Stability," by S. Timoshenko, First Edition, McGraw-Hill, 1936, p. 28.

The quantity ν is left explicitly in the matrix because it is a convenient variable to solve for in the solution of the flutter equation. (See Section VIII.)

VI. Equation of Motion for Free Vibrations

The determinantal equation for the still-air natural frequencies of the wing is:

$$\left| aX - f \right| = 0 \tag{4}$$

where

$$X = \frac{I_r}{K_r}\omega^2$$

or

$$\omega = \sqrt{\frac{K_r}{I_r}}\,X = 0.345\sqrt{X} \text{ cps}$$

Substituting the values of a and f in (4):

$$\begin{vmatrix} X-13.953 & 0 & -0.16945X \\ 0 & X-494.33 & 0.04840X \\ -0.16945X & 0.04840X & 0.15625X - 338.06 \end{vmatrix} = 0$$

Expansion of the determinant leads to the following cubic equation:

$$0.125194X^3 - 403.252X^2 + 172922X - 2{,}331{,}730 = 0$$

The roots of the cubic equation and the corresponding frequencies are given in Table 2, and are compared with the frequencies observed in the still-air vibration tests of the wing.

Table 2

	Values of Frequency, cps	
X	Calculated	Observed
13.94	1.29	1.29
492.3	7.65	7.7
2715	17.98	18.1

This check of the observed natural frequencies is satisfactory since the measured values of the two higher frequencies are accurate to only within $\pm 3\%$.

VII. Formation of Airforce Matrix

The airforce matrix is given by the methods of Reference 1, Section VII, and for the displacements of a uniform cantilever given by (1) is expressed by:

$$c = \begin{bmatrix} \int_0^1 \phi_{h1}^2 A_{ch}\,d\xi & \int_0^1 \phi_{h1}\phi_{h2} A_{ch}\,d\xi & \int_0^1 \phi_{h1}\phi_{\alpha 2} A_{c\alpha}\,d\xi \\ \int_0^1 \phi_{h1}\phi_{h2} A_{ch}\,d\xi & \int_0^1 \phi_{h2}^2 A_{ch}\,d\xi & \int_0^1 \phi_{h2}\phi_{\alpha 2} A_{c\alpha}\,d\xi \\ \int_0^1 \phi_{h1}\phi_{\alpha 2} A_{\alpha h}\,d\xi & \int_0^1 \phi_{h2}\phi_{\alpha 2} A_{\alpha h}\,d\xi & \int_0^1 \phi_{\alpha 2}^2 A_{\alpha\alpha}\,d\xi \end{bmatrix} \tag{5}$$

The airforce coefficients A_{ch}, and so on, were obtained from Reference 5. With the value $a = -0.40$, the expressions for the coefficients are:

$$R_{\alpha\alpha} = -0.285 + 0.09\left(\frac{2G}{k}\right) - 0.10\left(\frac{2F}{k^2}\right)$$

$$I_{\alpha\alpha} = \left[0.900 - 0.10\left(\frac{2G}{k}\right) - 0.09\,(2F)\right]1/k$$

$$R_{\alpha h} = -0.40 \quad +0.10\left(\frac{2G}{k}\right)$$

$$I_{\alpha h} = -0.10\left(\frac{2F}{k}\right)$$

$$R_{c\alpha} = -0.40 \quad -0.90\left(\frac{2G}{k}\right) + \left(\frac{2F}{k^2}\right)$$

$$I_{c\alpha} = \left[1.000 + \left(\frac{2G}{k}\right) + 0.90\,(2F)\right]1/k$$

$$R_{ch} = -1 - \frac{2G}{k}$$

$$I_{ch} = 2F/k$$

The quantities F, G, and k are defined in Reference 5.

Because the wing has a constant chord, the airforces given by A_{ch}, and so on, will remain constant over the span for a particular value of k. Using the integrals that have already been carried out for the inertia matrix, the airforce matrix becomes:

$$c = \begin{bmatrix} A_{ah} & 0 & -0.6778\,A_{a\alpha} \\ 0 & A_{ch} & 0.1936\,A_{a\alpha} \\ -0.6778\,A_{ah} & 0.1936\,A_{ah} & 0.5000\,A_{a\alpha} \end{bmatrix}$$

VIII. Flutter Solution

The flutter equation of Reference 1, Section VIII is:

$$\left| -\Gamma_a + c + \Upsilon_f \right| = 0 \tag{6}$$

where

$$\Gamma = \frac{I_r}{\pi\rho b^4 s} = 99.0$$

$$\Upsilon = \left(\frac{\omega_r}{\omega}\right)^2$$

and

$$\omega_r = \sqrt{\frac{K_r}{\pi\rho b^4 s}} = 3.44 \text{ cps}$$

The flutter speed and flutter frequency may be found from the following equations:

$$v = \frac{\omega_r b}{k\sqrt{\Upsilon}} = \frac{7.36}{k\sqrt{\Upsilon}} \text{ mph} \tag{7}$$

$$\omega = \frac{\omega_r}{\sqrt{\Upsilon}} = \frac{3.44}{\sqrt{\Upsilon}} \text{ cps}$$

The method of solution will be explained and followed by a sample calculation. Since a direct solution for k is very laborious, solutions for Υ and ν were made for successive values of k as suggested in Reference 1. Plots were made of k versus ν (Fig. 1) and flutter speed versus ν (Fig. 2). The method of solution is as follows. Equation (6) was set up and expanded. The real and imaginary parts of the expanded determinant were set equal to zero, yielding two simultaneous equations in Υ and ν. The quantity ν is eliminated between these equations and the

[5] See NACA Technical Report No. 496, 1934: "General Theory of Aerodynamic Instability and the Mechanism of Flutter," by Theodore Theodorsen.

resulting equation in Υ solved. The value of v is obtained from either of the simultaneous equations and the values of flutter speed and flutter frequency determined from (7).

Calculations are presented for the value of k corresponding to the flutter of the wing: $v = 274$, $k = 0.0965$ (Fig. 1). For $k = 0.0965$, the airforce matrix is:

$$c = \begin{bmatrix} 2.5278 + 17.345i & 0 & -123.71 + 7.1738i \\ 0 & 2.5278 + 17.345i & 35.336 - 2.0491i \\ 0.51023 + 1.1756i & -0.14574 - 0.33580i & -9.2880 + 5.7105i \end{bmatrix}$$

Damping was taken into account by entering into the matrix Υf the damping coefficients appropriate to the two bending modes and the torsional mode. These coefficients for this particular model were found in Reference 6, the values being:

$$g_{h1} = 0.0050$$
$$g_{h2} = 0.0140$$
$$g_{\alpha_2} = 0.0087$$

The matrix Υf then becomes:

$$\Upsilon f = \begin{bmatrix} 13.953\Upsilon(1 + 0.005i); & 0; & 0 \\ 0; & 494.33\Upsilon(1 + 0.014i); & 0 \\ 0; & 0; & 1.2338v\Upsilon(1 + 0.0087i) \end{bmatrix}$$

and the matrix $-\Gamma a$ is:

$$-\Gamma a = \begin{bmatrix} -99 & 0 & 16.776 \\ 0 & -99 & -4.7916 \\ 16.776 & -4.7916 & -15.469 \end{bmatrix}$$

Therefore, for $k = 0.0965$, the determinantal equation (6) is:

$$\begin{vmatrix} (-9.6472 + 1.7345i + \Upsilon[1.3953 + 0.006977i]) & 0 & (-10.694 + .71738i) \\ 0 & (-9.6472 + 1.7345i + \Upsilon[49.433 + 0.69206i]) & (3.0544 - 0.20491i) \\ (1.7286 + 0.11756i) & (-0.49373 - 0.033580i) & (-2.4757 + .57105i + v\Upsilon[0.12338 + 0.001073i]) \end{vmatrix} = 0$$

A factor of 10 has been taken out of the determinant to reduce the size of the numbers which occur in the expansion. Expanding and setting the real and imaginary parts equal to zero:

$$v\Upsilon[8.50799\Upsilon^2 - 60.7364\Upsilon + 11.1475] + [-171.494\Upsilon^2 + 2090.54\Upsilon - 397.646] = 0$$

$$v\Upsilon[0.235696\Upsilon^2 + 9.51790\Upsilon - 4.03241] + [36.1403\Upsilon^2 - 468.562\Upsilon + 168.940] = 0$$

Eliminating $v\Upsilon$:

$$347.902\Upsilon^4 - 5042.02\Upsilon^3 + 9803.62\Upsilon^2 - 3269.43\Upsilon + 279.787 = 0$$

[6] See NACA Technical Note No. 751, 1940: "Damping Formulas and Experimental Values of Damping in Flutter Models," by Robert P. Coleman.

Solving for Υ and then v:

$\Upsilon = 0.1406$	$v = 274.$
$\Upsilon = 0.2531$	$v = 129.$
$\Upsilon = 1.843$	$v = 21.$
$\Upsilon = 12.25$	$v = 0.0801$

The values of flutter speed corresponding to the first three values of Υ are:

$\Upsilon = 0.1406$	$v = 203$ mph
$\Upsilon = 0.2531$	$v = 151.5$ mph
$\Upsilon = 1.843$	$v = 56.1$ mph

It should be noted that the values of Υ other than 0.1406 do not correspond to the value of 274. for v, and therefore do not have significance in regard to the flutter of the actual model.

Table 3 gives the calculated and observed flutter speeds and flutter frequencies.

Table 3

Item	Calculated Values	Observed Values
v = Flutter Speed, mph	203	202
ω = Flutter Frequency, cps	9.18	10.2

IX. Flutter Mode

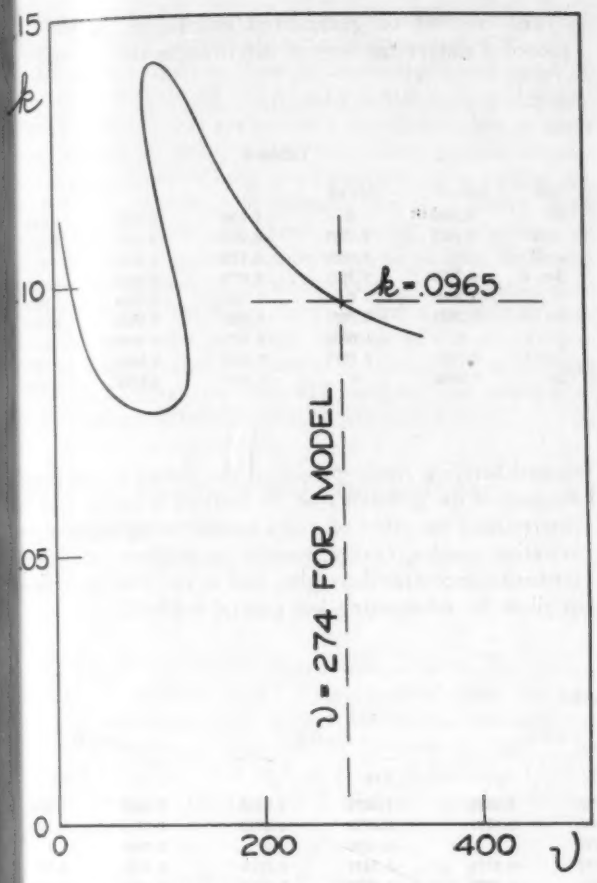
The appearance of the flutter mode was determined by plotting the displacement envelope curve at various stations along the span of the wing. The method will be outlined and the calculations shown.

From the determinantal equation (6), the three equilibrium equations in the three generalized displacement amplitudes can be obtained. (See Reference 1, Section VII.) with y_1, y_2, and y_3 as the displacement amplitudes these equations are:

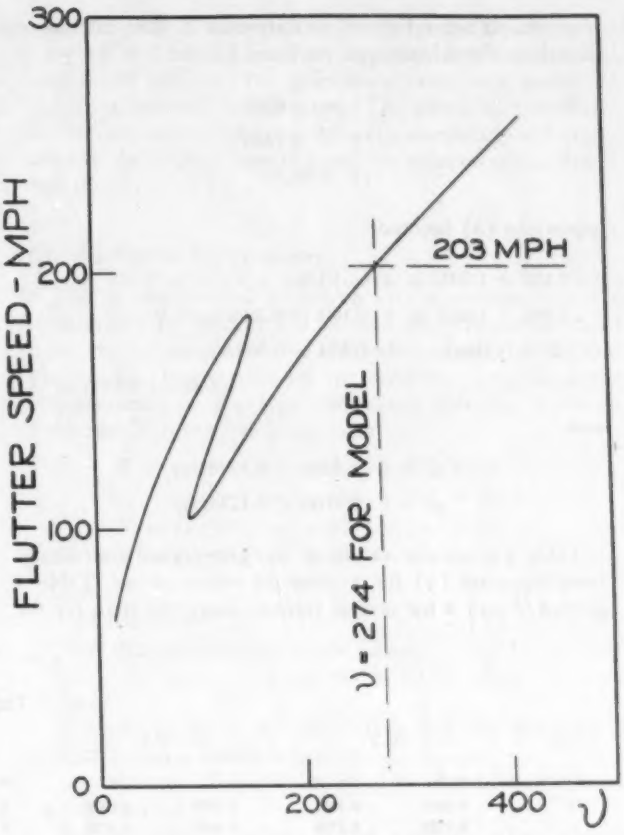
$$(-\Gamma a_{11} + c_{11} + \Upsilon f_{11})y_1 + (-\Gamma a_{12} + c_{12})y_2 + (-\Gamma a_{13} + c_{13})y_3 = 0$$
$$(-\Gamma a_{21} + c_{21} + \Upsilon f_{21})y_1 + (-\Gamma a_{22} + c_{22})y_2 + (-\Gamma a_{23} + c_{23})y_3 = 0$$
$$(-\Gamma a_{31} + c_{31})y_1 + (-\Gamma a_{32} + c_{32})y_2 + (-\Gamma a_{33} + c_{33} + v\Upsilon f_{33})y_3 = 0$$

Solving for y_1 and y_2 in terms of y_3:

$$y_1 = a_1 y_3$$
$$y_2 = a_2 y_3$$



■ Fig. 1 – Plot of k versus stiffness parameter υ



■ Fig. 2 – Plot of flutter speed versus stiffness parameter υ

where the a_i's are complex coefficients $a_{iR} + ia_{iI}$. Now $q_i = y_i e^{i\omega t} = y_i (\cos \omega t + i \sin \omega t)$ (see Reference 1, Section VII): since equations (8) are homogeneous, the absolute magnitudes of the y's are arbitrary; consequently, putting $y_3 = 1$,

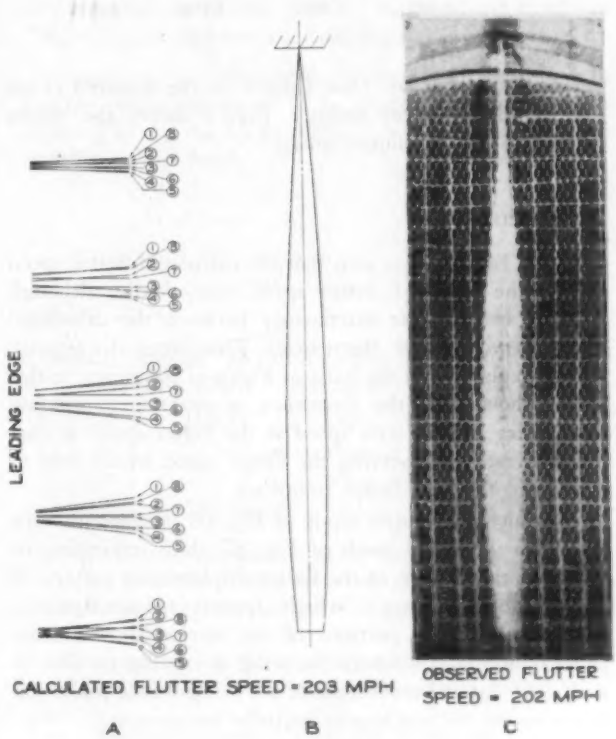
$$q_1 = a_1 (\cos \omega t + i \sin \omega t)$$
$$q_2 = a_2 (\cos \omega t + i \sin \omega t)$$
$$q_3 = (\cos \omega t + i \sin \omega t)$$

or, using only the real part (which corresponds to the actual motion):

$$q_1 = a_{1R} \cos \omega t - a_{1I} \sin \omega t$$
$$q_2 = a_{2R} \cos \omega t - a_{2I} \sin \omega t$$
$$q_3 = \cos \omega t \qquad (9)$$

The displacement forms are given by (1). At several wing sections along the span the displacements, h/b and α were determined at equally spaced instants for one complete cycle ($\omega t = 0$ to $\omega t = 2\pi$). These displacements were plotted to show the motions of the wing sections. The projected envelope of the displacement patterns in the front view was plotted to show the appearance of the flutter mode to compare with Fig. 18, Reference 2.

The section displacement positions for several wing sections and the corresponding section displacement patterns are shown in Fig. 3A. The appearance of the flutter mode in the front view is compounded in Fig. 3B from the section displacement patterns, and may be compared to an actual photograph of the flutter in Fig. 3C. Fig. 3C is



■ Fig. 3 – Comparison of calculated and observed flutter modes and speeds

reproduced from Fig. 18 of Reference 2. The calculations based on the scheme just outlined follow:

For:

$$k = 0.0965$$

$$T = 0.1406$$

$$\nu T = 38.58$$

equations (8) become:

$$(-9.450 + 1.735i)y_1 + (-10.694 + 0.717i)y_3 = 0$$

$$(-2.695 + 1.832i)y_2 + (3.054 - 0.205i)y_3 = 0$$

$$(1.729 + 0.118i)y_1 + (-0.494 - 0.034i)y_2 + (2.285 + 0.612i)y_3 = 0$$

and

$$y_1 = (-1.1081 - 0.1276i)y_3$$

$$y_2 = (\ 0.8105 + 0.4748i)y_3$$

Table 4 gives the values of the generalized coordinates from equation (9) for a series of values of ωt. Table 5 gives h/b and α for several stations along the span for the series of values of ωt. (See Table 1 for the required values for the displacement forms.) Fig. 3 shows the results plotted to give the flutter modes.

The method of generalized coordinates in this case proved a convenient way of determining the role that the second bending mode played in the flutter of the model. Because of the generality of the method it can be used for determining the effect of many factors (wing taper, higher vibration modes, curved weight or stiffness axis, sweep forward, concentrated weights, and so on) that are difficult to allow for when using less general methods.

Table 4

ωt	cos ωt	sin ωt	q_1	q_2	q_3
0	1.0000	0	−1.1081	0.8105	1.0000
$\pi/4$	0.7071	0.7071	−0.8933	0.2374	0.7071
$\pi/2$	0	1.0000	0.1276	−0.4749	0
$3\pi/4$	−0.7071	0.7071	0.8738	−0.9069	−0.7071
π	−1.0000	0	1.1081	−0.8105	−1.0000
$5\pi/4$	−0.7071	−0.7071	0.8933	−0.2374	−0.7071
$3\pi/2$	0	−1.0000	−0.1276	0.4749	0
$7\pi/4$	0.7071	−0.7071	−0.8738	0.9069	0.7071
2π	1.0000	0	−1.1081	0.8105	1.0000

Table 5

ωt	$\xi=0.2$		$\xi=0.4$		$\xi=0.6$		$\xi=0.8$		$\xi=1.0$	
	h/b	α	h/b	α	h/b	α	h/b	α	h/b	α
0	0.6261	0.3091	1.6165	0.5878	1.9765	0.8090	1.4974	0.9510	0.5952	1.0000
$\pi/4$	0.2305	0.2188	0.6430	0.4156	0.9190	0.5721	0.9738	0.6725	0.9118	0.7071
$\pi/2$	−0.3002	0	−0.7073	0	−0.6770	0	−0.1205	0	0.6948	0
$3\pi/4$	−0.6550	−0.2188	−1.6431	−0.4156	−1.8763	−0.5721	−1.1441	−0.6725	0.0702	−0.7071
π	−0.6261	−0.3091	−1.6165	−0.5878	−1.9765	−0.8090	−1.4974	−0.9510	−0.5952	−1.0000
$5\pi/4$	−0.2305	−0.2188	−0.6430	−0.4156	−0.9190	−0.5721	−0.9738	−0.6725	−0.9118	−0.7071
$3\pi/2$	0.3002	0	0.7073	0	0.6770	0	0.1205	0	−0.6948	0
$7\pi/4$	0.6550	0.2188	1.6431	0.4156	1.8763	0.5721	1.1441	0.6725	−0.0702	0.7071
2π	0.6261	0.3091	1.6165	0.5878	1.9765	0.8090	1.4974	0.9510	0.5952	1.0000

X. Discussion

From Table 3, it is seen that the calculated flutter speed checks the observed flutter speed very closely, although there is considerable discrepancy between the calculated and observed flutter frequencies. This latter discrepancy can be explained on the basis of Fig. 3 of Reference 7; this figure shows that the frequency of maximum response varies very rapidly with speed at the flutter speed so that a small error in observing the flutter speed would lead to a large error in the flutter frequency.

The calculated flutter mode of Fig. 3B corresponds very closely to the actual mode of Fig. 3C. It is interesting to note that the portion of the flutter displacement pattern of Fig. 3C near midspan, which appears semitransparent, corresponds to the portions of the section displacement patterns of Fig. 3A where the wing is moving parallel to itself (the parallel movement of the wing would make the displacement pattern appear partially transparent).

† See *Journal of the Aeronautical Sciences*, Vol. 8, June, 1941, pp. 313-318: "Some Experimental Results on Wing Flutter," by William Bollay and Charles D. Brown.

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Part II

I. Introduction

The treatment of the flutter of conventional airplane tails involves three or more fairly complicated vibration modes of the horizontal and vertical surfaces. For use in this and similar flutter problems, a method is presented for treating flutter with many degrees of freedom involving torsion, bending, and flap deflection of one or more airfoil surfaces having aerodynamically balanced flaps. This method consists essentially of a scheme for applying the general flutter analysis presented in Reference 1.

There are two fairly distinct parts in the application of the general flutter analysis of Reference 1 to a particular structure. The first part is the determination of the vibration modes or generalized coordinates of the structure; this part of a flutter analysis is treated in Part III. The second part is the determination of the flutter characteristics when the generalized coordinates are given. If the airforces are known, this part of a flutter solution is purely analytical and is largely a computational problem. The method described here is limited to this second part of a flutter analysis. An analytical method suitable for convenient numerical computation and presentation of results

is described and a convenient arrangement of numerical work including numerical checks is presented.

Most flutter analyses show as results the flutter speed of the structure analyzed. In treating flutter with a very few simple modes, it has always been found desirable to show the variation of flutter speed with some significant construction parameter (such as a frequency ratio) so that not only flutter speeds will be indicated but, if no flutter speed exists for the actual structure analyzed, the closeness of approach to a flutter condition may be judged. Where a number of fairly complicated modes are involved, it is difficult or impossible to choose a single significant construction parameter, and it is desirable in this case to present as a result some measure of the proximity to a flutter speed of a completely defined structure. The method of flutter analysis described leads to just such a result.

II. Flutter Equations and Expression for Displacements

The flutter equation, equation (60), of reference 1:

$$\left| -\Gamma a + c + \Upsilon f \right| = 0 \tag{1}$$

is valid for any structure with linear airforces. The inertia and elastic matrices a and f are obtained from the generalized coordinates and the mass and elastic characteristics of the structure as indicated in Sections IV and V of Reference 1. The value of Γ is given in equation (59) of Reference 1, and the variable Υ is, from equation (61) of Reference 1.

$$\Upsilon = \frac{1}{k_o^2}\left(\frac{v_r}{v}\right)^2 \quad \text{or} \quad k_o^2\Upsilon = \left(\frac{v_r}{v}\right)^2 \tag{2}$$

where

v = air-speed of structure,

$k_o = \omega b_o/v$ [defined in equation (45) of Reference 1]

$v_r = \omega_r b_o$ [quantities ω_r and b_o defined in Reference 1]

= reference air-speed

Since the airforces in the present analysis are limited to those resulting from twisting or linear deflection of an entire airfoil, relative rotation and linear displacement of a flap (these two motions include rotation of aerodynamically balanced flap) the generalized coordinates expressing the displacements of the aerodynamic surfaces must be limited to the following form:

$$h = Hq = q'H' \tag{3}$$

where h = column matrix of following section displacements:

h_1 = downward displacement of ¼ chord point referred to b_o,

h_2 = rotation of whole airfoil in radians, positive when stalling airfoil,

h_3 = rotation of flap relative to airfoil in radians, positive in the same sense as h_2,

h_4 = downward displacement of flap leading edge relative to airfoil chord line, referred to b_o,

b_o = reference length,

q = column of generalized displacements,

q' = transpose of q,

H = four-rowed rectangular matrix expressing generalized coordinate shapes, elements are functions of spanwise coordinate x of airfoil,

H' = transpose of H.

In the present analysis, the inertia and elastic matrices a and f; and the factor Γ in equation (1) are assumed known. In addition, the generalized coordinate matrix H in (3) is assumed to be given. The phase of the flutter analysis problem to be treated here is specifically the evaluation of the airforce matrix c and the solution of the flutter equation (1).

III. Airforce Expressions

The general method of setting up the airforce matrix c is described in Section VII of Reference 1, and is based upon the expression of the airforces in terms of the displacements. Expressions for the airforces in terms of the displacements of the type considered here are given in Reference 8 in the following form:

$$\begin{aligned} K &= \pi\rho v^2 b \ [Ak_a + Bk_b + Ck_c + Dk_d] \\ M_o &= \pi\rho v^2 b^2 \ [Am_a + Bm_b + Cm_c + Dm_d] \\ N &= \pi\rho v^2 b^2 \ [An_a + Bn_b + Cn_c + Dn_d] \\ R &= \pi\rho v^2 b \ [Ar_a + Br_b + Cr_c + Dr_d] \end{aligned} \tag{4}$$

where

b = semichord

K = upward force on whole airfoil

M_o = diving moment on whole airfoil about ¼ chord point

N = diving moment on flap about flap leading edge

R = upward force on flap

A = downward displacement of ¼ chord point referred to semichord b

B = rotation of whole airfoil, positive when stalling airfoil

C = rotation of flap relative to airfoil, positive in same sense as B

D = downward displacement of flap leading edge relative to airfoil chord line, referred to semichord b

$k_a \ldots r_d$ = airforces coefficients given in Reference 8

In order to put these airforce expressions in a form to use the displacements A, B, C, D defined above must be replaced by h_1, h_2, h_3, h_4, defined in Section II. The relationship is as follows:

$$\begin{aligned} A &= h_1/\theta \\ B &= h_2 \\ C &= h_3 \\ D &= h_4/\theta \end{aligned} \tag{5}$$

where $\theta = b/b_o$

Substitutions of (5) into (4) and the use of the relation $b = \theta b_o$ results in the following airforce expression:

$$\begin{bmatrix} Kb_o \\ M_o \\ N \\ Rb_o \end{bmatrix} = \pi\rho v^2 b_o^2 \begin{bmatrix} k_a & \theta k_b & \theta k_c & k_d \\ \theta m_a & \theta^2 m_b & \theta^2 m_c & \theta m_d \\ \theta n_a & \theta^2 n_b & \theta^2 n_c & \theta n_d \\ r_a & \theta r_b & \theta r_c & r_d \end{bmatrix} \begin{bmatrix} h_1 \\ h_2 \\ h_3 \\ h_4 \end{bmatrix} \tag{6}$$

The forces corresponding to the displacements h are:

M_1 = downward force on whole airfoil multiplied by b_o

M_2 = stalling moment on whole airfoil about ¼ chord point

M_3 = stalling moment on flap about leading edge

M_4 = downward force on flap multiplied by b_o

[8] See *Luftfahrt-Forschung*, Vol. 17, Dec. 10, 1940, pp. 337-354: "Der schwingende Flügel mit aerodynamisch ausgeglichenem Ruder," by H. G. Küssner and L. Schwarz. Translated in NACA Technical Memorandum No. 991, 1941: "The Oscillating Wing with Aerodynamically Balanced Elevators."

Forces have been multiplied by the reference length so that the dimensions of all forces are force $\times$ length. These forces are the negative of those given by (6). Using the relation

$$v = \frac{\omega b}{k} = \omega b_o \frac{\theta}{k} \tag{7}$$

to eliminate v^2 from (6), the airforces M_1---M_4 may be written

$$M = -\pi\rho\omega^2 b_o{}^4 \Theta\Lambda\Theta h \tag{8}$$

where M = column matrix of forces M_1---M_4

$$\Theta = \begin{bmatrix} \theta & 0 & 0 & 0 \\ 0 & \theta^2 & 0 & 0 \\ 0 & 0 & \theta^3 & 0 \\ 0 & 0 & 0 & \theta \end{bmatrix}$$

$$\Lambda = 1/k^2 \begin{bmatrix} k_a & k_b & k_c & k_d \\ m_a & m_b & m_c & m_d \\ n_a & n_b & n_c & n_d \\ r_a & r_b & r_c & r_d \end{bmatrix}$$

The aerodynamic terms in the section airforce expression (8) are contained in the matrix Λ, the terms of which are evaluated in Reference 8. If the expressions in Reference 8 are transformed in accordance with the following scheme:

$$1 + T = 2F + i2G \tag{9}$$
$$\omega = ik$$

the matrix Λ can be expressed as follows:

$$\Lambda = \left[\Lambda_1 + \frac{1}{k^2}\Lambda_2 + \frac{2F}{k^2}\Lambda_3 - \frac{2G}{k}\Lambda_4\right] + i\left[\frac{1}{k}\Lambda_5 + \frac{2G}{k^2}\Lambda_3 + \frac{2F}{k}\Lambda_4\right]; \tag{10}$$

where the matrices Λ_1 are functions only of the flap chord ratio τ_f and have the values given below. The functions Φ_1---Φ_{35} are tabulated by Küssner and Schwarz.[8]

$$\Lambda_1 = \begin{bmatrix} -1 & -\frac{1}{2} & -\frac{\phi_4}{2\pi} & -\frac{\phi_3}{\pi} \\ -\frac{1}{2} & -\frac{3}{8} & -\frac{\phi_7}{4\pi} & -\frac{\phi_6}{4\pi} \\ -\frac{\phi_4}{2\pi} & -\frac{\phi_7}{4\pi} & -\frac{\phi_{12}}{4\pi^2} & -\frac{\phi_{27}}{2\pi^2} \\ -\frac{\phi_3}{\pi} & -\frac{\phi_6}{4\pi} & -\frac{\phi_{27}}{2\pi^2} & -\frac{\phi_{17}}{\pi^2} \end{bmatrix}$$

$$\Lambda_2 = \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & \frac{\phi_5}{\pi} & 0 \\ 0 & 0 & \frac{\phi_{10}}{\pi^2} & 0 \\ 0 & 0 & \frac{\phi_{35}}{\pi^2} & 0 \end{bmatrix}$$

$$\Lambda_3 = \begin{bmatrix} 0 & 1 & \frac{\phi_1}{\pi} & 0 \\ 0 & 0 & 0 & 0 \\ 0 & \frac{\phi_8}{2\pi} & \frac{\phi_1\phi_8}{2\pi^2} & 0 \\ 0 & \frac{\phi_{31}}{\pi} & \frac{\phi_1\phi_{31}}{\pi^2} & 0 \end{bmatrix}$$

$$\Lambda_4 = \begin{bmatrix} 1 & 1 & \frac{\phi_2}{2\pi} & \frac{\phi_1}{\pi} \\ 0 & 0 & 0 & 0 \\ \frac{\phi_8}{2\pi} & \frac{\phi_8}{2\pi} & \frac{\phi_2\phi_8}{4\pi^2} & \frac{\phi_1\phi_8}{2\pi^2} \\ \frac{\phi_{31}}{\pi} & \frac{\phi_{31}}{\pi} & \frac{\phi_2\phi_{31}}{2\pi^2} & \frac{\phi_1\phi_{31}}{\pi^2} \end{bmatrix}$$

$$\Lambda_5 = \begin{bmatrix} 0 & 1 & \frac{\phi_3}{\pi} & 0 \\ 0 & 1 & \frac{\phi_6}{2\pi} & \frac{\phi_5}{\pi} \\ 0 & \frac{\phi_9}{2\pi} & \frac{\phi_{11}}{2\pi^2} & \frac{\phi_{10}}{\pi^2} \\ 0 & \frac{\phi_{32}}{\pi} & \frac{\phi_{36}}{\pi^2} & \frac{\phi_{35}}{\pi^2} \end{bmatrix}$$

IV. Formation of Airforce Matrix

The airforce matrix is set up following the method of Section VII of Reference 1. The work done by the airforces during an arbitrary displacement is

$$\delta W = \int \delta h' M dx = \delta q' \left\{ \int H' (-\pi\rho\omega^2 b_o{}^4 \Theta\Lambda\Theta) H dx \right\} q \tag{12}$$
$$= \delta q' \left[-\pi\rho\omega^2 b_o{}^4 s \int H' \Theta\Lambda\Theta H d\xi \right] q$$

where $x = s\xi$ and ξ is a dimensionless span coordinate.

The air forces are, therefore, given by

$$-\pi\rho\omega^2 b_o{}^4 sc$$

where c is the "airforce matrix" and is given by

$$c = \int H' \Theta\Lambda\Theta H d\xi \tag{13}$$

If there is more than one aerodynamic surface, the integral (13) must be carried out over each surface; it is only necessary that the scale s of the dimensionless span coordinate ξ be the same for all surfaces.

The integrand of expression (13) is a function of the geometrical characteristics of the aerodynamic surfaces and of the parameter $k = \theta k_o$, where $\theta = b/b_o$ is a geometrical characteristic. It would be convenient to carry out the integrations for a particular structure in such a way that the integral c is expressed explicitly as a function only of k_o. The difficulty in doing this is that the complicated functions F and G have the argument θk_o so that the geometrical quantity θ and the parameter k_o are not sepa-

rate; this difficulty can be overcome by expressing $F(\theta k_o)$ and $G(\theta k_o)$ in the following way:

$$F = F_o + \psi_1 F_o' + \psi_2 F_o'' \qquad (14)$$
$$G = G_o + \psi_3 G_o' + \psi_4 G_o''$$

where

F and G have the argument $\theta k_o = k$

F_o and G_o have the argument k_o

$\psi_1 \ldots \psi_4$ are functions of θ only.

Expressions of the form (14) for F and G are derived in the Appendix to Part II, where the degree of approximation is indicated in Table II. The values of the functions $\psi_1 \ldots \psi_4$ and of the required functions of F_θ, G_o, F_θ', G_θ', F_o'', G_o'' are given in the Appendix. By use of expressions (14) and (10), the airforce matrix c of (13) can be expressed explicitly in terms of k_o as follows:

$$c = \left[C_1 + \frac{1}{k_o^2} C_2 + \frac{2F_o}{k_o^2} C_3 + \frac{2F_o'}{k_o^2} C_4 + \frac{2F_o''}{k_o^2} C_5 - \frac{2G_o}{k_o} C_6 - \frac{2G_o'}{k_o} C_7' - \frac{2G_o''}{k_o} C_8' \right] + i \left[\frac{1}{k_o} C_9 + \frac{2G_o}{k_o^2} C_3 + \frac{2G_o'}{k_o^2} C_4' + \frac{2G_o''}{k_o^2} C_5' + \frac{2F_o}{k_o} C_6 + \frac{2F_o'}{k_o} C_7 + \frac{2F_o''}{k_o} C_8 \right] \qquad (16)$$

where

$$\lambda_i = H\Theta\Lambda_i\Theta H$$

and

$$C_1 = \int \lambda_1 d\xi \qquad C_2 = \int \lambda_2 \frac{1}{\theta^2} d\xi \qquad C_3 = \int \lambda_3 \frac{1}{\theta^2} d\xi$$
$$C_4 = \int \lambda_3 \frac{1}{\theta^2} \psi_1 d\xi \qquad C_4' = \int \lambda_3 \frac{1}{\theta^2} \psi_3 d\xi$$
$$C_5 = \int \lambda_3 \frac{1}{\theta^2} \psi_2 d\xi \qquad C_5' = \int \lambda_3 \frac{1}{\theta^2} \psi_4 d\xi$$
$$C_6 = \int \lambda_4 \frac{1}{\theta} d\xi \qquad C_9 = \int \lambda_5 \frac{1}{\theta} d\xi$$
$$C_7 = \int \lambda_4 \frac{1}{\theta} \psi_1 d\xi \qquad C_7' = \int \lambda_4 \frac{1}{\theta} \psi_3 d\xi$$
$$C_8 = \int \lambda_4 \frac{1}{\theta} \psi_2 d\xi \qquad C_8' = \int \lambda_4 \frac{1}{\theta} \psi_4 d\xi$$

The matrix integrals $C_1 \ldots C_9$ are functions only of the generalized coordinates H and the geometrical characteristics of the aerodynamic surfaces, and have definite values for a given structure and given generalized displacements. The airforce matrix c is therefore a function only of the parameter k_o. The flutter equation (1) may therefore be set up.

In dealing numerically with the flutter equation in the form (1), there are two computational inconveniences. The first is that, as indicated by (16), certain terms in the airforce matrix c become infinite as k_o approaches o and the second is that an infinite range, $o < k_o < \infty$, of the parameter k_o must be considered. The first of these difficulties may be overcome by multiplying each element in the determinantal equation (1) by the factor

$$\frac{k_o^2}{1 + k_o^2}$$

In doing this, it will be convenient to include the inertia terms $-\Gamma a$ with the first term of the airforce matrix C_1, therefore, C_1 will be replaced by

$$C_1' = -\Gamma a + C_1 \qquad (17)$$

The new combined air and inertia matrix is, therefore,

$$c_o = \left\{ C_1' \frac{k_o^2}{1+k_o^2} + C_2 \frac{1}{1+k_o^2} + C_3 \frac{2F_o}{1+k_o^2} + C_4 \frac{2F_o'}{1+k_o^2} + C_5 \frac{2F_o''}{1+k_o^2} - C_6 \frac{2G_o k_o}{1+k_o^2} - C_7' \frac{2G_o' k_o}{1+k_o^2} - C_8' \frac{2G_o'' k_o}{1+k_o^2} \right\} + i \left\{ C_9 \frac{k_o}{1+k_o^2} + C_3 \frac{2G_o}{1+k_o^2} + C_4' \frac{2G_o'}{1+k_o^2} + C_5' \frac{2G_o''}{1+k_o^2} + C_6 \frac{2F_o k_o}{1+k_o^2} + C_7 \frac{2F_o' k_o}{1+k_o^2} + C_8 \frac{2F_o'' k_o}{1+k_o^2} \right\} \qquad (18)$$

The new flutter equation is

$$\left| c_o + f X \right| = 0 \qquad (19)$$

where $X = \dfrac{k_o^2 \Upsilon}{1+k_o^2} = \left(\dfrac{v_r}{v}\right)^2 \dfrac{1}{1+k_o^2}$

according to equation (2)

The second inconvenience mentioned above can be overcome by using instead of k_o the parameter

$$\mu = \frac{k_o}{\sqrt{1+k_o^2}} \qquad (20)$$

which has the range $o < \mu < 1$ when k_o has the range $o < k_o < \infty$. The airforce integral c_o will then be considered a function of μ and the flutter equation (19) will then be a function of μ and X. The relationship between μ, X and the speed v is as follows:

$$X = \left(\frac{v_r}{v}\right)^2 (1-\mu^2) \quad \text{or} \quad \left(\frac{v}{v_r}\right)^2 = \frac{1-\mu^2}{X} \qquad (21)$$

and the vibration frequency is given by [See equation (61), Reference 1.]

$$\left(\frac{\omega}{\omega_r}\right)^2 = \frac{\mu^2}{X} \qquad (22)$$

The functions of μ entering the airforce matrix (18) are tabulated in Table III of the Appendix to Part II.

V. Solution of Flutter Equation

The condition for flutter is that the determinantal expression (19) be zero. The method of solution of the flutter equation described below will indicate the approach to flutter in each "flutter" mode at each air speed as well as showing actual flutter speeds.

The overall damping (structural plus air damping) of a structure can be measured by a quantity g which is the negative of an equivalent structural damping factor which must be added to whatever actual structural damping the structure has in order just to produce a condition of flutter. The form of the equivalent damping factor g will be the same as that introduced in Section VI of Reference 1, and will be assumed equal in every degree of freedom. Further,

the result g_t of "adding" g to the actual structural damping g_{ij} will be defined by the equation

$$(1 + ig_t) = (1 + ig_{ij})(1 - ig) \quad \text{[See Footnote 9]} \qquad (23)$$

SAMPLE WORK SHEET NO.1

TABULATION OF MASS AND ELASTIC DATA

$f =$ [ELASTIC MATRIX, N COLUMNS, N ROWS]

$a =$ [INERTIA MATRIX, N COLUMNS, N ROWS]

$\omega_r =$ _____

$v_r =$ _____

$\beta_o =$ _____

$\Gamma =$ _____ AT ALTITUDE OF _____

If this equivalent structural damping factor g is introduced into the flutter equation, there will be the three variables μ, X, and g in equation (19) and the two conditions that the real and imaginary parts of the determinantal expression be zero will serve to express any one in terms of the other two.

As a result of the definition (23), the quantity $(1 - ig)$ will be a factor in every term in X in the determinant of equation (19). A new variable

$$X_1 = (1 - ig)X \qquad (24)$$

may be introduced. Then, for each value of the parameter μ, the determinant in (19) can be expanded to form a polynomial equation in X_1 with complex coefficients. The roots of this polynomial equation will, in general, be complex; the real parts being equal to X and the ratio of the imaginary to the real part being $-g$. Each root of the polynomial will form for successive values of the parameter μ a branch of the solution of the flutter equation (19); these branches may be represented as lines in the three-dimensional g, X, μ space and each corresponds to a "flutter" mode. Spontaneous flutter will occur only when g is negative and the points where $g = 0$ correspond to flutter speeds. If the two variables X and μ are replaced by air-speed v and "flutter" frequency ω by using relations (21) and (22), these "flutter" modes may be plotted in a three-dimensional g, v, ω space. At any air-speed v the quantities g and ω which characterize the flutter modes can be picked out.

Instead of the three-dimensional plot, it will be more convenient in practice to make two two-dimensional plots: one a plot in the g-v plane, from which flutter speeds or approaches to flutter conditions can be obtained; and another plot in the ω-v plane, from which flutter frequencies can be obtained. These two plots will contain the principal results of a flutter analysis. The actual flutter mode shapes can be obtained by substitution of μ, X, and g into the flutter equation if desired.

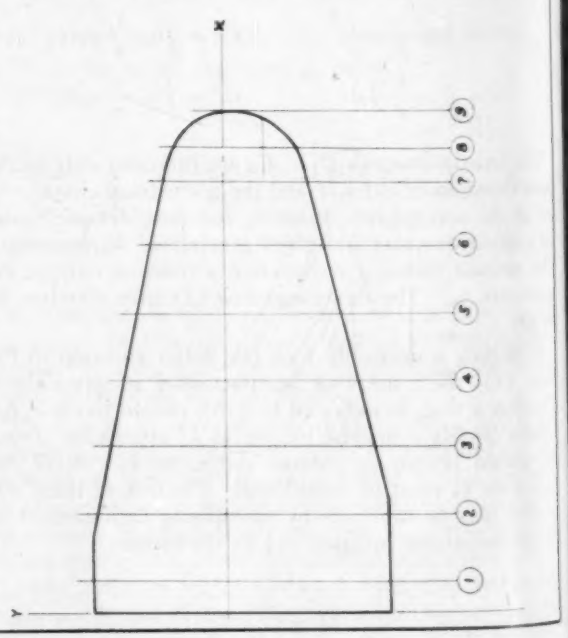
VI. Course and Arrangement of Calculations

As mentioned in the Introduction, the difficulties of the part of a flutter analysis described here are very largely computational ones. In this section, a detailed description of computational methods which have been found convenient is given; the actual arrangement of the numerical work step by step through the solution is shown and methods of numerical checking are described for each step. This section can serve both as a guide in carrying out a numerical analysis and as an index to analyses which have been carried out. At the end of this section, a set of pages are included showing the actual arrangement of the work sheets; these are described and referred to in the text. Also, a list of notations is included for convenience.

1. *Tabulation of Initial Data* – The initial data used in

SAMPLE WORK SHEET NO. 2

TYPICAL SCALE SKETCH LOCATING REF. STATIONS



[9] The operation of "adding" two damping factors may be defined in any convenient way. The operation defined by equation (23) results in $g_t = (g_{ij} - g) - ig_{ij}g$ and is very nearly equivalent to the intuitive meaning of addition $g_t = g_{ij} - g$ when both g_{ij} and g are small compared to "1." It is only when these damping factors are small that they have much quantitative significance.

SAMPLE WORK SHEET NO. 3

GENERALIZED COORDINATES FOR FLUTTER ANALYSIS

REFERENCE STATION	MATRIX H
E_1	[N COLUMNS, 4 ROWS]
E_2	[N COLUMNS, 4 ROWS]
E_3	[N COLUMNS, 4 ROWS]
E_4	[N COLUMNS, 4 ROWS]

setting up the flutter analysis described here are of two classes: (1) elastic and mass characteristics of structure in the form of the mass and elastic matrices a and f and the constants ω_r, v_r, and b_o, which enter directly into the flutter equation and calculation of flutter speeds, and (2) displacement shape matrix H and geometrical data θ and τ_R, which are functions of the span coordinates and enter into the integrals for the airforce matrix. The first class of data can be set down on a single sheet, as illustrated on Work Sheet No. 1. The form in which the latter class of data is required depends upon the method of integration. It is most convenient to carry out the integration by some approximate method, such as Simpson's rule, in terms of the values of the integrand at a set of selected reference stations. It will be convenient to indicate the reference stations chosen on a scale sketch of the aerodynamic surfaces; the sketch on Work Sheet No. 2 is an example.

The initial data of the second class mentioned above then consist in tabulating the values of H and θ and τ_R at the selected reference stations. These data may be conveniently tabulated in two tables arranged as indicated on Work Sheets Nos. 3 and 4.

SAMPLE WORK SHEET NO. 4

AIRFOIL GEOMETRICAL CHARACTERISTICS FOR FLUTTER ANALYSIS

REFERENCE STATION	τ_R	θ	$1/\theta$	$1/\theta^2$	ψ_1	ψ_2	ψ_3	ψ_4
E_1								
E_2								
E_3								
E_4								

In addition to the basic geometrical data θ and τ_R, other functions of θ that will be used are included in the table on Work Sheet No. 4.

2. *Steps in Forming Airforce Matrix* – The first step in forming the airforce matrix is to form the five matrix products

$$\lambda_i = H'\Theta\Lambda_i\Theta H \qquad (i = 1, 2, 3, 4, 5)$$

for each reference station. These products for each station are formed on a single sheet arranged as indicated on Work Sheet No. 5. The matrices $H'\Theta$ and ΘH are written down at the top of the sheet; the matrices Λ_i are written in a vertical column to the left. First, each product $\Lambda_i\Theta H$ is formed and written down opposite the corresponding factor Λ_i. Finally, the complete product $H'\Theta\Lambda_i\Theta H$ is formed and written in a column at the right. Each matrix multiplication is checked by forming the matrix product of the row of column sums of the first factor by the column of row sums of the second factor; this simply formed matrix product is a scalar and is equal to the sum of all the elements in the matrix product of the complete factors. The necessary row and column sums are indicated in the arrangement on Work Sheet No. 5. A work sheet similar to that on Work Sheet No. 5 must be made for each reference station.

The next step is the formation of the matrices C_k, C'_k, defined in Section IV. In each case the integrand is of form $f(\theta)\lambda_i$, and $f(\theta)$ can be evaluated for each reference station from the data which are given in a table of form

SAMPLE WORK SHEET NO. 5

FORMATION OF MATRIX PRODUCTS

$H'\Theta$ = [4 COLS., N ROWS] — COLUMN SUMS

ΘH = [N COLS., 4 ROWS] — ROW SUMS

i	Λ_i	$\Lambda_i\Theta H$	$\lambda_i = H'\Theta\Lambda_i\Theta H$
1	[4 COLS., 4 ROWS] COLUMN SUMS	[N COLS., 4 ROWS] ROW SUMS; SUM OF ELEMENTS	[N COLS., N ROWS] ROW SUMS; SUM OF ELEMENTS
2	[4 COLS., 4 ROWS] COLUMN SUMS	[N COLS., 4 ROWS] ROW SUMS; SUM OF ELEMENTS	[N COLS., N ROWS] ROW SUMS; SUM OF ELEMENTS
3	[4 COLS., 4 ROWS] COLUMN SUMS	[N COLS., 4 ROWS] ROW SUMS; SUM OF ELEMENTS	[N COLS., N ROWS] ROW SUMS; SUM OF ELEMENTS
4	[4 COLS., 4 ROWS] COLUMN SUMS	[N COLS., 4 ROWS] ROW SUMS; SUM OF ELEMENTS	[N COLS., N ROWS] ROW SUMS; SUM OF ELEMENTS
5	[4 COLS., 4 ROWS] COLUMN SUMS	[N COLS., 4 ROWS] ROW SUMS; SUM OF ELEMENTS	[N COLS., N ROWS] ROW SUMS; SUM OF ELEMENTS

SAMPLE WORK SHEET NO. 6

FORMATION OF MATRIX INTEGRAL C_i FOR AIRFORCES IN FLUTTER ANALYSIS

REF. STATION	λ_i	SUM OF MATRIX ELEMENTS
ξ_1	N COLUMNS N ROWS	
ξ_2	N COLUMNS N ROWS	
ξ_3	N COLUMNS N ROWS	
C_K	N COLUMNS N ROWS	

SAMPLE WORK SHEET NO. 7

SUMMARY OF MATRIX INTEGRALS C_i IN REAL PART OF AIRFORCE MATRIX

C_1' N COLUMNS N ROWS	C_3 N COLUMNS N ROWS
C_2 N COLUMNS N ROWS	C_6 N COLUMNS N ROWS
C_3 N COLUMNS N ROWS	C_7' N COLUMNS N ROWS
C_4 N COLUMNS N ROWS	C_8' N COLUMNS N ROWS

shown on Work Sheet No. 4. Since the integrals are to be evaluated by some process of numerical integration, they will be expressed as linear combinations of the values of the integrand at the reference stations, and $f(\theta)$ may be combined with the integration coefficient for each station. Hence the matrices C_k, C'_k are obtained directly as linear combinations of the matrices λ_i. A simple method of forming these linear combinations is to tabulate each λ_i on a single sheet and cut a stencil to lay over this table so that only corresponding elements of the λ_i are visible. If the coefficient of each λ_i is written beside the hole in the stencil showing an element of that matrix, the linear combinations can be easily formed. A stencil should be made for each C_k, C'_k, to correspond to the appropriate λ_i table. The work may be checked by forming the same linear combinations of the sums of the elements of the matrices λ_i, which should be equal to the sums of the elements of the matrices C_k, C'_k. The form of tables used in these calculations is shown schematically on Work Sheet No. 6.

The matrix C_1' [see equation (17)] is formed by adding a to C_1.

Finally, the actual values of the air-inertia matrix are found for a set of values of the parameter μ by forming the linear combinations of the C's indicated in equation (18). The coefficients for the linear combination of equation (18) are given in Table III of the Appendix. It is convenient to form these linear combinations by means of stencils as described above for the matrices C_k, C'_k. A stencil with the coefficients for each value of μ should be made, and for the real and imaginary parts of the airforce matrix. Summary sheets of the C matrices should be made for both real and imaginary parts of the airforce matrix and are illustrated on Work Sheets Nos. 7 and 8. The final values for the real and imaginary parts of the air-inertia matrix are entered in a table, the arrangement of which is indicated on Work Sheet No. 9.

3. *Expansion of Determinantal Flutter Equation* – The determinantal flutter equation (19) in X may now be set up for particular values of the parameter μ, from the air-inertia and the elastic matrices. A method of expanding the determinantal equation based upon the Laplace development is convenient to use. The Laplace development of a determinant is described in Chapter II of Reference 10. The method applied to third-, fourth-, and fifth-order determinants is described below, and a convenient arrangement of the work is indicated on Work Sheets Nos. 10, 11, and 12. The notation used in these outlines of the work is as follows:

a_{ij} = Element of determinant in "i"th row and "j"th column.

$D_{i \ldots k}$ = Minor determinant formed from the row numbers indicated in the subscript, the number of subscripts denotes the order; the columns from which these minors are taken will always be stated.

$\triangle$ = Complete expansion of determinant.

[10] See "Introduction to Higher Algebra," by M. Bocher, Macmillan, 1907.

The arrangement of work for the third-order determinant is shown on Work Sheet No. 10. Under the determinant in a column at the left, the expanded second-order minors from the second and third columns are placed. In the column to the right product of these second-order minors with the complementary element of the first column of the matrix are entered with the algebraic sign indicated. The sum of this column is the expanded determinant.

The arrangement of the work for expanding a fourth-order determinant is shown on Work Sheet No. 11. Here, the expansions of the second-order minors from the first two and the last two columns of the determinant are placed in two columns below the determinant in the order indicated on Work Sheet No. 11. A third column is formed of the products of the pairs of corresponding minors in these first two columns with the algebraic sign indicated. The sum of the terms in this column is the expanded determinant.

The arrangement of the work for expanding a fifth-order determinant is shown on Work Sheet No. 12. First, the expansions of the second-order minors in the fourth and fifth columns are formed and set down below the determinant in two columns. These minors are multiplied by elements in the third column of the determinant to form terms in the expansions of the third-order minors of the third, fourth, and fifth columns of the determinant. These products are placed in the array shown on Work Sheet No. 12. The sums of the rows of this array are the third-order minors and are set down in a column at the bottom of the sheet in the order shown. The second-order minors formed from the first and second columns of the determinant are placed beside the corresponding complementary third-order minors. The products of the corresponding third- and second-order minors with the algebraic sign indicated are set in a third column, the sum of which is the required expansion of the determinant. Similar arrangements of work may be made for higher-order determinants, if required.

A check of the work in expanding a determinantal equation may be obtained by expanding the numerical determinant formed by setting $X = 1$ in exactly the same way. Each term may be checked against the numerical one by setting $X = 1$.

The determinantal equation (19) must be written down and expanded on sheets similar to those on Work Sheets Nos. 10 to 12 for each value of the parameter chosen.

4. Solution of Expanded Polynomial and Presentation of Results – Solution by the method described in Section V calls for the determination of the roots of a polynomial equation with complex coefficients. If the polynomial is of the second, third, or fourth degree, exact solutions may be obtained algebraically. The algebraic solution of the quadratic with complex roots presents no difficulty; for a cubic or quartic, work sheets similar to those included as Work Sheets Nos. 13 and 14 of this report may be used conveniently. For higher-order equations, numerical methods can be used to find roots until the equation can be reduced to a fourth-degree equation and the algebraic solution used for the remainder of the roots.

SAMPLE WORK SHEET NO. 8

SUMMARY OF MATRIX INTEGRALS IN IMMAGINARY PART OF AIRFORCE MATRIX

C_9 N COLUMNS N ROWS	C_6 N COLUMNS N ROWS
C_3 N COLUMNS N ROWS	C_7 N COLUMNS N ROWS
C_4' N COLUMNS N ROWS	C_8 N COLUMNS N ROWS
C_5' N COLUMNS N ROWS	

SAMPLE WORK SHEET NO. 9

VALUES OF AIRFORCE MATRIX

VALUE OF μ	REAL PART	IMAGINARY PART
μ_1	N COLUMNS N ROWS	N COLUMNS N ROWS
μ_2	N COLUMNS N ROWS	N COLUMNS N ROWS
μ_i	N COLUMNS N ROWS	N COLUMNS N ROWS

As the parameter μ is changed, the roots of the polynomial vary continuously and form as many branches as there are roots.

A plot of the real and imaginary parts of the roots with μ will serve to identify the successive roots with the several branches of the solution. From the roots, a table of values of g, μ, X, v, and ω on each branch be made, and from the values in this table, the plots of the branches in the g-v and the ω-v planes can be made.

5. Summary of Steps in Flutter Analysis Computations – A summary of the computational steps necessary in carrying out the flutter analysis which has been described is presented in Table 1. In addition, a table defining the notation used in this section is included for convenience as Table 2.

Table 1 – Summary of Steps Necessary in Flutter Analysis Computations

Item No.	Description	Sample Work Sheet No.
1	Tabulation of mass and elastic data	1
2	Choice of method of integration for airforce matrix – Choice of reference stations and scale sketch showing locations	2
3	Tabulation of general coordinate shapes and geometrical data	3, 4
4	Formation of matrix products	5
5	Formation of integrands and integrals for matrices C_i	6, 7, 8
6	Formation of actual airforce matrix for values of parameter	9
7	Setting up and expanding determinantal flutter equation for values of μ	10, 11, 12
8	Solution of polynomial in X	13, 14
9	Plots of solution for flutter modes in g-v and ω-v planes	

The time required for carrying out several parts of the flutter analysis described here have been estimated and are summarized in Table 3. These times are given in man-hours and are based on making calculations with the latest

Table 2 – Notation Used in Section VI

Symbol	Brief Definitions	Section of Part II Where First Introduced and Defined
f	Elastic matrix	II
a	Inertia matrix	II
Γ	Air density ratio	II
ω_r	Reference frequency	II
v_r	Reference air speed	II
b_o	Reference length	II
H	Generalized coordinate matrix	II
ξ	Dimensionless span coordinate	IV
τ_R	Flap chord ratio	III
θ	Half chord divided by b_o	III
$\psi_1, \psi_2, \psi_3, \psi_4$	Geometrical and aerodynamic properties	IV
Θ	Matrix descriptive of geometry of airfoil	III
λ_i	Matrix product entering airforce matrix integrand	IV
Λ_i	Matrices describing aerodynamic properties of airfoil section	III
C_i, C_i'	Matrix constants entering airforce matrix	IV
μ	Airforce parameter	IV
a_{ij}	Element of determinant	VI
$D_{i \ldots k}$	Minor of determinant	VI
Δ	Expansion of determinant	VI
X	Variable in flutter equation	IV
X_1	Variable in flutter equation	V
ω	Frequency of vibration or flutter	II, III, IV
g	Damping in "flutter" mode	V
v	Air Speed	II

SAMPLE WORK SHEET NO. 10

EXPANSION OF THIRD ORDER DETERMINANT

$$\begin{vmatrix} a_{11} & a_{12} & a_{13} \\ a_{21} & a_{22} & a_{23} \\ a_{31} & a_{32} & a_{33} \end{vmatrix}$$

D_{23}	$+ a_{11}$	D_{23}
D_{13}	$- a_{21}$	D_{13}
D_{12}	$+ a_{31}$	D_{12}
		$\triangle$

SAMPLE WORK SHEET NO. 11

EXPANSION OF FOURTH ORDER DETERMINANT

$$\begin{vmatrix} a_{11} & a_{12} & a_{13} & a_{14} \\ a_{21} & a_{22} & a_{23} & a_{24} \\ a_{31} & a_{32} & a_{33} & a_{34} \\ a_{41} & a_{42} & a_{43} & a_{44} \end{vmatrix}$$

D_{12}	D_{34}	$+D_{12}\, D_{34}$
D_{13}	D_{24}	$-D_{13}\, D_{24}$
D_{14}	D_{23}	$+D_{14}\, D_{23}$
D_{23}	D_{14}	$+D_{23}\, D_{14}$
D_{24}	D_{13}	$-D_{24} D_{13}$
D_{34}	D_{12}	$+D_{34} D_{12}$
		$\triangle$

model Marchant calculating machine or the equivalent. The notation used in this table is as follows:

n = Number of generalized coordinates

A = Time required to obtain constant matrices $C_1 \ldots C_9$ occurring in the airforce matrix

B = Time required to obtain airforce matrix for *one* value of parameter μ

C = Time required to set up and expand determinantal flutter equation and solve the complex polynomial in X_1 for *one* value of the parameter μ

s = Number of reference stations used for airforce integrals

Table 3 – Estimated Time in Man-Hours Required to Carry Out Flutter Analysis

n	A	B	C
3	0.7 + 4.4s	0.9	4.0
4	1.3 + 7.4s	1.5	10.9
5	1.9 + 10.4s	2.3	30.
6	2.7 + 14.1s	3.3	81.
7	3.7 + 18.4s	4.4	220.
8	4.8 + 23.7s	5.7	590.

SAMPLE WORK SHEET NO. 12

EXPANSION OF FIFTH ORDER DETERMINENTAL EQUATIONS

$$\begin{vmatrix} a_{11} & a_{12} & a_{13} & a_{14} & a_{15} \\ a_{21} & a_{22} & a_{23} & a_{24} & a_{25} \\ a_{31} & a_{32} & a_{33} & a_{34} & a_{35} \\ a_{41} & a_{42} & a_{43} & a_{44} & a_{45} \\ a_{51} & a_{52} & a_{53} & a_{54} & a_{55} \end{vmatrix}$$

D_{12} D_{24}
D_{13} D_{25}
D_{14} D_{34}
D_{15} D_{35}
D_{23} D_{45}

$+a_{33} D_{45} - a_{43} D_{35} + a_{53} D_{34}$
$+a_{23} D_{45} - a_{43} D_{25} + a_{53} D_{24}$
$+a_{23} D_{35} - a_{33} D_{25} + a_{53} D_{23}$
$+a_{23} D_{34} - a_{33} D_{24} + a_{43} D_{23}$
$+a_{13} D_{45} - a_{43} D_{15} + a_{53} D_{14}$
$+a_{13} D_{35} - a_{33} D_{15} + a_{53} D_{13}$
$+a_{13} D_{34} - a_{33} D_{14} + a_{43} D_{13}$
$+a_{13} D_{25} - a_{23} D_{15} + a_{53} D_{12}$
$+a_{13} D_{24} - a_{23} D_{24} + a_{43} D_{12}$
$+a_{13} D_{23} - a_{23} D_{13} + a_{33} D_{12}$

D_{345}	D_{12}	$+D_{345} D_{12}$
D_{245}	D_{13}	$-D_{245} D_{13}$
D_{235}	D_{14}	$+D_{235} D_{14}$
D_{234}	D_{15}	$-D_{234} D_{15}$
D_{145}	D_{23}	$+D_{145} D_{23}$
D_{135}	D_{24}	$-D_{135} D_{24}$
D_{134}	D_{25}	$+D_{134} D_{25}$
D_{125}	D_{34}	$+D_{125} D_{34}$
D_{124}	D_{35}	$-D_{124} D_{35}$
D_{123}	D_{45}	$+D_{123} D_{45}$

Δ

SAMPLE WORK SHEET NO. 13

SOLUTION OF CUBIC $x^3+b_1x^2+c_1x+d_1=0$

$b_1=$		
$c_1=$		
$d_1=$		

$1/3 \cdot b_1=$		
$1/9 \cdot b_1^2=$		
$1/27 \cdot b_1^3=$		
$\gamma=1/3 \cdot b_1c_1-d=$		

$1/3 \cdot c_1-1/9 \cdot b_1^2=q=$		
$q^2=$		
$q^3=$		

$1/2\gamma-1/27 \cdot b_1^3=r=$		
$r^2=$		

	COMPLEX FORM	$(MOD)^2$	MOD.	TANφ	φ	COSφ	SINφ
q^3+r^2							
$\sqrt{q^3+r^2}$							
$r+\sqrt{q^3+r^2}=A_1^3$							
$r-\sqrt{q^3+r^2}=A_2^3$							
A_1							
A_2							

$A_1+A_2=$		
$A_1-A_2=$		

$-1/2(A_1+A_2)=$		
$i\sqrt{3}/2(A_1-A_2)=$		

$X_1=(A_1+A_2)-1/3 \cdot b_1=$		
$X_2=-1/2(A_1+A_2)+i\sqrt{3}/2(A_1-A_2)-1/3 \cdot b_1=$		
$X_3=-1/2(A_1+A_2)-i\sqrt{3}/2(A_1-A_2)-1/3 \cdot b_1=$		

CHECK: $X_1+X_2+X_3=-b_1=$		

$X_1X_2=$		
$X_1X_3=$		
$X_2X_3=$		
$X_1X_2+X_1X_3+X_2X_3=C_1=-3A_1A_2+1/3 \cdot b_1^2=$		

$X_1X_2X_3=-d=A_1^3+A_2^3+b_1A_1A_2-1/27 \cdot b_1^3=$		

EACH OF THESE ANGLES MAY HAVE $m \times 120°$ ADDED WHERE m = INTEGER; THE ANGLES OF A_1 AND A_2 MUST BE TAKEN SO THAT THEIR SUM EQUALS THE ANGLE OF $-q$ SINCE THE RELATION

$$A_1A_2=\{(r+\sqrt{q^3+r^2})(r-\sqrt{q^3+r^2})\}^{1/3}=(-q^3)^{1/3}=-q$$

HOLDS.

SAMPLE WORK SHEET NO. 14

SOLUTION OF BIQUADRATIC $x^4+bx^3+cx^2+dx+e=0$ WITH COMPLEX COEFFICIENTS

$b=$		
$c=$		
$d=$		
$e=$		

$1/4\ b^2$		
bd		
b^2-4c		
$e(b^2-4c)$		
d^2		

$b_1=-c=$		
$c_1=bd-4e=$		
$d_1=-[e(b^2-4c)+d^2]$		

SOLVE CUBIC: $y^3+b_1y^2+c_1y+d_1=0$ FOR ROOT $Y_1=$

$1/4\ b^2-c+y_1$		
$\alpha=\sqrt{1/4\ b^2-c+y_1}$		
$B_1=1/4\ b-1/2\ \alpha$		
$B_2=1/4\ b+1/2\ \alpha$		
$2\alpha-b$		
$(2\alpha-b)y_1+2d$		
$C_1=[(2\alpha-b)y_1+2d] \div 4\alpha$		
$2\alpha+b$		
$(2\alpha+b)y_1-2d$		

$C_2=[(2\alpha+b)y_1-2d] \div 4\alpha$		
$B_1^2-C_1$		
$B_2^2-C_2$		
$\sqrt{B_1^2-C_1}$		
$\sqrt{B_2^2-C_2}$		
$X_1=-B_1+\sqrt{B_1^2-C_1}=$		
$X_2=-B_1-\sqrt{B_1^2-C_1}=$		
$X_3=-B_2+\sqrt{B_2^2-C_2}=$		
$X_4=-B_2-\sqrt{B_2^2-C_2}=$		

CHECK: $X_1+X_2+X_3+X_4=-b=-2(B_1+B_2)=$		

$X_1X_2=$			$X_2X_3=$	
$X_1X_3=$			$X_2X_4=$	
$X_1X_3=$			$X_3X_4=$	

$X_1X_2+X_1X_3+X_1X_4+X_2X_3+X_2X_4+X_3X_4=C=C_1+C_2+4B_1B_2$		

$X_1X_2X_3$		
$X_1X_2X_4$		
$X_1X_3X_4$		
$X_2X_3X_4$		

$X_1X_2X_3+X_1X_2X_4+X_1X_3X_4+X_2X_3X_4=-d=-2(C_1B_2+B_1C_2)=$		

* $X_1X_2X_3X_4=e=C_1C_2=$

* THIS IS THE ONLY CHECK TERM THAT DEPENDS UPON THE VALUE OF Y_1.

APPENDIX TO PART II

The method of determining $\psi_1 \ldots \psi_4$ is given in this appendix. The airforce functions, F and G, can be represented approximately by:

$$F = F_0 + \psi_1 F_0' + \psi_2 F_0'' \quad (1)$$

$$G = G_0 + \psi_3 G_0' + \psi_4 G_0'' \quad (2)$$

where

$$\left.\begin{aligned} F_0' &= F - F_0 \\ G_0' &= G - G_0 \end{aligned}\right\} \text{ at } \theta = 0.40$$

$$\left.\begin{aligned} F_0'' &= F - F_0 \\ G_0'' &= G - G_0 \end{aligned}\right\} \text{ at } \theta = 2.00$$

The argument of F_0 and G_0 is k_0; the argument of F and G is θk_0. The values of $\psi_1 \ldots \psi_4$ at $\theta = 0.40$ and $\theta = 2.00$ are:

$\theta = 0.40$	$\psi_1 = 1$	$\psi_2 = 0$	$\psi_3 = 1$	$\psi_4 = 0$
$\theta = 2.00$	$\psi_1 = 0$	$\psi_2 = 1$	$\psi_3 = 0$	$\psi_4 = 1$

The functions $\psi_1 \ldots \psi_4$ are determined so that the equations (1) and (2) are satisfied as nearly as possible, or:

$$\int_0^1 (F - F_0 - \psi_1 F_0' - \psi_2 F_0'')^2 d\mu = \text{min.} \quad (3)$$

$$\int_0^1 (G - G_0 - \psi_3 G_0' - \psi_4 G_0'')^2 d\mu = \text{min.} \quad (4)$$

where

$$\mu = \frac{k_0}{\sqrt{1 + k_0^2}}$$

Table 2 – A

	$\theta = 0.20$		$\theta = 0.30$		$\theta = 0.50$		$\theta = 0.60$	
μ	A	B	A	B	A	B	A	B
0	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000
0.05	0.9996	0.9822	0.9727	0.9655	0.9544	0.9573	0.9451	0.9486
0.10	0.9635	0.9521	0.9449	0.9400	0.9089	0.9104	0.8916	0.8935
0.15	0.9446	0.9447	0.9172	0.9160	0.8670	0.8447	0.8448	0.8240
0.20	0.9253	0.9364	0.8901	0.8928	0.8293	0.8282	0.8038	0.8021
0.25	0.9065	0.9226	0.8644	0.8688	0.7959	0.7940	0.7680	0.7659
0.30	0.8875	0.9056	0.8400	0.8450	0.7656	0.7641	0.7370	0.7350
0.35	0.8688	0.8858	0.8165	0.8212	0.7386	0.7372	0.7097	0.7081
0.40	0.8501	0.8640	0.7938	0.7975	0.7139	0.7129	0.6855	0.6843
0.45	0.8309	0.8411	0.7719	0.7744	0.6915	0.6907	0.6639	0.6631
0.50	0.8121	0.8172	0.7505	0.7518	0.6708	0.6706	0.6444	0.6442
0.55	0.7941	0.7927	0.7296	0.7295	0.6516	0.6519	0.6266	0.6270
0.60	0.7731	0.7676	0.7091	0.7075	0.6335	0.6343	0.6103	0.6111
0.65	0.7525	0.7420	0.6886	0.6857	0.6165	0.6177	0.5951	0.5964
0.70	0.7309	0.7167	0.6680	0.6643	0.6002	0.6015	0.5808	0.5822
0.75	0.7079	0.6885	0.6469	0.6419	0.5845	0.5862	0.5672	0.5692
0.80	0.6828	0.6600	0.6250	0.6193	0.5690	0.5709	0.5542	0.5563
0.85	0.6544	0.6293	0.6016	0.5954	0.5535	0.5554	0.5414	0.5435
0.90	0.6209	0.5954	0.5757	0.5696	0.5376	0.5394	0.5285	0.5304
0.95	0.5775	0.5556	0.5450	0.5400	0.5205	0.5219	0.5152	0.5168
1.00	0.5000	0.5000	0.5000	0.5000	0.5000	0	0.5000	0.5000

	$\theta = 0.70$		$\theta = 0.80$		$\theta = 0.90$		$\theta = 1.10$	
μ	A	B	A	B	A	B	A	B
0	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000
0.05	0.9358	0.9388	0.9266	0.9287	0.9179	0.9188	0.9005	0.8996
0.10	0.8757	0.8769	0.8599	0.8608	0.8455	0.8456	0.8184	0.8176
0.15	0.8239	0.8048	0.8049	0.7874	0.7876	0.7864	0.7584	0.7569
0.20	0.7804	0.7791	0.7597	0.7587	0.7412	0.7407	0.7094	0.7096
0.25	0.7436	0.7419	0.7226	0.7212	0.7041	0.7034	0.6732	0.6736
0.30	0.7124	0.7109	0.6917	0.6905	0.6737	0.6732	0.6448	0.6450
0.35	0.6857	0.6844	0.6655	0.6647	0.6485	0.6482	0.6216	0.6219
0.40	0.6624	0.6614	0.6433	0.6426	0.6274	0.6271	0.6028	0.6025
0.45	0.6419	0.6412	0.6240	0.6236	0.6093	0.6091	0.5868	0.5869
0.50	0.6237	0.6236	0.6071	0.6071	0.5937	0.5937	0.5734	0.5744
0.55	0.6074	0.6077	0.5922	0.5924	0.5800	0.5802	0.5619	0.5617
0.60	0.5926	0.5933	0.5788	0.5793	0.5679	0.5682	0.5519	0.5516
0.65	0.5790	0.5802	0.5667	0.5675	0.5571	0.5575	0.5431	0.5423
0.70	0.5665	0.5676	0.5556	0.5564	0.5473	0.5476	0.5353	0.5350
0.75	0.5547	0.5563	0.5454	0.5465	0.5382	0.5368	0.5282	0.5278
0.80	0.5436	0.5453	0.5359	0.5369	0.5299	0.5305	0.5212	0.5214
0.85	0.5328	0.5345	0.5266	0.5278	0.5221	0.5226	0.5159	0.5154
0.90	0.5222	0.5238	0.5179	0.5189	0.5147	0.5151	0.5103	0.5098
0.95	0.5116	0.5127	0.5092	0.5098	0.5074	0.5076	0.5061	0.5047
1.00	0.5000	0.5000	0.5000	0.5000	0.5000	0.5000	0.5000	0.5000

	$\theta = 1.20$		$\theta = 1.30$		$\theta = 1.40$		$\theta = 1.50$	
μ	A	B	A	B	A	B	A	B
0	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000
0.05	0.8919	0.8903	0.8839	0.8817	0.8760	0.8736	0.8681	0.8657
0.10	0.8056	0.8049	0.7941	0.7930	0.7833	0.7820	0.7723	0.7713
0.15	0.7426	0.7431	0.7299	0.7306	0.7183	0.7190	0.7074	0.7081
0.20	0.6958	0.6964	0.6835	0.6844	0.6723	0.6734	0.6621	0.6632
0.25	0.6603	0.6610	0.6488	0.6497	0.6385	0.6396	0.6293	0.6304
0.30	0.6327	0.6334	0.6222	0.6231	0.6129	0.6140	0.6047	0.6057
0.35	0.6107	0.6113	0.6013	0.6020	0.5930	0.5937	0.5856	0.5864
0.40	0.5928	0.5932	0.5844	0.5848	0.5771	0.5775	0.5706	0.5711
0.45	0.5781	0.5783	0.5706	0.5709	0.5641	0.5644	0.5585	0.5589
0.50	0.5656	0.5655	0.5590	0.5589	0.5534	0.5532	0.5485	0.5484
0.55	0.5551	0.5548	0.5493	0.5490	0.5444	0.5440	0.5406	0.5398
0.60	0.5459	0.5455	0.5409	0.5404	0.5367	0.5361	0.5331	0.5325
0.65	0.5379	0.5375	0.5337	0.5330	0.5300	0.5293	0.5269	0.5263
0.70	0.5309	0.5305	0.5272	0.5268	0.5242	0.5237	0.5217	0.5213
0.75	0.5245	0.5240	0.5216	0.5208	0.5191	0.5182	0.5171	0.5161
0.80	0.5189	0.5182	0.5165	0.5157	0.5146	0.5136	0.5129	0.5120
0.85	0.5137	0.5130	0.5119	0.5110	0.5104	0.5095	0.5093	0.5083
0.90	0.5089	0.5082	0.5077	0.5069	0.5066	0.5058	0.5059	0.5051
0.95	0.5044	0.5038	0.5037	0.5031	0.5032	0.5026	0.5029	0.5022
1.00	0.5000	0.5000	0.5000	0.5000	0.5000	0.5000	0.5000	0.5000

	$\theta = 1.60$		$\theta = 1.80$		$\theta = 2.20$		$\theta = 2.40$	
μ	A	B	A	B	A	B	A	B
0	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000
0.05	0.8602	0.8581	0.8459	0.8444	0.8188	0.8205	0.8061	0.8102
0.10	0.7620	0.7612	0.7435	0.7432	0.7118	0.7124	0.6981	0.6993
0.15	0.6972	0.6978	0.6792	0.6797	0.6496	0.6492	0.6375	0.6365
0.20	0.6527	0.6536	0.6363	0.6370	0.6105	0.6098	0.6003	0.5985
0.25	0.6209	0.6218	0.6064	0.6070	0.5843	0.5834	0.5757	0.5738
0.30	0.5972	0.5981	0.5848	0.5851	0.5658	0.5650	0.5586	0.5567
0.35	0.5792	0.5798	0.5682	0.5686	0.5522	0.5516	0.5462	0.5450
0.40	0.5650	0.5653	0.5555	0.5557	0.5418	0.5416	0.5368	0.5361
0.45	0.5536	0.5539	0.5454	0.5457	0.5339	0.5342	0.5296	0.5296
0.50	0.5442	0.5441	0.5373	0.5370	0.5268	0.5275	0.5238	0.5240
0.55	0.5365	0.5361	0.5305	0.5302	0.5222	0.5224	0.5192	0.5197
0.60	0.5299	0.5294	0.5248	0.5244	0.5180	0.5183	0.5155	0.5162
0.65	0.5243	0.5237	0.5200	0.5196	0.5144	0.5148	0.5124	0.5133
0.70	0.5195	0.5193	0.5160	0.5162	0.5133	0.5130	0.5098	0.5121
0.75	0.5153	0.5145	0.5125	0.5119	0.5087	0.5093	0.5075	0.5087
0.80	0.5116	0.5107	0.5094	0.5088	0.5065	0.5071	0.5056	0.5068
0.85	0.5083	0.5074	0.5067	0.5061	0.5046	0.5052	0.5039	0.5051
0.90	0.5053	0.5045	0.5042	0.5037	0.5029	0.5034	0.5024	0.5035
0.95	0.5025	0.5019	0.5020	0.5016	0.5013	0.5017	0.5011	0.5019
1.00	0.5000	0.5000	0.5000	0.5000	0.5000	0.5000	0.5000	0.5000

A is the column of actual values of F
B is the column of values of $F_0 + \psi_1 F_0' + \psi_2 F_0''$

Then, from (3) and (4):

$$\psi_1\int_0^1 (F_0')^2 d\mu + \psi_2\int_0^1 F_0'F_0''d\mu = \int_0^1 (F-F_0)F_0'd\mu$$

$$\psi_1\int_0^1 F_0'F_0''d\mu + \psi_2\int_0^1 (F_0'')^2 d\mu = \int_0^1 (F-F_0)F_0''d\mu$$

with similar equations for ψ_3 and ψ_4. The values of $\psi_2 \ldots \psi_4$ for values of θ varying from 0.20 to 2.40 are given in Table 1. The degree of approximation of equations (1) and (2) is shown in Table 2 by a comparison of actual and approximate values of F and G. Table 3 gives the values of the required functions of μ.

Table 1

θ	ψ_1	ψ_2	ψ_3	ψ_4
0.20	2.3095	0.9481	2.0789	0.9886
0.30	1.5272	0.3497	1.4618	0.3389
0.40	1.0000	0	1.0000	0
0.50	0.6505	−0.1648	0.6814	−0.1593
0.60	0.4121	−0.2205	0.4330	−0.2243
0.70	0.2473	−0.2109	0.2656	−0.2135
0.80	0.1329	−0.1611	0.1348	−0.1739
0.90	0.0539	−0.0880	0.0551	−0.0934
1.00	0	0	0	0
1.10	−0.0359	0.0964	−0.0429	0.0970
1.20	−0.0553	0.2024	−0.0693	0.2002
1.30	−0.0665	0.3059	−0.0823	0.3054
1.40	−0.0711	0.4073	−0.0867	0.4090
1.50	−0.0667	0.5133	−0.0850	0.5115
1.60	−0.0584	0.6176	−0.0798	0.6122
1.80	−0.0360	0.8101	−0.0457	0.8112
2.00	0	1.0000	0	1.0000
2.20	0.0354	1.1701	0.0280	1.1572
2.40	0.0730	1.3292	0.0712	1.3148

Table 2 – B

	$\theta = 0.20$		$\theta = 0.30$		$\theta = 0.50$		$\theta = 0.60$	
μ	C	D	C	D	C	D	C	D
0	0	0	0	0	0	0	0	0
0.05	0.0377	0.0614	0.0565	0.0655	0.0855	0.0851	0.0957	0.0955
0.10	0.0754	0.0717	0.0959	0.0958	0.1296	0.1316	0.1429	0.1446
0.15	0.0963	0.0895	0.1233	0.1208	0.1567	0.1581	0.1670	0.1693
0.20	0.1171	0.1081	0.1436	0.1406	0.1731	0.1739	0.1807	0.1822
0.25	0.1315	0.1274	0.1581	0.1568	0.1823	0.1826	0.1869	0.1875
0.30	0.1452	0.1436	0.1689	0.1689	0.1872	0.1870	0.1886	0.1888
0.35	0.1557	0.1578	0.1770	0.1780	0.1887	0.1882	0.1874	0.1870
0.40	0.1647	0.1703	0.1827	0.1848	0.1878	0.1872	0.1840	0.1833
0.45	0.1726	0.1804	0.1864	0.1892	0.1850	0.1843	0.1790	0.1780
0.50	0.1783	0.1872	0.1883	0.1913	0.1808	0.1801	0.1728	0.1719
0.55	0.1829	0.1919	0.1886	0.1914	0.1753	0.1748	0.1657	0.1649
0.60	0.1862	0.1948	0.1873	0.1899	0.1687	0.1684	0.1577	0.1571
0.65	0.1882	0.1946	0.1845	0.1864	0.1609	0.1610	0.1489	0.1487
0.70	0.1886	0.1929	0.1801	0.1812	0.1521	0.1524	0.1392	0.1393
0.75	0.1872	0.1885	0.1737	0.1740	0.1419	0.1426	0.1285	0.1289
0.80	0.1834	0.1808	0.1650	0.1641	0.1300	0.1311	0.1165	0.1172
0.85	0.1762	0.1695	0.1529	0.1510	0.1159	0.1172	0.1026	0.1035
0.90	0.1631	0.1515	0.1355	0.1323	0.0979	0.0998	0.0855	0.0870
0.95	0.1368	0.1333	0.1068	0.1107	0.0727	0.0772	0.0621	0.0652
1.00	0	0	0	0	0	0	0	0

	$\theta = 0.70$		$\theta = 0.80$		$\theta = 0.90$		$\theta = 1.10$	
μ	C	D	C	D	C	D	C	D
0	0	0	0	0	0	0	0	0
0.05	0.1059	0.1049	0.1161	0.1136	0.1228	0.1214	0.1361	0.1350
0.10	0.1518	0.1541	0.1606	0.1621	0.1666	0.1679	0.1764	0.1765
0.15	0.1747	0.1764	0.1804	0.1818	0.1841	0.1846	0.1880	0.1874
0.20	0.1853	0.1865	0.1878	0.1890	0.1887	0.1892	0.1874	0.1871
0.25	0.1887	0.1888	0.1883	0.1888	0.1868	0.1867	0.1814	0.1810
0.30	0.1877	0.1876	0.1850	0.1853	0.1815	0.1815	0.1729	0.1729
0.35	0.1840	0.1836	0.1795	0.1795	0.1743	0.1743	0.1634	0.1636
0.40	0.1786	0.1779	0.1724	0.1722	0.1660	0.1659	0.1535	0.1537
0.45	0.1719	0.1710	0.1645	0.1641	0.1572	0.1570	0.1435	0.1437
0.50	0.1644	0.1636	0.1560	0.1556	0.1480	0.1478	0.1336	0.1337
0.55	0.1562	0.1556	0.1471	0.1467	0.1387	0.1385	0.1239	0.1239
0.60	0.1473	0.1469	0.1378	0.1374	0.1291	0.1290	0.1142	0.1142
0.65	0.1379	0.1378	0.1281	0.1279	0.1194	0.1193	0.1046	0.1045
0.70	0.1279	0.1280	0.1180	0.1179	0.1093	0.1093	0.0950	0.0948
0.75	0.1171	0.1175	0.1073	0.1073	0.0988	0.0988	0.0852	0.0848
0.80	0.1052	0.1059	0.0956	0.0959	0.0877	0.0878	0.0748	0.0745
0.85	0.0918	0.0926	0.0829	0.0830	0.0754	0.0754	0.0637	0.0632
0.90	0.0756	0.0769	0.0677	0.0683	0.0613	0.0616	0.0515	0.0510
0.95	0.0546	0.0564	0.0483	0.0490	0.0435	0.0437	0.0362	0.0359
1.00	0	0	0	0	0	0	0	0

	$\theta = 1.20$		$\theta = 1.30$		$\theta = 1.40$		$\theta = 1.50$	
μ	C	D	C	D	C	D	C	D
0	0	0	0	0	0	0	0	0
0.05	0.1427	0.1409	0.1471	0.1462	0.1516	0.1510	0.1561	0.1554
0.10	0.1802	0.1796	0.1826	0.1820	0.1850	0.1840	0.1863	0.1855
0.15	0.1887	0.1878	0.1886	0.1876	0.1881	0.1870	0.1871	0.1862
0.20	0.1857	0.1853	0.1836	0.1831	0.1811	0.1807	0.1785	0.1781
0.25	0.1780	0.1776	0.1744	0.1740	0.1706	0.1704	0.1668	0.1667
0.30	0.1683	0.1684	0.1637	0.1638	0.1591	0.1593	0.1546	0.1549
0.35	0.1580	0.1582	0.1527	0.1530	0.1476	0.1480	0.1427	0.1432
0.40	0.1475	0.1478	0.1418	0.1422	0.1365	0.1369	0.1314	0.1319
0.45	0.1372	0.1375	0.1313	0.1318	0.1258	0.1264	0.1207	0.1214
0.50	0.1272	0.1273	0.1212	0.1215	0.1157	0.1161	0.1107	0.1110
0.55	0.1174	0.1174	0.1115	0.1116	0.1061	0.1062	0.1012	0.1012
0.60	0.1079	0.1077	0.1006	0.1020	0.0968	0.0968	0.0921	0.0920
0.65	0.0984	0.0981	0.0928	0.0925	0.0878	0.0875	0.0833	0.0829
0.70	0.0891	0.0886	0.0837	0.0832	0.0789	0.0785	0.0747	0.0742
0.75	0.0795	0.0790	0.0745	0.0739	0.0700	0.0695	0.0661	0.0655
0.80	0.0695	0.0690	0.0650	0.0644	0.0611	0.0603	0.0576	0.0567
0.85	0.0593	0.0583	0.0554	0.0542	0.0518	0.0506	0.0487	0.0475
0.90	0.0477	0.0468	0.0443	0.0434	0.0414	0.0404	0.0389	0.0379
0.95	0.0333	0.0330	0.0309	0.0309	0.0288	0.0291	0.0268	0.0278
1.00	0	0	0	0	0	0	0	0

	$\theta = 1.60$		$\theta = 1.80$		$\theta = 2.20$		$\theta = 2.40$	
μ	C	D	C	D	C	D	C	D
0	0	0	0	0	0	0	0	0
0.05	0.1605	0.1596	0.1685	0.1685	0.1763	0.1779	0.1801	0.1824
0.10	0.1878	0.1868	0.1887	0.1881	0.1876	0.1896	0.1861	0.1897
0.15	0.1859	0.1852	0.1827	0.1823	0.1746	0.1766	0.1703	0.1735
0.20	0.1757	0.1755	0.1698	0.1697	0.1579	0.1592	0.1521	0.1541
0.25	0.1631	0.1631	0.1556	0.1557	0.1418	0.1426	0.1354	0.1366
0.30	0.1503	0.1506	0.1420	0.1422	0.1273	0.1276	0.1208	0.1211
0.35	0.1381	0.1385	0.1294	0.1297	0.1145	0.1145	0.1081	0.1078
0.40	0.1266	0.1270	0.1179	0.1182	0.1032	0.1030	0.0970	0.0965
0.45	0.1159	0.1165	0.1073	0.1079	0.0931	0.0933	0.0873	0.0872
0.50	0.1060	0.1062	0.0978	0.0978	0.0839	0.0837	0.0783	0.0779
0.55	0.0965	0.0965	0.0886	0.0885	0.0756	0.0752	0.0703	0.0698
0.60	0.0877	0.0875	0.0800	0.0800	0.0678	0.0676	0.0630	0.0627
0.65	0.0791	0.0786	0.0718	0.0715	0.0607	0.0601	0.0563	0.0557
0.70	0.0707	0.0702	0.0640	0.0638	0.0541	0.0535	0.0500	0.0496
0.75	0.0625	0.0618	0.0585	0.0581	0.0474	0.0470	0.0437	0.0437
0.80	0.0546	0.0534	0.0491	0.0484	0.0408	0.0406	0.0377	0.0379
0.85	0.0458	0.0447	0.0411	0.0405	0.0343	0.0342	0.0316	0.0321
0.90	0.0367	0.0356	0.0328	0.0325	0.0270	0.0278	0.0248	0.0264
0.95	0.0252	0.0266	0.0250	0.0255	0.0183	0.0243	0.0170	0.0244
1.00	0	0	0	0	0	0	0	0

C is the column of actual values of G
D is the column of values of $G_0 + \psi_3 G_0' + \psi_4 G_0''$.

Table 3 – Functions of μ Entering Airforce Matrix

μ	k_o	$\frac{k_o^2}{1+k_o^2}$	$\frac{1}{1+k_o^2}$	$\frac{2F_o}{1+k_o^2}$	+	$\frac{2F_o'}{1+k_o^2}$	$\frac{2F_o''}{1+k_o^2}$	−	$\frac{2G_o k_o}{1+k_o^2}$	$\frac{2G_o' k_o}{1+k_o^2}$	−	$\frac{2G_o'' k_o}{1+k_o^2}$	$\frac{k_o}{1+k_o^2}$	$\frac{2G_o}{1+k_o^2}$	$\frac{2G_o'}{1+k_o^2}$	$\frac{2G_o''}{1+k_o^2}$	$\frac{2F_o k_o}{1+k_o^2}$	$\frac{2F_o' k_o}{1+k_o^2}$	$\frac{2F_o'' k_o}{1+k_o^2}$
0	0	0	1	2.0000		0	0		0	0		0	0	0	0	0	0	0	0
0.05	0.0501	0.0026	0.9975	1.8135		0.1091	−0.1542		0.01263	−0.00533		0.00219	0.0500	−0.2562	0.1061	−0.0878	0.0908	0.00547	−0.00772
0.1	0.1005	0.01	0.9900	1.6460		0.1883	−0.2069		0.03433	−0.01119		0.00321	0.0995	−0.3416	0.1113	−0.0319	0.1654	0.01893	−0.02078
0.15	0.1517	0.0225	0.9775	1.5085		0.2332	−0.2119		0.06525	−0.01279		0.00219	0.1483	−0.3642	0.0843	−0.0145	0.2288	0.03539	−0.03208
0.2	0.2041	0.04	0.9600	1.3910		0.2565	−0.1960		0.07382	−0.01058		0.00964	0.1959	−0.3617	0.0518	−0.0472	0.2839	0.05236	−0.04001
0.25	0.2582	0.0625	0.9375	1.2892		0.2625	−0.1749		0.08908	−0.00804		0.00719	0.2421	−0.3450	0.0195	−0.0666	0.3329	0.06778	−0.04517
0.3	0.3145	0.09	0.9100	1.1979		0.2575	−0.1527		0.10148	−0.00240		0.02461	0.2862	−0.3227	−0.0076	−0.0783	0.3767	0.08099	−0.04803
0.35	0.3736	0.1225	0.8775	1.1128		0.2450	−0.1311		0.11074	0.01128		0.03106	0.3278	−0.2964	−0.0302	−0.0832	0.4158	0.09152	−0.04898
0.4	0.4364	0.16	0.8400	1.0315		0.2273	−0.1109		0.11708	0.02100		0.03629	0.3666	−0.2683	−0.0482	−0.0833	0.4502	0.09919	−0.04838
0.45	0.5039	0.2025	0.7975	0.9524		0.2062	−0.0920		0.12073	0.03078		0.04019	0.4019	−0.2396	−0.0611	−0.0798	0.4799	0.10391	−0.04638
0.5	0.5774	0.25	0.7500	0.8739		0.1836	−0.0782		0.12177	0.04005		0.04350	0.4330	−0.2109	−0.0694	−0.0753	0.5046	0.10600	−0.04400
0.55	0.6585	0.3025	0.6975	0.7963		0.1597	−0.0617		0.12033	0.04855		0.04534	0.4593	−0.1827	−0.0737	−0.0689	0.5237	0.10518	−0.04061
0.6	0.7500	0.36	0.6400	0.7156		0.1366	−0.0488		0.11645	0.05568		0.04582	0.4800	−0.1553	−0.0742	−0.0611	0.5387	0.10166	−0.03858
0.65	0.8554	0.4225	0.5775	0.6346		0.1117	−0.0375		0.11026	0.06110		0.04539	0.4940	−0.1289	−0.0714	−0.0530	0.5428	0.09554	−0.03211
0.7	0.9803	0.49	0.5099	0.5513		0.0887	−0.0267		0.10167	0.06435		0.04314	0.4999	−0.1037	−0.0657	−0.0440	0.5404	0.08698	−0.02619
0.75	1.1339	0.5625	0.4375	0.4661		0.0670	−0.0195		0.09082	0.06519		0.03955	0.4961	−0.0801	−0.0575	−0.0349	0.5285	0.07600	−0.02213
0.8	1.3333	0.64	0.3600	0.3783		0.0472	−0.0127		0.07754	0.06278		0.03478	0.4800	−0.0582	−0.0471	−0.0261	0.5044	0.06288	−0.01690
0.85	1.6135	0.7225	0.2775	0.2878		0.0296	−0.0073		0.06177	0.05678		0.02819	0.4477	−0.0383	−0.0352	−0.0175	0.4644	0.04773	−0.01173
0.9	2.0647	0.81	0.1900	0.1946		0.0150	−0.0033		0.04566	0.04393		0.02018	0.3923	−0.0213	−0.0221	−0.0098	0.4019	0.03107	−0.00682
0.95	3.0419	0.9025	0.0975	0.0967		0.0045	−0.0009		0.02046	0.03086		−0.00848	0.2966	−0.0067	−0.0101	−0.0028	0.3001	0.01382	−0.00261
1.0		1.00	0	0		0	0		0	0		0	0	0	0	0	0	0	0

$$\mu = \frac{k_o}{\sqrt{1+k_o^2}}$$

Part III

I. Introduction

1. Phase of Flutter Problem Treated – It was pointed out in Section I of Part II that the whole problem of flutter analysis by the method described in Reference 1 consists of two more or less distinct problems. The first problem is the determination of the generalized coordinates to use, and is the subject of the present part. The second problem is the determination of the flutter characteristics after the generalized coordinates have been chosen; this phase of the flutter analysis has been described in Part II. Since a number of the present arguments follow from results in Part II, this part logically follows Part II, although the subject matter treated here must be covered first in an actual flutter analysis.

2. General Type of Structure Treated – The flutter analysis described in Part II was based on a set of airforce expressions for the section airforces on an airfoil with flap in two-dimensional airflow, and is, therefore, restricted to the treatment of elongated aerodynamic surfaces of this form. The displacements of the surfaces are limited to rotation and translation of chordwise sections of airfoil or flap and can generally be attributed to twisting or bending of the airfoil and flap or to deflections of the supporting structure. The arguments in this paper concerning displacements will be phrased in terms of these types of movements although many of them are of much wider applicability.

3. Practical Aspects of the Flutter Problem – A large proportion of flutter analyses are concerned with similar structures of successive airplane designs such as wings, single tail surfaces, and twin tail surfaces. The members of each class of structure will most often be similar enough so that the shapes of the possible critical flutter modes will be of the same general character. In these cases, the choice of satisfactory generalized coordinates will be determined largely from experience and familiarity with previous analyses and vibration tests, and should eventually present no particular difficulties. The problem which does present the greatest practical difficulties is the making of the first flutter analysis of a class of structures which has not previously been analyzed or built and tested, and for which all specific knowledge of the flutter modes is lacking. It is this phase of the flutter problem that is treated in the present part. The considerations and methods discussed should make it possible to carry out flutter analyses of new types of structures and to build up specific experience useful in analyzing other similar structures.

4. Manner of Treatment of Problem – The actual generalized displacements which should be used in a flutter analysis depend to such a degree on the specific type of structure to be analyzed that the general problem of choosing them cannot be treated as specifically as the subsequent flutter analysis described in Part II. For this reason, the present part will discuss, in general terms, what conditions enter into determination of the shapes and number of generalized coordinates to be used and general methods by which they can be obtained in any case.

II. Significance of Normal Coordinates

1. Representation of Flutter Modes – Assuming that the airforce expressions used are correct, the precision of the

results of the flutter analysis by the method of Reference 1 depends upon the precision with which the actual flutter mode can be represented as a linear combination of the generalized coordinates. The determination of the generalized coordinates consists, then, of finding a satisfactory set of independent components of the flutter mode. For computational ease, it is desirable to have generalized coordinates shaped so that the smallest possible number is satisfactory.

Any flutter mode can be represented in terms of the normal coordinates[11] so that the use of normal generalized coordinates would always be satisfactory if enough of them are included. A number of considerations are discussed in the succeeding paragraphs which indicate that normal generalized coordinates tend to be shaped so that the smallest number is required.

2. *Geometrical Considerations* – Any flutter mode and any set of generalized coordinates must have the following two properties:

1. They must satisfy the conditions of restraint on the structure.

2. The displacements in each must be continuous functions of the space coordinates within the structure.

The normal coordinates have these properties, of course; they also have the property of orthogonality and a definite tendency of being ordered (the order being determined by the associated natural frequencies) according to complexity. The latter characteristics tend to make a finite set of consecutive normal coordinates including the fundamental optimum for representing the widest variety of displacement shapes up to the order of complexity of the highest ordered coordinate. Lacking specific knowledge concerning the actual shape of the flutter mode, a consecutive set of normal coordinates including the fundamental would, therefore, tend to be optimum generalized coordinates from purely geometrical considerations.

3. *Stiffness Considerations* – The distortions of elongated structures being considered are due almost entirely to bending and twisting under the action of bending and twisting moments. Since the bending and twisting moments are integrals of the loading, they are continuous and tend to be smooth functions of the space coordinates of the structure. Abrupt changes in the stiffness of the structure will, therefore, reflect themselves as "breaks" in the distortion form of any flutter mode. The lack of abrupt stiffness changes or continuous stiffness variation reflects negatively in any flutter mode by making the distortion form smooth. The effects of the stiffness variation will reflect similarly in all the normal coordinates which will, therefore, constitute a desirable set of generalized coordinates from the point of view of effects of stiffness variation on the shape of the flutter mode.

4. *Mass Considerations* – The variations in mass distribution have some effect on the shape of any flutter mode, since concentrated masses cause concentrated inertia loadings and abrupt changes in mass distribution cause abrupt changes in inertia loadings. As in the case of stiffness variations, these effects will reflect similarly in the shapes of the normal modes.

5. *Aerodynamic Considerations* – The aerodynamic loadings always tend to be smooth functions of the space coordinates of a structure and will, therefore, introduce no "breaks" or discontinuities in the space derivatives of the flutter mode shapes.

6. *Summary of Conclusions* – A flutter mode shape will be a continuous function of the space coordinates within a structure and "breaks" or discontinuities in its space derivatives are introduced only by discontinuities of the stiffness or mass distribution of the structure. The "breaks" and discontinuities of the space derivatives of a flutter mode shape are, therefore, reflected in the normal coordinates. These coordinates, accordingly, make desirable generalized coordinates to use in a flutter analysis. The properties of the normal coordinates are such that the consecutive set, including the fundamental, tend to make an optimum set of generalized coordinates if specific knowledge of the flutter mode is lacking.

III. Number of Generalized Coordinates Needed

1. *Effect of Frequency on Flutter Speed* – If the structure analyzed has one or more flutter speeds, the lowest one is of greatest practical interest. The form of the expression for the flutter speed equation (61), Reference 1, shows that the flutter speed is proportional to the flutter frequency. The lowest flutter speed will, therefore, tend to occur at a low frequency and its mode shape to have as components the lower-order normal modes.

2. *Role of Higher-Order Normal Modes in Flutter Based on an Examination of Flutter Equations* – If normal generalized coordinates are used, the matrices a and f in the flutter equation

$$\left| c - \Gamma a + \Upsilon f \right| = 0$$

[Equation (59), Reference 1] will be diagonal. The only nondiagonal terms are the elements of the airforce matrix c. The nondiagonal elements of c in the "i"th column and the "j"th row are equal to the integral along the span of a homogeneous quadratic expression in the shapes of the "i"th and "j"th generalized coordinates containing cross-products but no squared terms. If either or both of the "i"th or "j"th generalized coordinates represent higher modes with shapes that have a number of nodal lines across the aerodynamic surfaces, each term in the homogeneous quadratic integrand of c_{ij} will have positive and negative intervals along the span which will tend to make the integral c_{ij} small. The integrand of the diagonal element c_{ii} is a quadratic expression in the "i"th generalized coordinate shapes and contains squared terms which are always positive. Therefore, the integral c_{ii} does not tend to be small. Accordingly, the nondiagonal elements in the row and column corresponding to a higher normal mode with a number of nodal lines tend to be small compared to the common diagonal elements. The zeros of the determinant then tend to be determined by the zeros of the minor determinant formed by striking out the rows and columns corresponding to the higher complex modes. If the complexity of a normal generalized coordinate be measured by the number of nodal lines crossing the aerodynamic surfaces, it may be stated from the above observations that the greater the complexity of the normal mode, the less likely it is to be important in flutter.

3. *Summary of Conclusions* – From the observations

[11] The term "normal coordinates" is used to denote the free vibration modes of a structure.

based on the flutter equation made in the preceding paragraph, the procedure in making a flutter analysis should be to take as generalized coordinates sets of successive normal modes, including the fundamental. Preliminary analyses should be made with a very few coordinates, greater numbers of coordinates being taken in succeeding analyses. If a flutter mode is obtained, the shape of which, considering its order of complexity, character of displacements, and so forth, should be favorably represented in terms of the generalized coordinates chosen, the resultant flutter speed, in all probability, is the lowest, and the flutter speed and mode accurately determined. When a flutter speed and mode have been obtained, studies with additional generalized coordinates can be made to determine definitely how many coordinates are necessary to represent the mode satisfactorily. If the flutter speed and mode remain unchanged when extra generalized coordinates are inserted into the analysis, these extra modes are not required in representing the flutter mode. If no flutter speeds are obtained with successive sets of normal generalized coordinates up to a fair degree of complexity, it becomes increasingly improbable that the structure has any critical flutter modes. This process may, of course, be carried as far as desired if there is sufficient interest or importance to warrant the carrying out of the extended numerical work for large numbers of generalized coordinates. It is probable that the effects described in paragraphs 1 and 2 would become evident on inspection of the numerical flutter equations and it might be "proved" in this way that a particular structure could have no flutter speeds.

IV. Determination of Normal Modes

1. Review of Problem – There is a considerable volume of engineering and scientific literature on the subject of normal modes and methods of obtaining them. Instead of reviewing all this material, the present section will describe a completely general method of obtaining normal modes to any desired degree of accuracy which can be used if no simpler method is available. The method described is not new but consists of a combination of two well-known methods in a way that is superior to either for general purposes and has been found suitable for engineering computations.

2. General Method of Obtaining Normal Modes – Any set of generalized coordinates of a structure may be made orthogonal by the use of Lagrange's equations as indicated in Section VI of Reference 1. The resulting orthogonal coordinates are approximations to the actual normal modes of the structure in so far as the latter are expressible in terms of the initially chosen generalized coordinates. If the general shape of the normal modes is known by previous experience or by analogy with similar analyzed structures, it is usually possible to choose satisfactory initial generalized coordinates. If, however, a specific knowledge concerning the shape of the normal modes is lacking, it is possible to improve upon the first set of generalized coordinates chosen in the following way:

The inertia forces corresponding to each of the orthogonalized initial generalized coordinates are placed on the structure and the resulting set of displacement forms is taken as a new set of generalized coordinates to be made orthogonal by Lagrange's equations. This process can be continued until each set of inertia forces results in the same displacement form from which it was obtained; the orthogonal forms of these displacement forms are true normal modes. In reality, this method of obtaining normal modes by successive approximations is an extension for higher modes of the usual successive approximations method, which always converges to the fundamental mode. The extension is obtained by introducing an orthogonalizing condition (solution to Lagrange's equations for orthogonal modes) at each step.

It is often convenient to use the successive approximation method described above to obtain normal modes of a structure with extra restraints of various kinds, and then use these modes as initial generalized coordinates for the structure without the extra restraints. For example, normal modes of a structure restrained first against bending, then against twisting, will usually serve very satisfactorily as final generalized coordinates on the structure without these extra restraints, that is, in its actual condition of restraint.

3. Conclusions – The use of the method described above for obtaining normal modes is possible where other simpler methods fail so that it is possible to find the normal modes of any structure whatsoever, provided its deflections under a specific loading can be obtained. The deflections of a structure can always be obtained by energy methods from stress analyses. By the considerations outlined in the present part and the methods described in Part II, the flutter analysis of any structure of the type considered is reduced to a series of well-defined computational steps. Besides complete mass distribution and geometrical data, these computational steps require only that a stress analysis of a structure can be made. Where a stress analysis of the structure can be represented in terms of the simple beam bending and torsional theories, stress analyses need not be made explicitly since the deflections can be expressed in terms of the loading and the "flexural" and "torsional" rigidity.

■ Acknowledgements

The author wishes to thank J. A. Skoog and M. J. Turner of the Chance Vought engineering staff for their assistance in the preparation of these papers. He also wishes to thank the NACA for the permission to reproduce Fig. 18 of NACA Technical Report No. 685.

■ ■ ■

Recent PROGRESS in SYNTHETIC RUBBERS

by P. M. TORRANCE

Manager of General Laboratory, The Firestone Tire & Rubber Co.

IT is the intention of this paper to give a review of the accomplishments in practical uses of synthetics in products of interest to the automotive industry, namely, tires and automotive mechanical goods. The creation of our synthetic rubber supply is a triumph of chemical research, chemical engineering, and industrial planning for which the industry and various government agencies concerned deserve great credit. However, it is a fair assumption that you are all reasonably familiar with this phase of the subject and I shall touch on it but briefly.

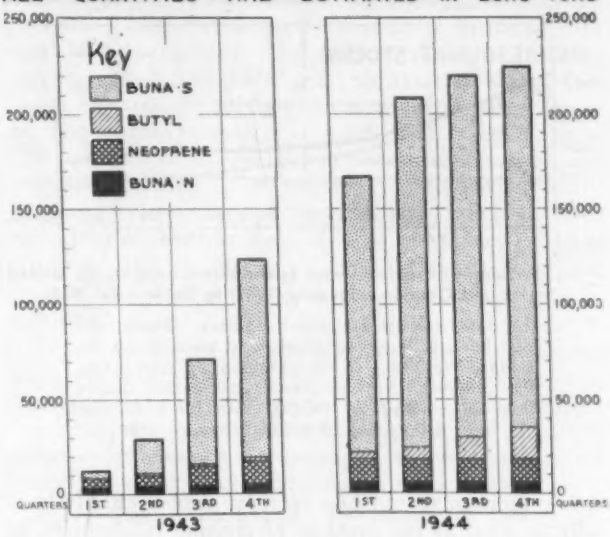
Fig. 1 shows the estimated quarterly production of synthetic rubber. It will be noted that by the second quarter of 1944 synthetic rubber will be produced at the annual rate of 800,000 long tons, which is equal to the consumption of natural rubber in the United States and Canada in the peak year 1941.

Fig. 2 sums up the entire rubber situation. Our combined stocks of synthetic and crude will reach a low point early in 1944. This makes it clear that rubber products must be converted from natural to synthetic as rapidly as it becomes available. The Office of Rubber Director is scheduling these conversions, taking first those products that are fully developed and in which the synthetics deliver the best relative performance. Thus, in the field of tires the smaller sizes are, in general, converted before the larger ones. At the present time, the rubber industry is over 50% converted and should reach at least 80% by the end of 1943.

[This paper was presented at the SAE Meeting on Materials – War and Post-War, Cleveland, Ohio, Nov. 10, 1943.]

	1943					1944				
	1ST	2ND	3RD	4TH	YEAR TOTAL	1ST	2ND	3RD	4TH	YEAR TOTAL
BUNA-S	3,102	18,792	56,741	105,000	183,635	145,000	185,000	190,000	190,000	710,000
BUTYL	35	393	364	1,500	2,292	4,000	6,000	12,000	17,000	39,000
NEOPRENE	4,372	5,853	10,049	12,300	32,574	12,300	12,300	12,300	12,300	49,200
BUNA-N	2,977	3,335	4,063	4,500	14,875	5,000	5,000	5,000	5,000	20,000
TOTAL SYNTHETIC	10,486	28,373	71,217	123,300	233,376	166,300	208,300	219,300	224,300	818,200

ALL · QUANTITIES · ARE · ESTIMATED · IN · LONG · TONS



■ Fig. 1 – 1943-1944 estimated quarterly production of synthetic rubber for buna S, butyl, neoprene, and buna N (includes neoprene and buna N capacities of private plants)

THE desire never again to risk the disaster of being cut off from our main source of rubber and the development of "tailor-made" synthetic elastic materials with special characteristics to take care of particular applications much better than natural rubber – these two reasons, Mr. Torrance feels, will assure that after the war synthetic materials will continue to be used in large quantities and will take their rightful place alongside of natural rubber in our economy.

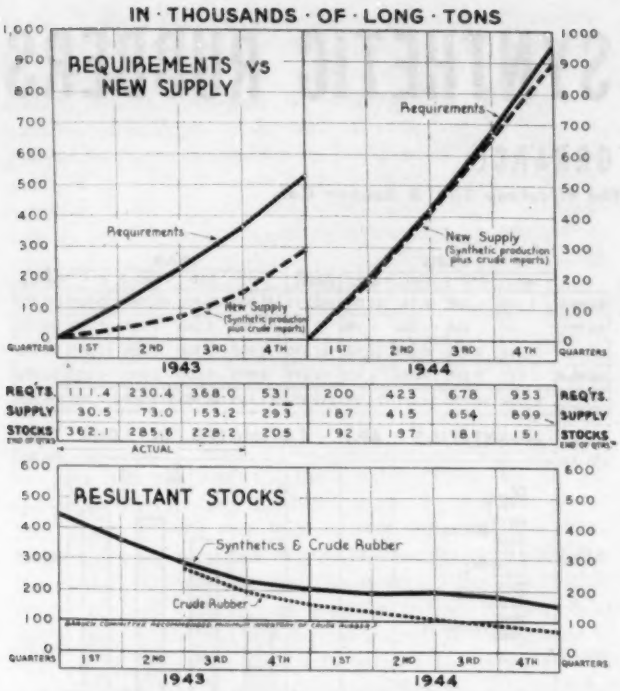
Even before the war, use of some of these special-purpose synthetics had been well established. The oil resistance of some synthetics has made them much more satisfactory than natural rubber for many applications where oil is encountered, such as in water pump seals, radiator hose, and fan belts.

Continued development of synthetics has led to synthetic tires that are satisfactory when used at reasonable speeds and normal loads on passenger cars and light trucks. Use of rayon and even nylon cord in place of cotton results in greatly improved performance because of the reduction in running temperatures.

GR-S inner tubes are reasonably satisfactory, except for the smaller sizes used on drop-center rims, where the high well stretch causes many premature failures. Good progress has been made in synthetic bogie rollers and rubber tank tracks, for the properties of synthetics appear to be very satisfactory in such solid-tire applications.

■ ■ ■

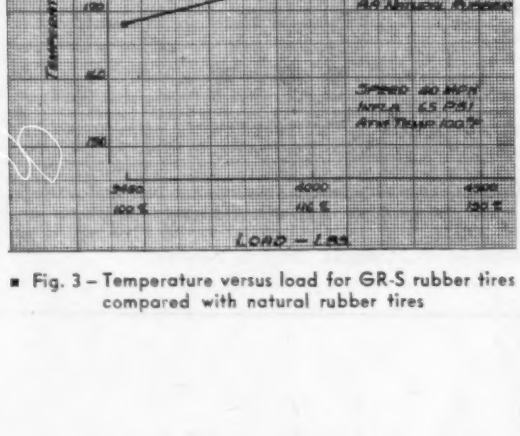
THE AUTHOR: P. M. TORRANCE, manager of the general laboratory of Firestone Tire & Rubber Co., has been with that company for the past 13 years, concentrating on the development of rubber compounds for tires and related products. A graduate from the University of Illinois in 1926 with a B. S. degree in chemistry, Mr. Torrance is a member of the American Chemical Society.


IN · THOUSANDS · OF · LONG · TONS
REQUIREMENTS VS NEW SUPPLY
Requirements
New Supply (Synthetic production plus crude imports)
QUARTERS 1ST 2ND 3RD 4TH 1ST 2ND 3RD 4TH
1943
1944
REQ'TS. 111.4 230.4 368.0 531 200 423 678 953 REQ'TS.
SUPPLY 30.5 73.0 153.2 293 187 415 654 899 SUPPLY
STOCKS 362.1 285.6 228.2 205 192 197 181 151 STOCKS
ACTUAL
RESULTANT STOCKS
Synthetics & Crude Rubber
Crude Rubber

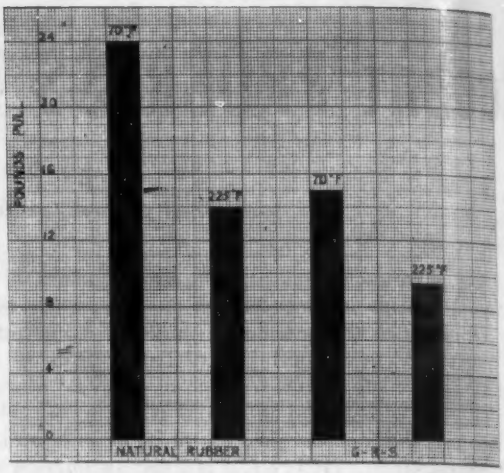
■ Fig. 2 – Supply of rubber versus essential requirements for United States and Canada – January, 1943, to December, 1944

All data are on a cumulative basis. Crude rubber, buna S, butyl, neoprene, and buna N are included. Resultant stocks obtained by subtracting the difference between requirements and supply from initial stock of 443,000 tons for 1943 and from initial stock of 205,000 tons for 1944

The problem of utilizing synthetic rubber effectively is fully as great as the problem of creating it. It must be properly compounded for each part of every product, which must also be properly designed. For 104 years, ever since Charles Goodyear discovered vulcanization, we have



■ Fig. 3 – Temperature versus load for GR-S rubber tires compared with natural rubber tires



■ Fig. 4 – Effect of temperature on tear resistance of tread compounds

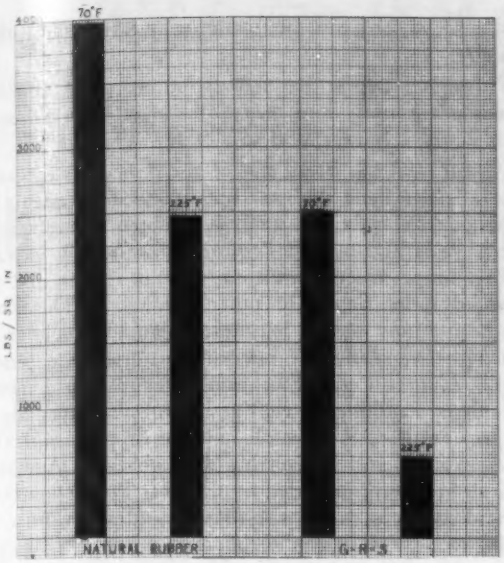
gradually developed the natural rubber compounds of today, which protect the weaknesses of the material and take full advantage of the strong points. In like manner, we have gradually worked out the best designs and construction for each product to compensate these same weaknesses and to utilize fully the strong features.

You are all familiar with the fact that the name "synthetic rubber" is misleading to the point of being a misnomer. The synthetic rubbers now available are really synthetic elastic materials somewhat similar to natural rubber in many physical and chemical properties, but differing in many important respects. It is hardly necessary to point out that the full utilization of synthetic rubber requires a similar cycle of development, not only of compounds but also of designs and constructions.

It has been approximately three years since the synthetic rubber most suitable for tire manufacture, namely, GR-S or buna S, became available in sufficient quantities for experimental work on a factory scale. The major rubber companies had made and tested experimental tires previously, but the fall of 1940 is the time they rolled up their sleeves and began such experimental work in a big way. At this time, we were all preparing to use synthetic rubbers in large quantities to supplement and extend our natural rubber supply, but it was not until after Pearl Harbor and the subsequent fall of Singapore and the Dutch East Indies that anyone realized we were faced with practically complete replacement of natural rubber with synthetics. The problem then assumed astronomical proportions.

It was realized that complete cooperative effort by all was necessary in order to make many years' normal progress within the year and a half that remained before our natural rubber stockpile would be exhausted at normal rates of consumption. The major rubber companies had long before pooled all resources in creating the synthetic rubber supply and, in February, 1942, they organized a committee on compounding and use of synthetics under the auspices of the Rubber Reserve Co. Great progress was made due to the whole-hearted cooperation of an industry that had always been intensively competitive.

In June, 1942, the Army Ordnance Department began large-scale testing of synthetic tires at Camp Normoyle, Tex., and Camp Seeley, Calif. These constitute the largest tire testing operations in history. In the Camp Normoyle



■ Fig. 5 – Effect of temperature on tensile strength of tread compounds

operations alone more than 500 people are employed and more than 70 test vehicles are in active service. These operations have been extremely important in the development of reasonably satisfactory synthetic tires. The Army deserves great credit for this contribution to synthetic rubber development.

In August, 1942, the SAE Ordnance Advisory Committee expanded its Rubber Products Subcommittee into several subgroups, one of which is the Pneumatic Tire Subcommittee. Through it, the various rubber companies and the Ordnance Department have cooperated to exchange information on synthetic tires and plan tests for these Army operations. Performance of synthetic Army truck tires, which was unsatisfactory at the start, has been improved to give reasonably satisfactory service at the present time.

In December, 1942, a committee of chemists and engineers, known as the Synthetic Tire Construction Committee, was formed by the tire companies and the Office of Rubber Director to correlate all development of synthetic tires for both civilian and military use. Since May, 1943, this group has operated a fleet of 26 large commercial trucks at Phoenix, Ariz., testing 9.00-20 tires from the 16 principal tire manufacturers. There is complete exchange of information and splendid cooperation so that rapid progress is being made in the development of civilian truck tires.

In May, 1943, a second large fleet of trucks, known as the Government Test Fleet, was also started testing civilian highway truck tires. This is financed by the Rubber Reserve Co. and is operated for the Office of the Rubber Director by the Copolymer Corp., which in turn is owned by a group of smaller manufacturers. This fleet runs tests on products of all manufacturers as well as special tests on matters of interest to the Office of the Rubber Director, such as independent evaluation of rayon and cotton tire cord fabrics.

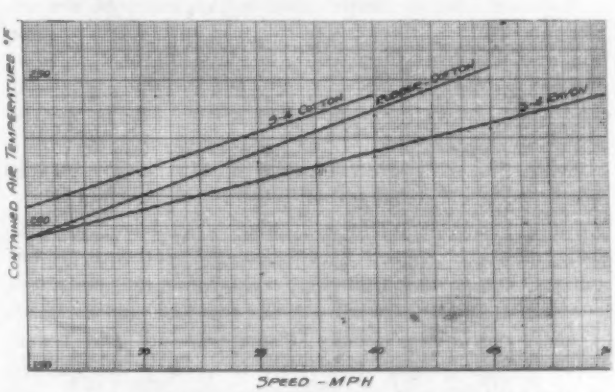
The synthetic passenger-car tire is today a satisfactory product. At reasonable speeds it will deliver performance almost equal to the best pre-war tire. The small civilian truck tires are reasonably satisfactory for use on light trucks, although their relative performance drops rapidly with extreme overloads. The great majority of Army truck tires fall in this size group and may be expected to give reasonable performance, because generally speaking they are not overloaded. Tread cracking, which is a most serious problem in highway tires, is not serious in Army tires because of the nondirectional cross-bar design, which minimizes this type of failure. The larger truck tires, 8.25 section and up, are still a most serious problem and must be considered in some detail.

Fig. 3 shows the operating temperature of 9.00-20, 70% synthetic truck tires compared with natural rubber at several truck-loading conditions. It will be noted that the temperature difference is large and becomes greater with increasing overload. Also, the temperature of the synthetic tires at rated load is approximately equal to the natural rubber tire at 30% overload. This excess heat causes premature fabric breaks and blowouts of synthetic tires and also aggravates the other great weakness of synthetic rubber, namely, its lower tear resistance, which causes excess cracking or splitting when a cut or other injury breaks the tread surface. Fig. 4 shows the deterioration in tear resistance of natural and synthetic rubber at elevated temperatures. Fig. 5 shows the deterioration in tensile strength at elevated temperatures. A large number of tests have shown that the total mileage expectancy of these synthetic tires is increased threefold when loads are dropped to Tire and Rim Association standard from 30% overload.

Fig. 6 shows the operating temperature of a 7.50-20, 90% synthetic Army tire made with cotton compared to the same tire made with rayon, and also a natural rubber tire made with cotton. It will be noted that the rayon reduces the running temperature of the synthetic tire making it equal to or lower than natural rubber.

Fig. 7 shows the operating temperature of 9.00-20 rubber cotton tires compared with rayon. It will be noted that here again rayon reduces temperatures and greatly increases mileage at failure.

Tests to date have shown that large synthetic truck tires will give reasonably satisfactory service, if made of high-tenacity rayon and if extremes of overload and speed are avoided. Sufficient high-tenacity rayon has been made available for large Army tires and for intercity bus tires. There is further expansion in progress, but there will not be sufficient high-tenacity rayon available for large civilian truck tires for some time to come. As a result, present plans call for the use of much more natural rubber in these



■ Fig. 6 – Comparison of operating temperatures – 7.50-20 Army tire, 8-ply

tires made with cotton fabric than would otherwise be required. While there is sufficient natural rubber available to use considerable quantities in these large tires, it will shortly become necessary to use appreciable proportions of synthetic in them before rayon becomes available.

We are all familiar with the fact that extreme overloads and high speeds have been common, particularly in present-day intercity trucking operations. Overloads and high speeds must be eliminated in order to operate successfully on synthetic tires. Just how this can best be accomplished without crippling the trucking industry is a problem receiving a great deal of attention from the rubber industry and the government agencies concerned. In the meantime, we are doing everything possible to improve synthetic tires.

Since the truck manufacturers are also concerned with this problem in relation to truck maintenance, it is probably desirable for them to include the synthetic tire story in their educational programs for their field organizations. It is hardly necessary to point out that the need for reduced loads should be kept in mind in designing any new equipment utilizing synthetic rubber tires.

GR-S inner tubes are reasonably satisfactory, except for the smaller sizes used on drop-center rims, where the high well stretch causes many premature failures. There are indications that tubes for drop-center rims may be improved through the use of other synthetics, particularly butyl rubber. The largest sizes have not yet been adequately tested for evaluation.

Good progress has been made in synthetic bogie rollers and rubber tank tracks. The properties of synthetic appear to lend themselves better to replacing rubber in such solid-tire applications. The Army has done extensive testing, and reasonable performance relative to natural rubber can be expected. Half-tracks have not yet been tested sufficiently for evaluation, but it is probable that a reasonably satisfactory product can be made.

■ Automotive Mechanical Goods

In the field of automotive mechanical goods, we have a much brighter picture. Before the war, the special-purpose synthetics had established themselves in many uses due to their superior performance and in spite of their higher cost. You are all familiar with the oil-resistant neoprene water pump seal, which was an almost miraculous improvement over the old mechanical packings. Such seals may also be made of buna N when it is available in sufficient quantities.

Radiator hose is much better when made of the oil-resistant synthetics, especially in these days of rust inhibitors and closed pressure cooling systems. Oil-resistant brake hose may make improved types of brake fluids practical. Hot-water heater hose and many other parts present no problem.

Synthetics show great promise in fan belts, especially neoprene, where its oil resistance is of some advantage.

Motor mounts present a number of problems, particularly due to the greater hardness of the synthetics. It has not yet been possible to make the extremely soft 30-durometer stocks, and this may necessitate redesign of some types.

Seals, grommets, gaskets, insulators, and dust covers create no serious problems and, in most cases, the oil-resistant synthetics give superior performance.

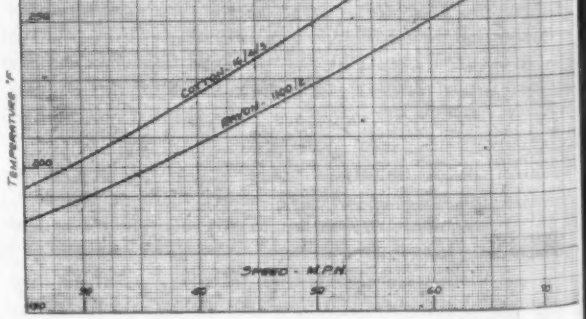
Channel rubbers for windshields, windows, door edgings, and so on, can be made from synthetic without difficulty.

Windshield wipers are actually improved because they do not get gummy and tacky due to oil splashed on by passing cars, oily rags, and so forth.

A great deal has been said of the difficulties of factory processing synthetic rubbers. This is especially true of the special-purpose rubbers, such as neoprene and buna N. Several years ago buna S required considerably increased milling times as compared to natural rubber and also processed more slowly in other operations. This would have been a most serious handicap because it would greatly reduce the capacities of our tire factories. Cooperative effort has produced softer buna S and also improved methods of handling as well as improved compounds. At the present time, the overall average drop in capacity is not over 20% and is probably somewhat less. Additional facilities are being provided by the War Production Board to compensate for this loss so that our rubber factories will be able to process all of the rubber required.

In closing, I would like to look into the future and predict that chemical research will produce much better synthetics. It is even probable that they will surpass natural rubber in other properties as they have already surpassed natural rubber in oil resistance. There is no good reason to believe that Mother Nature, in creating the natural latex of the rubber tree, designed the best possible material for tread wear, heat generation, and other desirable characteristics. There are numerous well-known samples of man-made synthetic products created for specific purposes which have far surpassed the natural materials previously used. In our own tire industry, we have high-tenacity rayon fibers, which give vastly superior results to natural cotton fibers. This was not accomplished overnight, but was the result of many years of painstaking research and development. At the present time, nylon fibers give every indication of doing a still better job. Given sufficient time, there is every reason to believe that synthetic rubbers will be created vastly superior to the present materials.

After the war it is entirely probable that we will resume the use of natural rubber for at least part of our requirements. It is imperative that the rubber industry make every effort to develop superior synthetic rubbers, which can be produced economically, so that we will never again risk the disaster of being cut off from our sources of such a vital material as rubber.



■ Fig. 7 – Comparison of operating temperatures – 9.00-20 tire, 10-ply (rubber)

A New Laboratory Method of Evaluating Ring-Sticking Tendencies of Aircraft Oils

by L. W. GRIFFITH, V. E. YUST, and M. E. SCHRAMM
Shell Oil Co.

PRESENT-DAY airplanes, with their tremendous weight and high speeds, require engines of correspondingly great horsepower. The attainment of such power involves the use of high bearing loads, reciprocating forces, and temperatures which in turn frequently introduce lubricating problems including excessive wear and scuffing, excessive oxidation of the oil, and ring sticking.

While none of these lubricating problems can be considered the outstanding one, because differences in engines and other variables determine which problem will be encountered first, nevertheless each must be considered chronic and worthy of special study.

In this discussion we have chosen the one problem of ring sticking, and from all that could be said on that one subject, we shall restrict ourselves to the testing of lubricating oils to compare their abilities to prevent ring sticking. Furthermore, of several methods that could be discussed for measuring this factor, we will discuss only one method which we have developed on the CFR engine in which oils are compared under constant conditions on the basis of the time required for one ring to stick.

[This paper was presented at the SAE National Fuels and Lubricants Meeting, Tulsa, Okla., Nov. 5, 1943.]

CONTINUOUS increase in the power output of aircraft engines introduces from time to time lubricating problems including excessive wear and scuffing, excessive oxidation of the oil, and ring sticking. The one problem of ring sticking was chosen and the discussion is limited to the testing of lubricating oils to compare their abilities to prevent this type of failure. Although the best answer as to the ring-sticking tendencies of a lubricant rests with the full-scale engine in service, a simple test is needed during the development period.

The development work which led up to the selection of an L-head CFR engine for a ring-sticking test is discussed. Various criteria used for detecting incipient ring sticking are mentioned and a method for direct measurement of incipient ring sticking is described. This direct method utilizes an opening cut in the cylinder wall so that a tool can be inserted into a notch in a pinned ring to determine the tangential force required to move the ring at any time during the test. The tangential force required to move the ring was found to agree well with the degree of ring sticking.

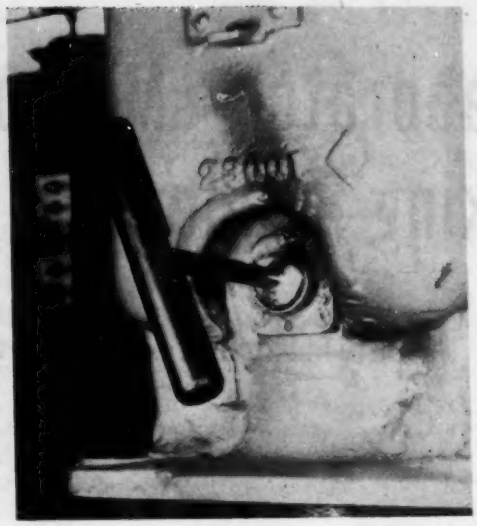
The engine operating conditions necessary to give ring sticking in a reasonable length of time on an average oil are enumerated.

Data are also given concerning the effect of air-fuel ratio, spark timing, and jacket and oil temperatures on piston-ring-land temperatures. Ring-sticking ratings on a number of oils are given and satisfactory separation and relative performance of the lubricants are evident.

■ ■ ■

THE AUTHORS: L. W. GRIFFITH (J '39) is senior engineer in charge of engineering research on lubricants at Shell Oil Co., Inc. He previously was connected with the Ex-Cell-O Corp., after having received his degree in mechanical engineering from the University of Wisconsin in 1937. V. E. YUST (J '43) is senior research engineer in charge of engineering fuels research at Shell Oil Co., Inc. During the development of the subject test method, he was head of engineering projects on aircraft lubricants research. Graduating from Bethel College with an A. B. degree in chemistry in 1935, he took post-graduate courses at Kansas University, and was later associated with Derby Oil Co. Mr. Yust has been active in CRC projects in recent years. M. E. SCHRAMM (M '35), who joined Shell Oil Co. in 1933 as motor testing engineer, has been automotive research engineer at the company's Wood River Research Laboratories for the past three years. He graduated from Armour Institute of Technology in 1932, where he studied mechanical engineering.

■ Fig. 1 (left) – Opening in cylinder wall of CR L-head engine, which exposed the lower half of the top ring when the piston was on bottom dead center



■ Fig. 2 (right) – Cylinder window of Fig. 1 plugged

■ Requirements of a Ring-Sticking Test

The best answer to the ring-sticking tendencies of a lubricant rests with the full-scale engine in service. A valuable substitute is a full-scale engine on a test stand which, however, is usually used only to obtain an answer preliminary to the actual service tests. However, although these methods must be used for the final evaluation of an oil, they are not at all suitable for development work in which a large number of samples must be tested.

A laboratory ring-sticking test, to be useful for development work preceding the submission of the lubricant to full-scale tests, must have the following characteristics:

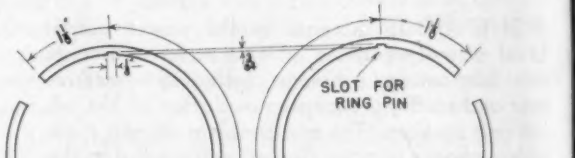
1. It must be inexpensive and rapid to run and (in wartime) use a minimum of critical materials.
2. It must be sufficiently sensitive to differentiate between materials of significantly different performance and the results must be reproducible.
3. The ratings obtained must be significant in terms of full-scale performance.

■ Methods of Detecting Ring Sticking

The development work which led up to the test we are describing covered a period of several years and was primarily a study of ring sticking in various types of engines using a wide variety of test methods. During the course of this work it soon became apparent that the most important single factor in any ring-sticking test procedure was the ability to detect when ring sticking had progressed to the same degree on all tests. It was necessary, therefore, to study the various criteria used for detecting incipient ring sticking in order to find a satisfactory method. The most widely used of these were as follows:

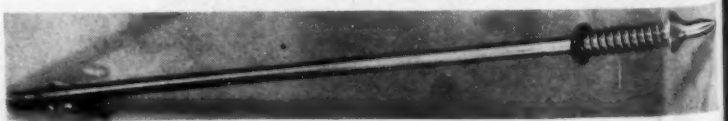
1. Loss in power output.
2. Increase in blowby.
3. Increase in oil consumption.
4. Increase in the cylinder-liner (or piston) temperature.
5. Disassembly of engine.

All of the above methods have been used, many times with good results, by investigators working on the problem, but we believe that each has some fundamental weakness. The first four – power loss, blowby, oil consumption, and liner temperature – are indirect methods and as such, because of the infinite number of ways in which a ring can stick, permit a wide variation in the amount of ring sticking for the same indication. They all depend on the failure of the ring to seal the moment it begins to stick. In an engine maintained with the close tolerance necessary for a reproducible ring-sticking test this is not always true. It has been our experience that the failure of the ring to seal does not necessarily correspond to a definite amount of ring sticking. Dismantling the engine, although it is a direct and positive method of determining ring sticking, is to be avoided if possible because of the excessive amount of time required. Also it can never be determined whether the rings were stuck under operating temperatures.

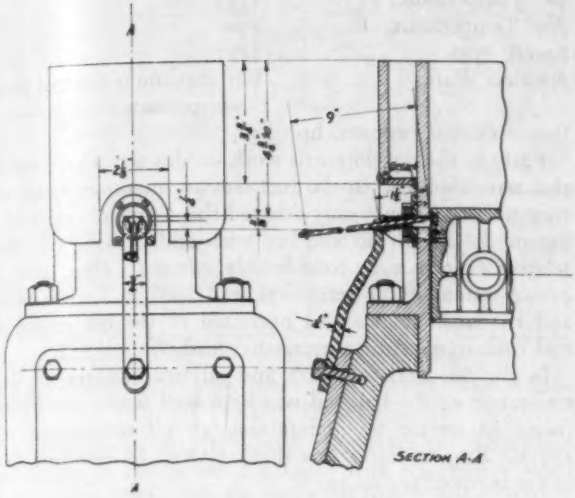


■ Fig. 3 – Compression-ring alterations: left, second compression ring; right, first compression ring

To eliminate the undesirable features of the methods discussed above, a method was developed which permitted the actual tangential force required to move the ring in the groove to be measured at momentary stops periodically



■ Fig. 4 – Ring rotating tool

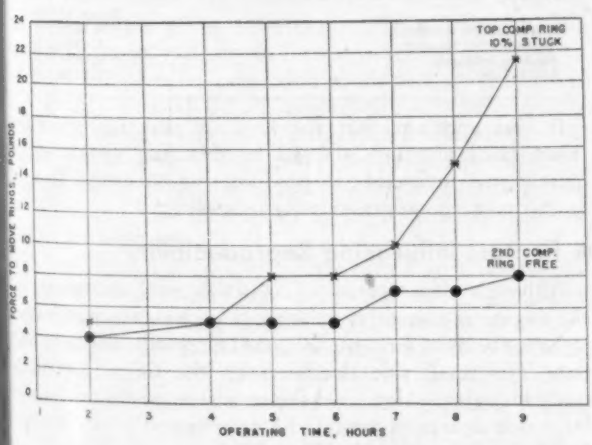


■ Fig. 5 – Modified CFR cylinder alterations and accessories

during a test. The time at which a ring began to stick and the rate at which sticking progressed was readily apparent.

The application of this method to a CFR L-head engine was accomplished by cutting an opening in the cylinder wall which exposed the lower half of the top ring when the piston was on bottom dead center (see Figs. 1 and 2). A small notch was cut in the bottom half of the first and second rings (see Fig. 3) and the rings pinned to keep the notch in line with the opening in the cylinder wall. By means of a pointed steel lever (Fig. 4) pivoted outside the engine and engaging the notch in the ring, the force required to rotate the ring in the groove within the limits allowed by the pin can be measured (Fig. 5). A spring scale is used to determine quantitatively the force required (Fig. 6).

During the operation of the engine under conditions which eventually produced ring sticking, the force necessary to rotate the ring in the groove gradually increased until movement was no longer possible. Repeated tests showed that if the test was terminated the moment the ring failed to move under a tangential force of 20 lb,



■ Fig. 7 – Force to move rings plotted against operating time for representative test (using special rotating tool)

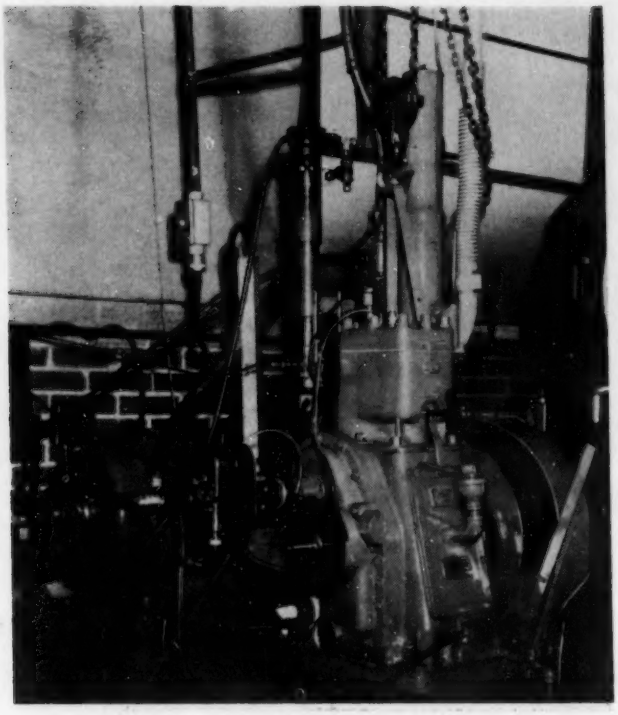
■ Fig. 6 – Measuring force required to move ring with ring rotating tool

approximately 10% of the circumference of the ring would be stuck in the groove when the piston was removed. A force of 20 lb was therefore arbitrarily chosen as the "ring sticking point." The variation of the force required to move the top ring with running time is shown in Fig. 7. A distinct advantage of this method is that the engine can be stopped at any time during the test and the extent of ring sticking determined without significantly disturbing the deposits in the ring grooves or the temperature equilibrium in the engine. The engine can then be restarted after the 15- or 20-sec stop necessary for this measurement to be made and the test continued.

■ Ring-Sticking Test

With the development of a method of detecting ring sticking, the ring-sticking test is only partially complete, for conditions must be established which will produce significant results. Since any test not run in a full-scale aircraft engine under service conditions is not specific, but must depend on correlation for the validity of the results, the choice of an engine depends on such factors as availability of parts, ease of overhaul, and, most important of all, the ability of the engine to maintain fixed mechanical clearances and alignment over long periods of testing.

The CFR L-head engine was chosen for the work because, with some modification, it satisfied most of these requirements and was easily adapted to permit control of jacket, oil, and intake-air temperatures. This engine as set up for this test is shown in Fig. 8. In addition to the changes described earlier in this report, the only major modification in the engine was the use of a Plymouth cast-iron piston and the fabrication of a connecting rod of the proper length and fitted with a removable orifice in the oil passage at the upper end. Changing the size of this orifice, which controlled the rate of cooling oil flow to the



■ Fig. 8 – Revised CFR engine equipped for ring-sticking testing

piston, permitted some control of the piston temperature independently of the jacket temperature and engine load. Automatic devices were used to control temperatures and a synchronous motor was used for power absorption to ensure constant speed. Complete details of the other changes made in the standard engine are given in the Appendix.

With the engine set up as described, the severity of the test was adjusted by varying the jacket and piston temperatures until ring sticking would occur in a reasonably short length of time on present-day commercial aircraft oils of known satisfactory performance in this respect. The performance of unknown materials could then be judged by determining the length of time necessary for them to stick the rings to the same degree as the reference oils.

The conditions chosen were briefly as follows, while the complete specifications are given in Table 1.

Table 1 – Test Conditions for Ring-Sticking Test for Aircraft Oils (Modified CFR Engine)

Speed, rpm	1200
Horsepower (approximate, not used for control)	4.5
Spark Advance, degrees BTDC	17
Mixture Strength	That required for maximum thermal plug temperature
Coolant	Ethylene glycol + water
Coolant Temperature, F	300
Inlet Air Temperature, F	150
Crankcase Oil Temperature, F (at lowest point of gear case)	175
Lubricating Oil Charge, ml	2500
Compression Ratio (fixed cylinder head)	5.00:1
Oil Pressure, psi (constant)	40 ± 5
Cylinder Bore, in.	3.2525
Piston-Skirt Clearance, in.	0.0025 ± 0.0003
Compression-Ring Side Clearance, in.	0.0035 ± 0.0002
Compression-Ring Gap Clearance, in.	0.020 ± 0.001
Oil Control Ring Side Clearance, in.	0.0020 ± 0.0002
Oil Control Ring Gap Clearance, in.	0.020 ± 0.001
Wristpin Bushing Clearance, in.	0.0002
Fuel – Aviation straight-run type containing 1.00 ml tel per gal of antiknock quality to ensure knockfree operation.	
Valve Clearance	
Intake, in.	0.008
Exhaust, in.	0.010
Jacket Temperature, F	300
Oil Temperature, F	175
Air Temperature, F	150
Speed, rpm	1200
Air-Fuel Ratio	For maximum thermal plug temperature
Power Output, approx. hp[1]	4½

Early in the development work on this test it was found that tetraethyl lead in the fuel was an important factor in ring sticking. It not only affected the degree of ring sticking or the length of test, but with additive-type oils, the relative ratings were considerably affected. Thus, after a consideration of the tetraethyl lead used in aircraft fuels and the maintenance and operation of the test engine a fuel containing 1 ml of tetraethyl lead was selected.

In the selection of jacket and oil temperatures the deterioration of the bulk oil was kept well within the limits found in service by maintaining an oil temperature of 175 F. Ring sticking was then induced by elevating the jacket temperature to 300 F.

Under these conditions the average commercial straight mineral aircraft oil produced ring sticking in about 8-10 hr. One oil was chosen as a reference and the ratings of all other oils expressed as a percentage of the ring-sticking time on this oil. To eliminate the effect of gradual changes in the severity of the test, all unknown samples are bracketed by runs on the reference oil.

The sensitivity of this test is illustrated by the ratings obtained on a series of oils of known performance, as shown in Table 2.

Table 2 – Ring-Sticking Ratings

Oil	Relative Ring-Sticking Rating, % of reference oil	Service Performance
Reference Oil	100	Average
Commercial Oil A	130	Average
Commercial Oil B	80	Average
SAE Group IV Oil No. 17	60	Poor
SAE Group IV Oil No. 18	80	Fair
SAE Group IV Oil No. 18	180	Good
Experimental Oil D	210	....
Experimental Oil E	280	....

Although the service ratings on the oils were very indefinite, it appeared that a satisfactory separation was obtained in this test. No service ratings were available on the experimental oils, but the additives used were materials with good performance records in other applications.

Typical properties of the reference oil after a test are shown in Table 3.

Table 3 – Properties of Reference Oil After Ring-Sticking Test

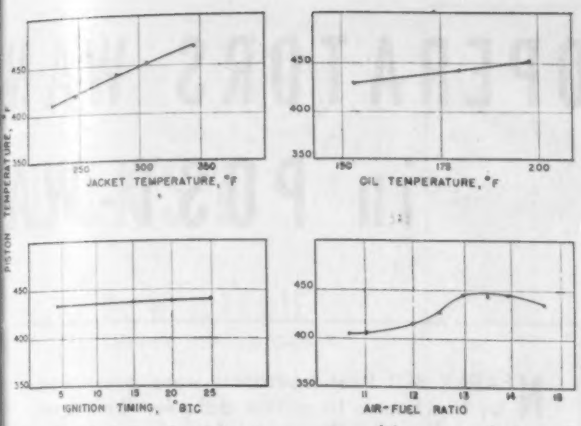
	Range of Values
Neutralization Number	0.05 to 0.4
Saponification Number	0.84 to 5.0
Naphtha Insoluble	0.03 to 0.04
$CHCl_3$ Soluble	<0.01 to 0.02

It was apparent that the bulk oil was not seriously deteriorated although the oil in the ring grooves had deteriorated sufficiently to produce ring sticking. This is in line with conditions existing in service.

■ Factors Influencing Reproducibility

Although close control of all mechanical clearances in the engine is necessary, the ring side clearance was found to have the most serious effect on the reproducibility of the test. Too small side clearances on the rings (less than 0.002 in.) greatly decreased the sensitivity of the test, while large side clearances produced long tests and erratic results. It has been found desirable to hold the side clearance of

[1] Not used for control.



■ Fig. 9 – Variation in piston temperature (thermocouple in antithrust side of top land)

the compression rings to 0.0035 ± 0.0002 in. The rings are lapped to obtain this clearance with fine emery cloth on a surface plate with the rings held in a special fixture to ensure uniformity. The rings are polished on fine crocus cloth to remove the "fuzz" and prevent rapid increase in clearance early in the test.

It is realized that the two notches put in the ring, as shown in Fig. 3, may affect the ring tension, therefore these notches are cut with great care so that all rings will be similar. Any slight impairment in ring tension caused by the notches, if held constant, should have no significant effect on an oil rating which is obtained by comparison with a reference oil.

Engine operating variables are, of course, very closely controlled. Of these variables, those affecting piston temperatures are probably the most important. In this connection, some interesting data were obtained on another L-head CFR engine fitted with thermocouples in the piston and a special intermittent contact device[2] for the thermocouple leads. When the engine was operated under conditions comparable with those of the ring-sticking engine except that the speed was necessarily kept at 900 rpm, the following temperatures were measured:

Thermal Plug, in cylinder head, F	730
Piston Crown, center, F	500
Top Land, antithrust side, F	445
Second Land, antithrust side, F	380
Second Land, thrust side, F	375

By varying the spark timing, air-fuel ratio, and oil and jacket temperatures the relative effects of these variables on piston temperatures were found. Fig. 9 shows that the temperatures in the ring-belt area were very sensitive to changes in jacket temperature and air-fuel ratio, but was not greatly affected by oil temperature or spark timing. The relation between jacket temperature and piston temperature is interesting since it was determined in the development of the ring-sticking test that the ring-sticking time on a straight mineral oil was an exponential function of the jacket temperature.

It can be seen from the above results that it is essential, if reproducible results are to be obtained, to hold the variations in air-fuel ratio and jacket temperature within very narrow limits. Oil temperature did not markedly affect piston temperatures in this engine, probably because the effect of hotter oil was compensated by greater cooling from the increased flow of oil to the piston. However, variation in oil temperatures can not be permitted because of changes in bulk oil deterioration and the influence of this factor on ring sticking.

[2] The method used was similar to that developed by General Motors Research Laboratories. See SAE Transactions, Vol. 48, January, 1941, pp. 20-27: "An Instrument for Continuous Measurement of Piston Temperatures," by Arthur F. Underwood and A. A. Catlin.

■ Interpretation of Results

Although a test of this type can be used only in early development work, care must be used in the interpretation of results to avoid erroneous conclusions. Ratings obtained from this test as it is described above represent only one point on the temperature-ring sticking curve, and ratings obtained at another temperature may conceivably show somewhat different trends, particularly with additive-type oils. It is therefore essential that, for results which will be valid for a wide range of service conditions, a complete temperature ring-sticking curve be obtained.

Appendix

Engine Modifications for Ring-Sticking Test for Aircraft Oils

The engine comprises a standard ASTM-CFR low-speed crankcase and a CFR fixed-compression-ratio cylinder, augmented by the following changes and accessories:

Cylinder – Fixed compression ratio, valve-in-side L-head, fitted with a special window to accommodate the use of a special ring rotating tool. See Figs. 5 and 8.

Cylinder head – 5.00:1 compression ratio.

Spring scale – 0-4 lb (Fig. 6).

Valves – Plain uncooled inlet valve of ⅜-in. stem and sodium-cooled exhaust valve of 7/16-in. stem used.

Cooling system – Evaporative type.

Piston – Part 946252-C Piston Assembly, 986466 Standard 1942 Plymouth, cast iron.

Piston rings – 2 compression 3.25 x 3/32 in. (uncoated). Altered as per Fig. 3. – 2 oil control 3.25 x 5/32 in. (uncoated).

Connecting rod – Top bushing channeled and bored. – Top of rod drilled for piston cooling oil jet.

Oil heater – Crankcase heating element controlled by immersion-type thermostat located in crankcase drain opening.

Intake air heater – CFR diesel type intake air heater and silencer located on adapter placed before carburetor and controlled by thermostat located in adapter.

Carburetor – Zenith No. TU4 with needle valve for metering jet adjustment.

Carburetor insulating block – ¾-in. thick fiber block placed between carburetor and engine block.

Jacket temperature thermometer – Located in coolant return line.

Oil temperature thermometer – Located in lowest part of gear case.

Flywheel hand brake – Approximately 12 sq in. braking area applied by conveniently located handle (for quickly stopping engine).

Ignition – Magneto driven by camshaft.

Spark plug – Champion RJ-11.

Power absorption and speed control – Standard ASTM-CFR 4-hp synchronous motor belt connected with proper pulley ratio to give specified engine speed.

Combustion chamber temperature obtained with an F-3 thermal plug and potentiometer.

Camshaft – Low-speed low-lift (Waukesha Part No. 23123F).

WHAT DO FLEET OPERATORS WAN
in POST-WA

MOST operators have their own definite interpretation of what constitutes the desirable and undesirable feature of a truck as it applies to their operation. However, it appeared that, to convey a message back to the manufacturer as representative of the thoughts of operators as a group, it would be highly desirable to canvass operators throughout this country to obtain their views on this subject. Towards this end, therefore, three different types of questionnaires were sent out to a total of 290 addresses, represented as follows:

25 to private-carrier operators (special)
81 to private-carrier operators
184 to common, contract, and local cartage groups

The group marked "special" received an individual letter indicating the general outline of the problem and they were asked to give their views on "What do fleet operators want in post-war trucks?" without requiring them to answer any set questions. The response here was practically 100% and the information submitted quite complete.

The other groups listed were sent a typical questionnaire requiring, in most instances, a "yes" or "no" type of answer with the provision to discuss any or all the questions if they so desired. The response here was equally heartening, and the data gathered were used essentially to act as confirmation to the discussion which will follow.

I wish to acknowledge and thank all that have taken the time to submit their valuable opinions on this subject and to assure them that the information submitted was used freely throughout the paper.

■ Cooling System

The cooling system is without question a very important item of the automotive equipment for obtaining proper engine operation and efficiency. The lack of proper functioning of the cooling system results in increased maintenance effort, varying from poor lubrication, sludge deposition, and poor economy at low-temperature levels, to actual destruction of component metal structures at excessively high temperatures. This information is equally well known by the manufacturer as well as the user; and yet, in spite of this, there is generally prevalent an apparent negligence of the cooling system. In speaking of negligence, the term is not only applicable to the operator but also involves to a considerable degree the manufacturer of the vehicle as a unit.

When the manufacturer of the engine incorporates all the items necessary to keep temperatures at some predetermined figure for the best operation of his unit, he has combined in the design, laboratory and road tests to arrive at the desired goal. Many operators may question certain items or features of any particular unit, but, as a

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NEARLY 300 fleet operators were questioned by Mr. Laurie to gather data for this paper on what features the users of trucks would like to have the manufacturers incorporate in post-war trucks.

The cooling system is one important item that came in for its share of criticism. For example, maintaining the proper coolant level is most important, and yet many of today's systems require filling into the filler neck before the liquid level can be seen. Petcocks or sight gages properly installed could solve this problem, according to Mr. Laurie.

Accessibility for maintenance should also be improved in the post-war truck. Some of today's trucks have batteries that are not located for ease of servicing and spark plugs that it is almost impossible to remove and replace.

Improvements should also be possible in cold starting of the engine, based on the experience of the Army in cold climates.

Truck cabs could be much improved, so that they could accommodate the long-legged and the short-legged driver equally comfortably and provide satisfactory vision for both.

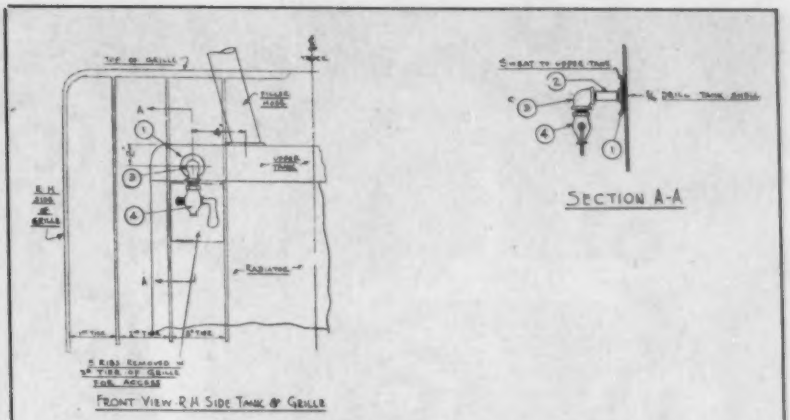
Another item on which the operators agreed was that they weren't interested in fancy hardware or grille work, such as is found on passenger cars. More important to them are sturdy construction and well-designed safety features.

■

THE AUTHOR: GAVIN W. LAURIE (M '35), manager of automotive transportation, Atlantic Refining Co., has been with that company since 1926, during which time he has been in charge, successively, of construction equipment, maintenance of measuring and recording instruments, maintenance of refinery equipment and refinery transportation. In 1933, after motorizing the organization's teaming department, he reorganized the operation of the entire Atlantic truck and passenger car fleet. Mr. Laurie, who was educated as a mining engineer and for a time worked in that capacity, was SAE vice-president representing Transportation & Maintenance Engineering Activity in 1940, and also has served as a member of the Motor Truck Rating Committee, Transportation Division of the Standards Committee, and as Philadelphia Section Representative on the National Meetings Committee. In 1942 he was appointed technical consultant to Office of Defense Transportation, Motor Transport Division.

RUCKS?

by G. W. LAURIE
The Atlantic Refining Co.



■ Fig. 1 – A top-tank petcock installed at the correct cooling fluid level makes maintenance easier

general rule, it can be said that it functions reasonably well when new, even considering the diversity in types of operations, terrain, and climatic conditions to which the equipment may be subjected in this country.

In the operation and maintenance of automotive equipment, we as users have found that the rather important function of *maintenance of the cooling system* has been in many cases completely overlooked by the manufacturer, some to a lesser, others to a greater degree, depending on the model design.

For example, one of the basic instructions on equipment operations calls for maintaining the proper coolant level. There are, in some fleets, units that have the radiator fill and subsequent top hose so arranged that the top tank must be filled completely to the filler neck before one can see the liquid level. Any loss in coolant drops the level out of sight. It is quite obvious that attempting to operate at such a high level results in considerable loss of coolant both by expansion and surge. This is not so important in summer as in winter, the latter with its resultant loss of antifreeze. It should be stated here, however, that continuous and daily adding of even water in warmer operation is not advocated as satisfactory, if for no other reason than the greater possibility of scale deposition on the inside of the cooling system. The need for some type of a properly operating coolant-level indication is quite apparent. In this respect, suggestions have been offered on this problem (some actually installed and in operation) comprising:

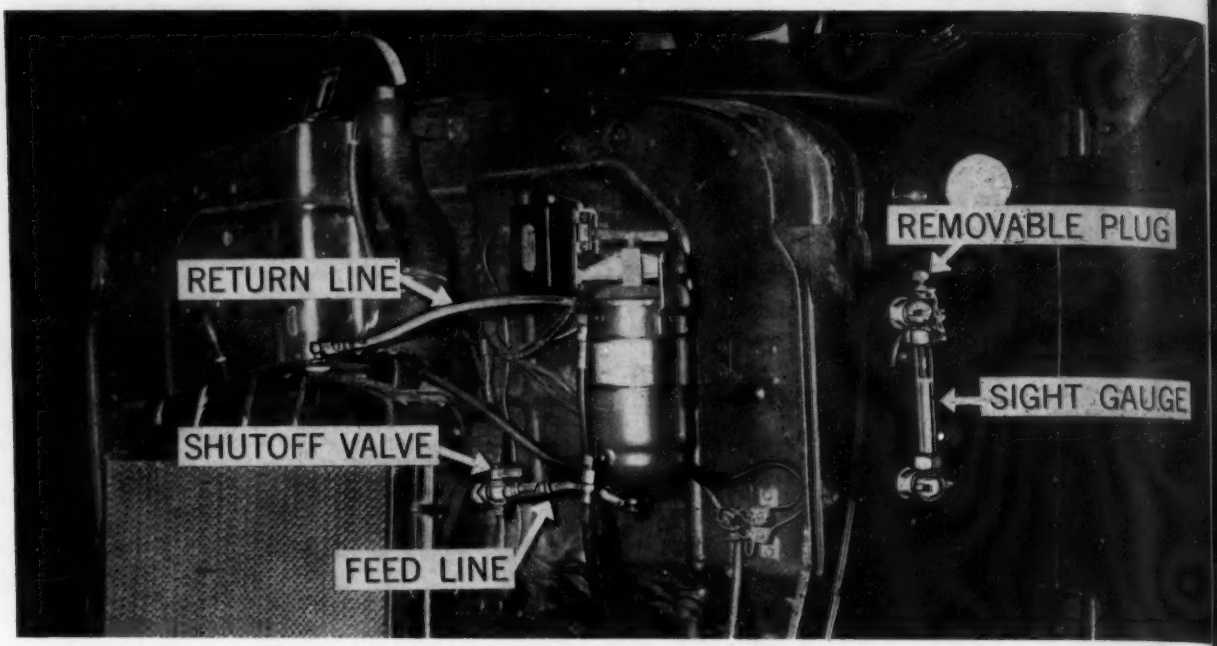
1. A readily accessible petcock installed from a point in the system where the coolant level is desired. Liquid is only added until it reaches the petcock location and none is added if liquid flows from the petcock when checked. (See Fig. 1.)

2. Another scheme which has been suggested and found practical is the installation of a liquid sight gage, also installed in a readily accessible location for observation. (See Fig. 2.)

The examples given above are an indication that a problem on coolant level does exist, requiring in some cases make-shift arrangements by the operator to overcome basic deficiencies in the system. It certainly would be more satisfactory to have the manufacturer provide the means in the original design for arriving at the desired solution.

Cooling systems should be made as leaktight as possible, and in lieu of any future revolutionary ideas on the art of cooling, then our present conventional system should at least be more accessible for checking, tightening, and replacing connections and parts. On some model trucks, it is practically impossible to note slight leaks in the cooling-system plumbing without removing a considerable number of parts and metal for observation. Bad leaks which are usually easily traced are not so much a problem as the slight leaks which go by unnoticed except when indicated by the necessity of regular liquid make-up. These so-called slight (even intermittent) leaks will reach considerable volume over a period, and are a source of expense particularly when the matter of antifreeze is considered. Recently, some experiments on this subject were completed, the purpose of which was to ascertain coolant losses in units which were considered reasonably tight and which would not normally be shopped for leakage repairs. During these tests, a certain number of newer trucks were used and data were obtained on permanent- and alcohol-type antifreeze materials, together with some units operating on water as control units. The work was done this past fall with atmospheric temperatures ranging from 40 to 75 F, the latter observed during midday. In order not to get involved in a discussion on the merits of types of antifreeze, I shall avoid mentioning comparative operating figures. However, since we are discussing cooling systems in general at this time, one of the significant points brought out during this investigation was that the *slight leaks* referred to above amounted to an average of 8 cc per mile, using water. On a day basis, this averaged 1.6 qt per day for the units in question. Liquid losses due to surge and/or expansion were overcome by operating the truck with a predetermined outage in reference to the overflow pipe level. From the above data we arrive at a figure of about 473 mpg of coolant lost; and, with a large fleet, the loss over a period of time becomes quite substantial.

The question might well be asked, "Why not reduce this leakage by tightening and maintenance?" On some of the units under consideration, the inaccessibility for even observation of all the cooling plumbing makes this problem considerably difficult for location of the slight leaks that we are endeavoring to observe. Even the use of a colored coolant or the usual telltale rust marks are difficult to see on these installations because of inaccessibility. Therefore, accessibility is not only a function of re-



■ Fig. 2 – Another method for maintaining proper coolant level – installation of a liquid sight gage installed in a readily accessible location for observation

moval or replacement but also, under preventive maintenance, a function of ease of inspection and observation.

You will note that it was necessary to keep the coolant level at some predetermined level below the overflow pipe to overcome losses due to expansion and surge. This source of coolant loss is quite prevalent in various fleet operations, judging by the number of references made to surge tanks in a questionnaire sent out on this subject. The indications are that some means is certainly desirable in post-war trucks to overcome this problem and, at this writing, the opinions lean to built-in surge tanks or providing surge capacity in the present type of top tank. Experiences of operators who have installed their own surge tanks have not been, in general, too successful.

We now come to the circulating pump, and our experience together with experiences of many other operators indicates that the present gland packing type of pumps are the source of considerable leakage and maintenance. Examples were submitted citing the packless pumps (as used in many passenger-car installations) as a trend of improvement for truck operation. Some operators have resorted to the use of stainless-steel type shafts and spraying of the inside of the pump and impeller with rust- and/or corrosion-resisting materials to improve the wearing quality of the present truck pump. It is evident that fleet operators desire that the important functional parts of the cooling system receive attention by the manufacturer of the post-war trucks.

Few will deny that it is desirable to bring operating temperatures up to normal quickly and to maintain fairly uniform temperatures during operation. Three types of controls are in general use for accomplishing this: namely, the block or line thermostat; the radiator shutter, automatic or manual operated; and cardboard (often cut from advertising signboards) or covers made of fabrics, placed in front of the radiator. The necessity of resorting to placing cardboard in front of the radiator surely indicates that engine operation in cold weather is unsatisfactory without blanking-off some radiation capacity. All three systems described are wanting in one feature or another, and operators in general agree that improvements can and should be made for better temperature control.

In summing up the remarks of operators on this question, there is a general agreement on the following:

1. Cooling systems with all connections should be more accessible so that they can be readily checked and serviced.

2. Coolant operating level to be more clearly defined.

3. Consideration be given to some type of surge and expansion control to overcome liquid loss from this source.

4. Better water pumps with reduction in rust, corrosion and leakage. (Packless pump mentioned in a number of reports.)

5. Better thermostats in all trucks and consideration to bypass systems, particularly in the larger units.

6. Provision for ease of cleaning cooling systems with larger-size drain facilities.

7. Under-hood temperature control, particularly along the lines for quicker warm-up.

Some suggestions that received individual mention for improving cooling-system maintenance and operation were:

Drain opening of at least 1 in. diameter and unobstructed.

Expansion tank with water return.

Reasonably even thermal distribution without local hot spots.

Liquid cooling (that is, ethylene glycol) as is current practice in aircraft engines.

Steam cooling.

Heat interchangers between engine oil and coolant liquid.

Easily removable oil pump screws.

Sealed cooling systems.

Depth gage for water level.

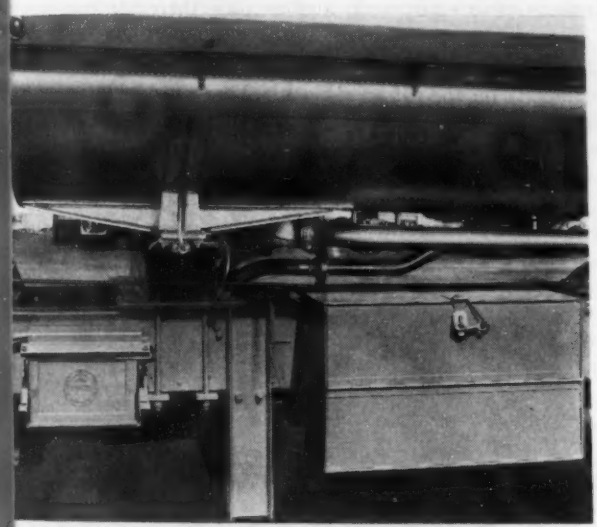
Audible water level and heat indicator.

Closed systems with high-pressure heat exchangers and better plumbing.

Thermostatically controlled variable-pitch fans.

■ Maintenance and Accessibility

Accessibility for maintenance shares the limelight in its unfavorable comments from the operators in general. In appraising this question of accessibility, it becomes readily apparent that the starting point would be consideration of those items requiring frequent attention, some daily. Fuel, oil, and coolant must be checked daily; therefore convenience for these operations is highly desirable. The cooling system has been covered in detail above. The engine oil filler in many designs could be considerably improved by providing easy access to the oil fill pipe, and dip stick used should be easily returned to position, as one operator comments, "without the use of flashlight or ladder." The fuel tank and fill opening, although accessible in most installations, receives its share of comment from two other directions:



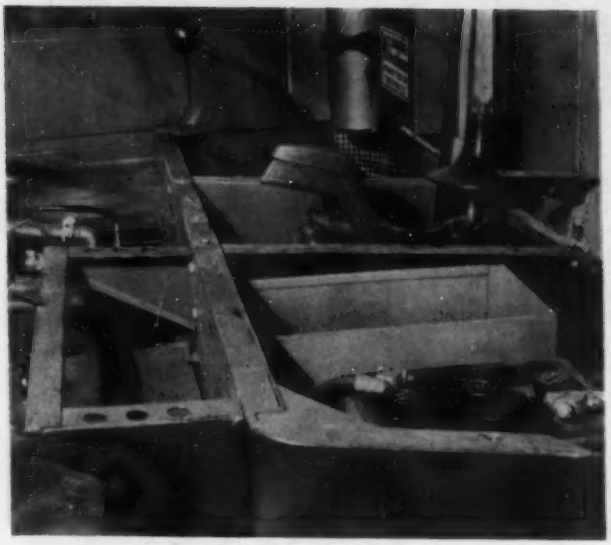
■ Fig. 3 – A battery located for ease of servicing

(a) Construction and strength.

(b) Spillage of fuel during filling due mainly to the inability of the tank to absorb the normal discharge rates of service station pumps.

The disadvantages of (b) are so apparent that no further comment is necessary. Under (a), practically all operators agree that the average fuel tank as supplied with the equipment needs to be fabricated better and be made of heavier material.

The matter of accessibility and maintenance of various items of the equipment in their order of importance depends to a large degree on the operation and the individual operating policy, so that the following discussion will not be classified in any attempt to adhere to such an order.



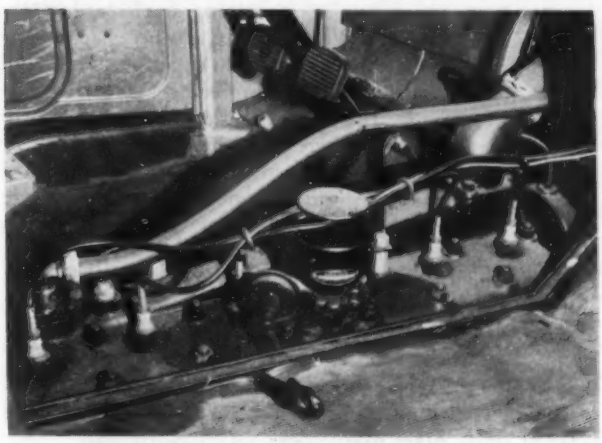
■ Fig. 4 – It is necessary to remove the seat cushion to service the battery in this location

■ Electrical

The electrical system, including ignition, in many of the operations reporting, constitutes a large percentage of the maintenance effort, and ease of accessibility would be one of the major factors in improving the overall picture.

The battery should be located for ease of servicing, including checking and filling, removal and reinstallation, washing and/or cleaning exterior. (See Figs. 3 and 4.) The battery box or holder must be strong and well supported and the battery hold-down arrangement be more positive in holding the battery firmly in position. The chaffing and short-circuiting of the battery cables in most instances are traced to the battery hopping around in the holder and wires not properly protected from sharp edges.

Spark-plug maintenance in many models is quite simple; in others, it becomes a major undertaking. (See Fig. 5.) For example, one operator reports that it takes ¾ to 1 hr to remove No. 1 plug in a certain make truck, whereas another operator indicates that on a certain group of trucks in his fleet it is practically impossible to remove No. 6 plug



■ Fig. 5 – Spark plugs, distributor, and other items requiring routine maintenance are readily accessible in this design

■ Fig. 6 – Oil filter mounted on a base integral with or directly connected to the engine block eliminates external connections

without special tools and a contortionist mechanic. In general, with the exceptions cited, we find that plugs would be accessible if they were not surrounded by hose, brackets, and miscellaneous accessories.

Distributors usually are on some type of periodic inspection and adjustment in most preventive maintenance programs, and a pertinent suggestion was made to bring this unit higher in some units by some extension drive for ease of maintenance, particularly where the distributor is located in a low position.

Electrical circuits and wiring, with but few exceptions, were condemned by operators because of inferior quality and lack of accessibility. The subject is of such importance that it cannot be dismissed lightly by simply stating that it must be improved. It is one of the modern mysteries of our times why the workmanship of automotive wiring, together with poor quality of wires, switches, connectors, fuse blocks, or lack of adequate fuses, is such as we find, when we consider the slight extra cost for the best, and the fact that wiring installations are presumed to be installed for the life of the truck. It only takes one quick glance behind the instrument panel to see the conglomeration of intertwined wires with open connectors for proof of our statement concerning haphazard workmanship and design. When it becomes necessary to rewire the cab wiring, then you have an expensive, long, and tedious job in front of you. Before being asked why it is necessary to rewire, the question can be answered here simply by stating that where the wire insulation has not become brittle from the heat generated by inadequate size for the current carried, it will in time reach the same stage of failure by aging. There are also instrument panels which are hinged for maintenance accessibility to wiring and instruments. Here again, the wires are not secured and arranged properly to prevent the accidental pinching of one of the wires on reclosing the panel section. A short circuit from such pinching is the usual result of an otherwise completed repair job. The proper fastening of wires to the chassis running to the rear of the truck in many types of units leaves much room for improvement.

As stated above, it is the contention of many operators that the wiring on a truck should be designed and installed with the view of lasting the life of the unit, and towards this end it is the earnest desire of operators to have at least the following points considered in the post-war truck:

1. Wire size to be adequate for the load carried.
2. Better workmanship in installation, and, where a number of wires are together, have them bound or placed in conduit or loom.
3. Better insulation with noninflammable characteristics.
4. More liberal use of fuses and fuse blocks of good construction.
5. Firm and positive junction points.
6. Wiring running to rear of unit be well supported with good clips.
7. Stronger and well-designed electrical switches.

The interesting part of this discussion is that all of the above requested items have been available on the market, and we, like others, did procure *quality* electrical items to replace the original, inferior-quality parts installed by the manufacturer after the trucks had been in service for a relatively short time.

■ Oil Filters

The oil filter has become a standard accessory in most automotive equipment, and as such there are two points that might be worthy of consideration:

1. Oil filter base to be an integral part of/or mounted directly on the cylinder block. This would eliminate to a large degree the use of external connection. (See Fig. 6.)
2. Since oil filtration on most present-day oil filters is

■ Fig. 7 – Steering wheel interferes with brake and clutch operation by long-legged driver

directly related to temperature, the use of a jacketed filter, whereby use is made of the engine cooling-system liquid to elevate filter temperatures, appears to have merit for consideration.

■ Engine Valves

Accessibility for engine valve maintenance seems to be centered around, first, the location of the engine in respect to cab or chassis; and, secondly, to the general use of cylinder heads as a mounting place for miscellaneous accessories. Fortunately, most engines give a fairly creditable valve-operating performance to help reduce the periodic checking or replacement of valves. The proper functioning of valves and their condition must be maintained if the engine is going to perform the work it has been designed to do. The importance of engine valves, therefore, dictates that their accessibility be kept within reasonable limits consistent with the general chassis design, and the avoidance of using the engine head as a mounting place for miscellaneous accessories.

■ Unit Replacement

We now come to the matter of unit replacement, and our survey of fleet operators on this question, worded to cover primarily ease of engine replacement in the chassis, resulted in two classifications of replies:

1. The medium- and heavy-duty truck operators, as a general rule, indicated their desire to have the powerplant more easily removable from the chassis. They cited that at least one item covering engine removal would aid the problem, that being, to have certain connections of water, air, oil, and fuel easily broken and reattached.

2. The light truck operators went on record to the effect that engine removable in this truck classification is not generally practiced and see no reason for any major redesign to permit accomplishing this removal feature. Of course, they would not object if the design were incorporated to make it quite easy to remove the powerplant if it could be done without extra cost to the original buyer.

Accessibility and unit replacement were answered in no uncertain terms when the question was put to fleet operators, and most of them feel that accessibility is of major importance. They would be willing to sacrifice compactness and perhaps even a certain amount of maneuverability in order to gain ease of repair. This is particularly important when we consider the high cost of labor now prevailing and when we realize that this high labor cost per hour will probably remain after peace has been declared.

In line with this, easily removable units may permit a unit repair system which will prove both economical from a repair standpoint and effective in keeping equipment on the highway a maximum period of time and in the shop a minimum.

It might be worthy to note that the repair system of the Army is based upon unit replacement and overhaul, and we can feel sure that, as a direct result of this method of teaching, the newer mechanics in the trucking industry will be unit conscious.

■ Cold-Weather Engine Starting

The post-war truck should benefit by the design and performance of automotive equipment as developed for the strenuous job of waging warfare. For example, in cold-weather operation, the starting and warming up of equipment under severe conditions is aided by some form of heating device which functions while the equipment is parked or stored for any appreciable period at low temperatures. For the most part, truck operation in this country is not subject to —20 to —40 F; nevertheless, even relatively mild winter weather of 0 or 10 F creates a considerable problem in starting and warming up for the average truck parked outdoors, or in an unheated garage.

The ease of starting a cold engine is affected by many variables, such as, fuel, oil, electrical system, battery, mechanical condition of the engine, and temperature. Although fleet operators were not unanimous in asking for some type of coolant or crankcase oil heater, the majority felt some form or type of heating to be advisable.

Quoting from a large fleet operator, "Some type of temperature control for oil and/or coolant is very desirable. For winter, this could take the form of an electric heater that could be plugged in a lighting system when the truck is not in use."

The advantages of quick starting and warming up are reflected in decreased wear, decreased general maintenance costs, and saving in fuel. The problem of starting and warming up may be attacked from other angles; however, the purpose of this section of the paper is to indicate that a cold-weather starting problem does exist and should be added to the list of items "the fleet operator desires improved in post-war trucks."

■ Rustproofing

Rustproofing of exposed under metal parts, including inside of cab door panels, was generally accepted as a desirable item, though again not unanimously. Here again, with divided opinion, those favoring rustproofing were quite emphatic on the subject, even to the extent of willingness to pay extra to secure it. With a large percentage of our own fleet retained over a rather long operating period before replacement, we see economic justification in rustproofing. No doubt those operators who can justify operating equipment over a rather long time period can justify rustproofing to eliminate expensive replacement of those parts rusted through. In line with this type of conservation, mention should be made of untreated wood used in cab construction which degenerates into a powder form after an extended period of operation. This type of maintenance, involving either rebuilding the cab or replacing it, should not be contingent on the matter of structural wood failure, as this source of trouble can be so easily overcome by using treated wood.

■ Truck Cab

This now leads us to a discussion of truck cabs, about which practically all fleet and truck operators had considerable to say. Better cab vision is essential, and most truck cabs could be improved in this respect. Even in engine-under-seat jobs where vision should be at its best, the designer has not sufficient glass, or has placed posts and framework at an undesirable location. An improved cab with improved accessory installation is important to driver comfort. A large driver cannot properly act in emergencies when squeezed into some of today's cabs. (See Fig. 7.) Windshield defrosters, cab heaters, flare contain-

■ Figs. 8 (left) and 9 (right) – Accelerator pedal operation so cramps driver's leg that it is necessary to place a block under his heel

ers, fire extinguishers, boxes for driver's records, and so on, are all standard and necessary equipment today and should be designed and built in the cab as integral units.

■ Standardization and Safety

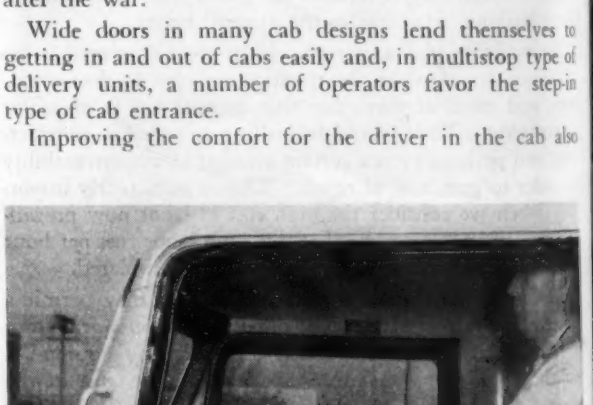
A typical reply from an operator will be used here as representative of thoughts on this question: "A lot of safety could be built into the present cabs, which are full of blind spots, cramp the driver, and generally encourage driver fatigue. (See Figs. 8 and 9.) Windshields should be high enough for a reasonably tall driver to see out of without having to keep his head ducked down on his chest. (See Figs. 10 and 11.) Seats should be adjustable so that both the short and tall men can reach the controls and still be able to sit in a comfortable position. Built-in windshield defrosting and cleaning is a must."

There has been a general standardization of the location of the foot pedals, which should be continued. However, the hand-brake type and location does differ considerably in various cab designs. There is need for a standard hand-brake lever easily reached. It should be mounted in a standard position and operated by pulling back, as shown in Fig. 12. On some trucks the hand brake requires pulling up on the lever to set the brake, as shown in Figs. 13 and 14. This is awkward and difficult, which lends itself to having a runaway truck.

Most cabs of today could be roomier and permit easier entrance and exit. It appears that where the manufacturer had to stay within certain legal unit overall lengths, he increased body length at the expense of the cab. In this respect, it should be mentioned here that more uniformity in state regulations would aid the manufacturer in designing vehicles that can be used universally. No doubt the lifting of restrictions by some states for the duration will help in bringing about more uniformity in regulations after the war.

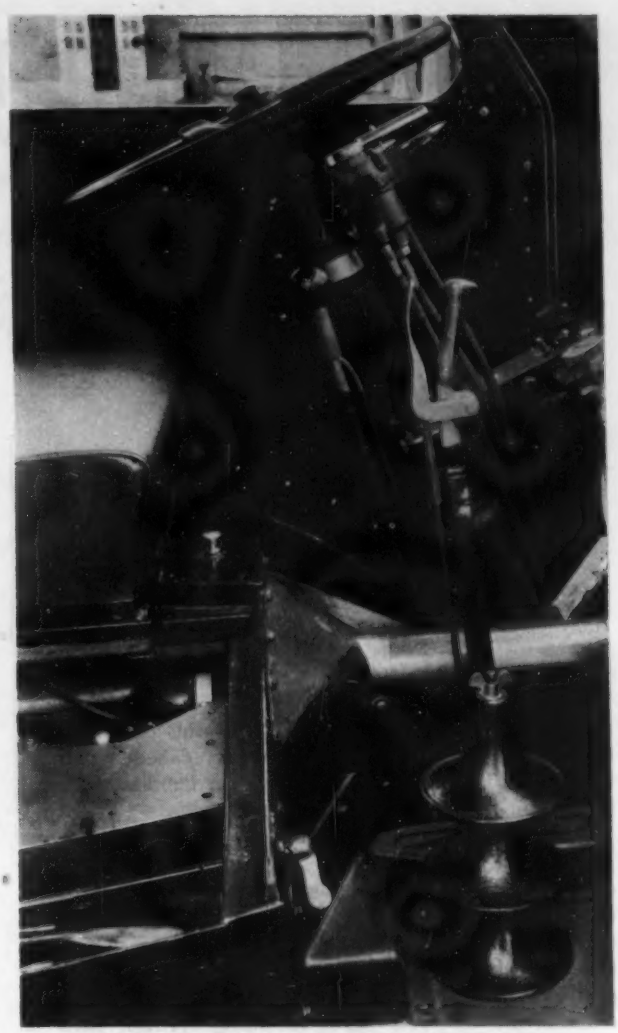
Wide doors in many cab designs lend themselves to getting in and out of cabs easily and, in multistop type of delivery units, a number of operators favor the step-in type of cab entrance.

Improving the comfort for the driver in the cab also



■ Fig. 10 – Windshield does not permit tall driver a full view of the road

■ Fig. 11 – Only by ducking his head can the driver see out of the windshield of some cabs



■ Fig. 12 – The hand-brake lever should be mounted in a standard position and operated by pulling back

entails better sealing around doors and pedals, insulation of the cab roof for summer heat, and exclusion of engine fumes and heat from the cab. In some models, cab parts directly over the engine exhaust line get quite hot, in fact too hot to touch; this added cab heat may be welcomed in winter but condemned quite emphatically by the driver in the summer. Seals around the door are a function of winter comfort, and some operators have had sponge rubber sealing fastened to the door with glue that had a rather short operating history. Where this rubber became loose or disintegrated, the draft in the cab during cold weather was quite disagreeable. The manufacturer subsequently improved the situation by applying weather stripping by another more positive means. The sealing of *passenger-car interiors* from heat, cold, fumes, noise, together with comfort details, certainly indicates the necessary thought that was given by the manufacturer to the problem, and it is hoped that the truck cab in the future will be guided by such accomplishments.

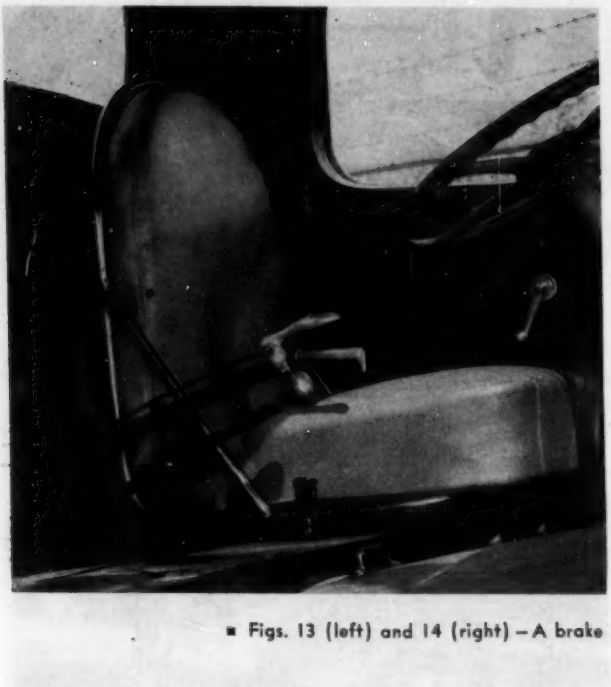
Cab hardware, on the other hand, appears to be one item borrowed from standard passenger-car design and construction which is simply not sturdy enough for the truck cab. Here again, on the multistop unit and also on other types of operations, the frequent use of door handles, latches, controls, and so forth, dictates hardware construction in keeping with the severity to which it is put to use. Finally, before leaving the subject of cabs, there are operations requiring certain treatment and design, for example, power take-offs and controls, which should be built in and not hooked up after the purchase of the unit. Utilities and some over-the-road operations require larger-capacity cabs, that is, five- or seven-man cabs for crew members and, in the latter operation, sleeper accommodation, and these operators indicate that better accessibility is desired, particularly on the COE units.

■ Bumpers

One of the main functions of bumpers (excluding decorative types) is to minimize or to prevent damage to the vehicle to which it is attached. It is certainly desirable that, where a collision involves two vehicles equipped with bumpers, contact be made between the bumpers. In this respect, a survey of bumper heights reveals a closer standard height of front bumpers than rear bumpers, which is obvious, considering the various types and construction of bodies. Without attempting to designate any standard dimensions, suffice to say that the advisability of installing substantial bumpers at heights to give the greatest coverage for the number and types of vehicles on the highway is worthy of consideration.

■ Safety

Under the heading of safety, the operators had much to complain about. The following two paragraphs appear



■ Figs. 13 (left) and 14 (right) – A brake lever that requires pulling up is awkward to set

to be in addition to the general complaints of insufficient head room, poorly arranged cab interior, inadequate vision, unbalanced brakes, steps too high, grab handles too high, ventilation inadequate, and many others.

"During the war period, disclosures of malfinished stress parts – particularly parts of steering mechanism – have come to notice. Fatigue failures of such parts have been initiated at places where inadequate fillets, sharp corners, scratches, tool marks, grinding scratches, sharp changes in section, and so on, have been the occasion for 'stress raisers' resulting in ultimate failure. Attention to detail of manufacture of stress parts, including springs, parts of steering mechanism, universal joints, and drive shafts, would, at small additional expense, obviate the fatigue failures which too frequently now occur in such parts.

"Appended below you will find statistics obtained from examination of 1200 accidents involving tractor-semitrailer combinations, selected at random throughout the months of the year in equal numbers and 600 accidents involving trucks (single units) similarly selected. The data are separated as to accidents involving collisions and noncollisions, respectively, with a separation for each category as to the number of overturns and nonoverturns. A brief analysis follows the tabular material:

	Tractor-Semitrailers		Trucks		Both	
	Number	%	Number	%	Number	%
Collisions						
Overturns	100	9.9	36	6.9	136	8.9
Nonoverturns	915	90.1	488	93.1	1403	91.1
Total	1015	100.0	524	100.0	1539	100.0
Noncollisions						
Overturns	118	64.0	31	40.7	149	57.0
Nonoverturns	67	36.0	45	59.3	112	43.0
Total	185	100.0	76	100.0	261	100.0
Collisions and Noncollisions						
Overturns	218	18.2	67	11.1	285	15.3
Nonoverturns	982	81.8	533	88.9	1515	84.7
Total	1200	100.0	600	100.0	1800	100.0

"The most important items are:

1. The percentage of overturns in noncollision accidents, both for those involving tractor-semitrailer combinations and straight trucks, are disproportionately high compared to collision accidents.

2. The percentage of overturns in which tractor-semitrailer combinations are involved in noncollision accidents is about 50% greater than for single-unit trucks; and the disproportionality is of the same order for tractor-semitrailer combinations and collision accidents.

3. Without regard to type of vehicle or type of accident, overturns occur in about 15% of the cases involving collisions and noncollisions, there being a small category of accidents such as those involving "fire only" not defined either as a collision or a noncollision accident.

"The percentage of overturns involving tractor-semitrailer combinations is so much greater than the percentage involving straight trucks as to be strongly indicative of the possibility of design improvement such as to cut down the disproportionality."

■ Miscellaneous

The installation of fancy grille work, decorative bright metal, and so forth is not too favorably received by the truck operators, for the simple reason that it serves no useful function and, as a matter of beauty, it loses its appeal once the unit has left the showroom floor. What the operator desires is good sturdy construction, which in itself can be made quite pleasing to the eye without resorting to excess decoration. Eye appeal in the attempt to make some type of trucks look big gives the operator no greater payload privilege and usually results in a cab design leaving much to be desired in comfort and accessibility. It appears that the operators are not requesting the

concluded on p. 158

On the STRENGTH of Highly Stressed, Dynamically Loaded BOLTS and STUDS

by J. O. ALMEN
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THE strength of most highly loaded bolts and studs is determined by the man with the wrench and not by the designer, the metallurgist, or by the manufacturing processes. In highly loaded bolts and studs we rarely find the design so bad, the fabrication practice so poor, or the material so weak as to cause failure in service provided the nut is properly tightened against reasonably rigid members.

There are, of course, advantages to be gained by careful design. Good materials and heat-treatments should be used; some manufacturing practices are better than others, but all of these are relatively unimportant in severe service in comparison to proper tightening of the nut during assembly and to maintaining that tightness during service. We should, therefore, be less concerned with design details and give more attention to the technique of tightening nuts and to the ability of the bolted assembly to maintain the required tightness.

An estimate of the relative responsibility for the strength of bolts and studs among the individuals who are concerned from design to assembly is shown in Fig. 1.

A properly tightened nut is one that applies a tension load to the bolt or stud that is equal to or greater than the external load that is to be supported in service. When this condition is fulfilled and maintained against reasonably rigid bolted assemblies, the bolt can not fail by fatigue because it can experience practically no change in stress regardless of the fluctuating nature of the operating load. It can not fail statically because, to be tightened as specified, it must be capable of supporting the greatest operating load.

Fatigue Strength Varies with Stress Range

Numerous fatigue tests have shown that the fatigue strength of metal decreases as the range of dynamic stress (stress change) to which the metal is subjected is increased and, conversely, that the fatigue strength is increased as the dynamic stress range is decreased. As the stress change approaches zero, the dynamic load that can be supported approaches the tensile strength of the material.

Consider the case of a bolt subjected to fluctuating load, such as in a connecting rod. If the nut is tightened just enough to make contact with the bearing cap, the load that will be applied to the bolt will vary from zero, the initial tightness, to the maximum inertia load of the piston and the connecting rod. Under this large stress change,

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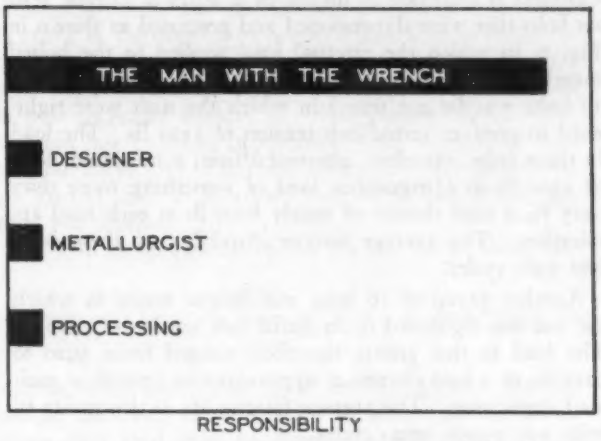

■ Fig. 1 – Relative responsibility for fatigue durability of bolts

EXTENSIVE study by Mr. Almen on the strength of highly stressed, dynamically loaded bolts and studs has led him to the following conclusions:

1. The strength of dynamically loaded, highly stressed bolts and studs is determined by the man with the wrench. Other considerations, such as design, material, and processing, are of relatively minor importance.

2. The elasticity of bolts and studs should be as great as possible.

3. The bolted members should be as rigid as possible.

4. Loss of dimension of the bolted members is particularly hazardous to the strength of short bolts and studs.

5. No adequate means are available for determining nut tightness except when the elongation of the bolt can be measured.

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the strength of the bolt will be less than one-fifth of its strength under static load and fatigue failure can only be avoided by making the bolt very large. If we now tighten the nut so as to load the bolt to one-half of the inertia tension load, the stress change will be one-half as great and its operating strength may be one-third to one-half of its static strength. When the nut is tightened against reasonably rigid abutments to produce a tension in the bolt *equal* to or greater than the working tension load, there is practically no stress change in the bolt and its operating strength, therefore, approaches its static strength.

■ Bolt Fatigue Tests

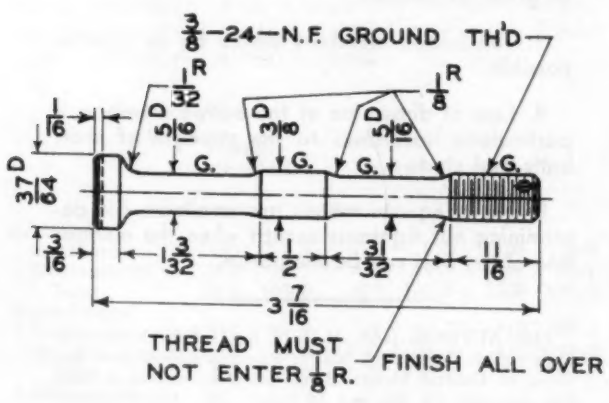
The increased fatigue strength of bolts as the initial tightness is increased is shown by a series of fatigue tests on bolts that were dimensioned and processed as shown in Fig. 2, in which the external load applied to the bolted members alternated from zero to 9215 lb. One group of 17 bolts was fatigue tested in which the nuts were tightened to produce initial bolt tension of 1420 lb. The load in these bolts, therefore, alternated from a minimum load of 1420 lb to a maximum load of something more than 9215 lb, a load change of nearly 8000 lb at each load application. The average fatigue durability of these bolts was 5960 cycles.

Another group of 16 bolts was fatigue tested in which the nut was tightened to an initial bolt tension of 5920 lb. The load in this group, therefore, ranged from 5920 to 9215 lb, or a load change of approximately 3300 lb at each load application. The average fatigue life of this group of bolts was 35,900 stress cycles.

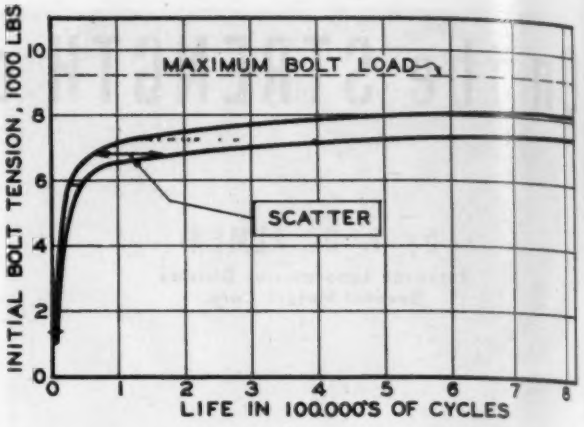
In a third group of 15 bolts, the load alternated between the initial nut tension of 7220 and 9215 lb, a load change at each load application of about 2000 lb. The fatigue life average of this group was 214,500 stress cycles.

In the fourth group, only two bolts were tested, one failing after 4,654,000 stress cycles and the other had not failed when the test was stopped after more than 10,000,000 stress cycles. These bolts were initially tightened to 8420 lb, and since the external load was the same as for the other groups, the load change at each load application was only 800 lb.

The results of these tests are graphically shown in the chart in Fig. 3, in which the initial bolt tension in pounds



■ Fig. 2 – Test bolt detail



■ Fig. 3 – Effect of initial tightness on bolt life

is plotted against the fatigue durability in stress cycles. The test points for the two bolts tightened to 8420 lb initial tension are not shown because their inclusion would require contraction of the life scale to undesirable proportions.

These test data are also shown in the log-log plot in Fig. 4, in which the load or stress range to which the bolts were subjected is plotted against durability in stress cycles. For this plot, the stress range R is calculated as:

$$R = 1 - \frac{\text{Minimum Stress}}{\text{Maximum Stress}}$$

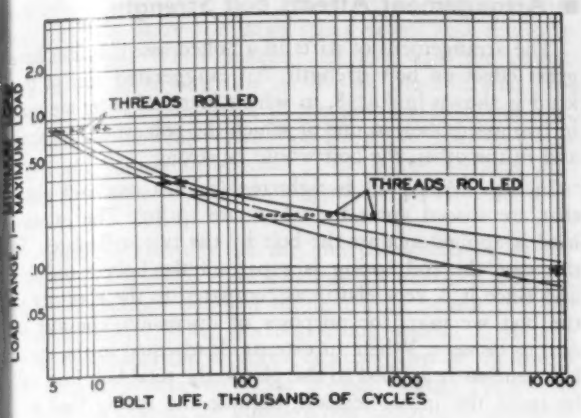
Log-log plotting is used because the lines joining the test points are only slightly curved, which permits estimating the trend of the curve between test points and also permits projecting the curve into the region of the fatigue endurance limit.

■ Variation in Durability

Note that there is a considerable variation in the durability of the bolts in each group of bolts tested. Similar fatigue-life variability occurs in all kinds of specimens because it is impossible that any specimen can be exactly like any other specimen, just as no person can be exactly like any other person. The fatigue-life variability among these bolts, as is shown in Fig. 4, is small as compared to the life variability of other types of machine parts. The ratios of life of the best to the poorest in each group, in decreasing order of stress range, are 1.5, 1.65, and 4.0. Although there are not a sufficient number of tests in each group to establish definitely the limits of the scatter band, increased scatter with increased life is to be expected from the nature of the fatigue curve. The data on bolt fatigue tests that have been given above are summarized in Table 1.

Table 1—Bolt Fatigue Test Data

Initial Tension, lb	Operating Load, lb	Load Change, lb	Stress Range	Average Life, cycles	Life Variation
1420	0 to 9215	8000	0.85	5,960	1.5
5920	0 to 9215	3300	0.36	35,900	1.65
7220	0 to 9215	2000	0.215	214,500	4.
8420	0 to 9215	800	0.087	5,000,000+	-



■ Fig. 4 – Bolt fatigue curve

■ Stress Fluctuation Unavoidable

When the nut is insufficiently tightened, each load application will elastically elongate the bolt and a gap will be formed between the bolted members, the magnitude of which will be reduced as the initial tightness is increased. When the nut is tightened to near the operating load, this separation of the bolted members will not occur, but there will, nevertheless, be a small elastic elongation of the bolt.

When the nut is tightened, the bolt is elastically elongated and, at the same time, the bolted abutments are elastically shortened; the amount of shortening depending upon such variables as the design, the elastic characteristics of the bolted materials, and the area of the material supporting the bolt load. On application of the external operating load, the bolt is acted upon by the external load plus the load of elastic recovery of the bolted assembly. The sum of these forces will always be greater than the initial bolt tension and, therefore, the bolt will experience a change of stress at each load application regardless of the magnitude of the initial tension. The stress change will be greater as the elastic deformation of the bolted assembly is increased and it will be reduced as the rigidity of the bolted assembly is increased.

■ Stress-Strain Measurements

As a preliminary to the fatigue testing of the bolts described above, stress-strain measurements were made for each bolt. The fatigue machine used for these tests is equipped with gravity scales whereby the load applied to the specimen may be accurately weighed at any position of the load-applying device. It was, therefore, a simple matter to take stress-strain measurements on each bolt while in place in the fatigue machine by weighing the load and measuring the elongation of the bolt at frequent intervals.

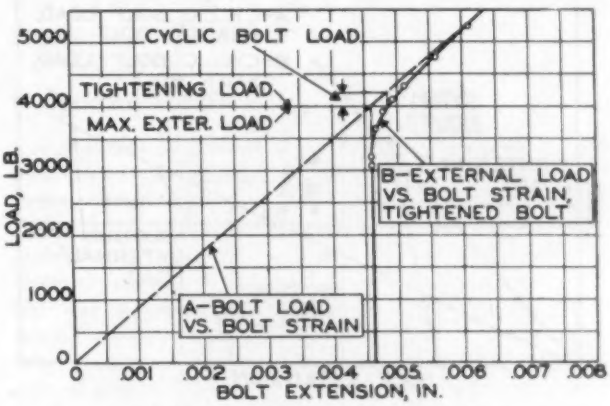
A typical plot of stress-strain data taken from one of the test bolts is shown in Fig. 5. In this plot the inclined straight line, having its origin at zero load and zero elongation, is the load extension curve of the bolt alone (curve A). The plotted points, curve B, were taken from a bolt that had been tightened against relatively rigid abutments to an initial load of 4000 lb, under which load the bolt elongated 0.0046 in. As the external load was applied to this tightened bolt, the load extension curve B, as shown by the plotted points, does not rise vertically to join the inclined load extension curve of the bolt alone (A) as would occur under ideal conditions of an elastic bolt and perfectly rigid abutments. The plotted points (B) join the vertical line at 0.0046 in. elongation and the inclined line (A) by an easy curve thus showing that the bolted members are elastic and that the load on the bolt is augmented by the elastic recovery of the bolted members.

■ Effect of Elasticity

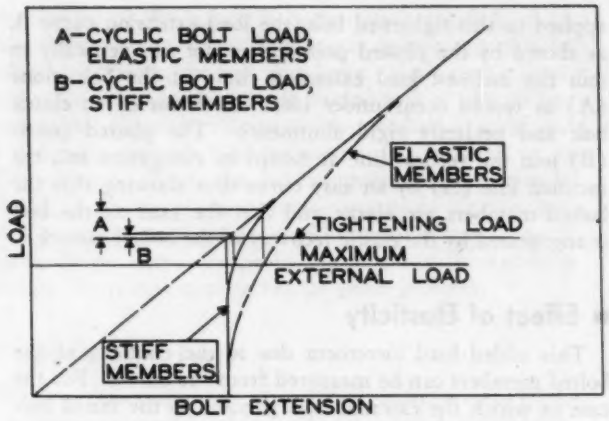
This added load increment due to the elasticity of the bolted members can be measured from the curves. For the case in which the external load is equal to the initial bolt tension, extend the bolt preload line (4000 lb) to intersect the curve of the plotted points *B*, which will be found to be at a bolt extension of 0.00485 in. Since the extension under the bolt preload was 0.0046 in., the bolt elongation has now increased 0.00025 in. Now project the new bolt extension line (0.00485 in.) vertically to intersect the load extension curve (A) of the bolt alone. This will correspond to a load of 4220 lb or a load increase of 220 lb over the initial bolt tension.

The load range experienced by the bolt under these conditions would be $1 - \frac{4000}{4220} = 0.052$, a stress range so small as to be practically zero (see Fig. 4) and the bolt load is, therefore, almost static. In like manner, the resultant bolt load may be found for any external load whether it be greater or less than the initial bolt tension.

The plot Fig. 5 was made from a fatigue test fixture in which the bolted members were relatively massive and rigid and care was used to assure parallelism and smoothness of the contacting surfaces. For these reasons the elastic compression curve of the bolted members is so steep as to be immeasureable and it, therefore, coincides with the vertical preload line at 0.0046-in. bolt extension up to a load of about 3400 lb. From this point the elastic curve (B) swings to the right and joins the bolt elastic curve (A) at about 0.0060-in. bolt extension and 5220-lb load. The elastic compression curve for this bolted assembly is such that external loads up to 3400 lb add very little to the bolt preload. The nonlinear portion of the curve is presumably due to bending of the bolted members result-



■ Fig. 5 – Determination of cyclic bolt load

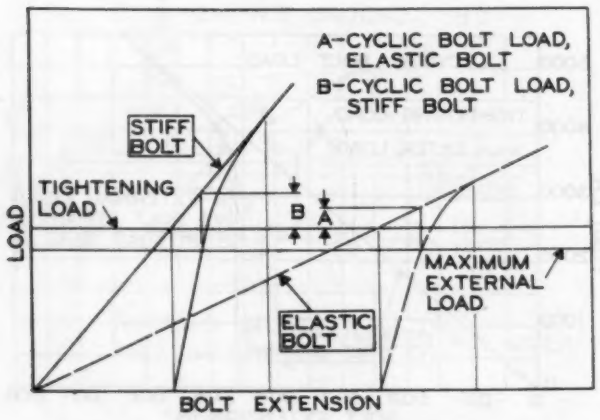


■ Fig. 6 – Effect of elasticity of bolted members

ing from imperfect surfaces and to bending of the loading beam under the external load. As the external load is increased through the nonlinear range, the cyclic increment of bolt load above the initial bolt tension increases rapidly until complete separation of the bolted parts occurs at 5220 lb and 0.0060-in. bolt extension.

The schematic chart Fig. 6 shows two elastic curves to illustrate the effect of varying the elasticity of bolted members. From the method of analysis described above, it will be seen that the cyclic load on the bolt is increased as the elasticity of the bolted members is increased. Similarly, Fig. 7 shows that the cyclic bolt load decreases as the elasticity of the bolt is increased.

The strength of a bolt may, therefore, be said to be influenced by the design of the bolted assembly and the materials used in the assembly. A properly tightened bolt will be weaker when used in combination with aluminum abutments than when used in combination with steel abutments of similar proportions because of the greater elastic deformation of aluminum. It will be weaker when used against abutments of small area than when used against abutments of large area. The number of parts making up the bolted assembly will also affect the bolt strength because mating parts will always fit imperfectly, thus adding bending deflection to elastic compression, and each part will, therefore, add to the total elastic deformation of the assembly.



■ Fig. 7 – Effect of elasticity of bolt

■ Arrangement Affects Bolt Strength

The arrangement of parts in a bolted assembly can have great effect on bolt strength. An exaggerated case of this kind is shown in Fig. 8, in which identical parts are used in two assemblies, in one of which the bolt would be weak and in the other, the bolt would be strong.

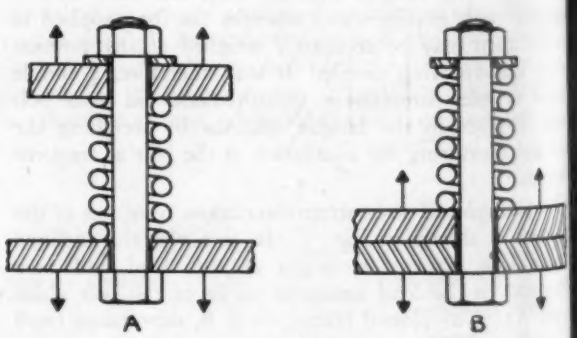
In Fig. 8A, a bolt is tightened against two stiff plates that are spaced apart by a low-rate spring. The external load is applied against the bolt by the two stiff plates. In this case the coil spring is a part of the bolted assembly and since it is very elastic as compared to the elasticity of the bolt we may, for purposes of discussion, consider its rate to be zero. When an external load equal to the initial bolt tension is applied to the plates, the total bolt load will be twice the initial tension, because the spring load will not be appreciably reduced by the additional extension of the bolt and its load will be added to the external load. The stress range experienced by this bolt will, therefore, be $1 - \frac{1}{2} = 0.5$. For the stress magnitudes of the bolts shown in Fig. 4, it will be seen that at a stress range of 0.5 early failure may be expected.

In Fig. 8B, the external load is also applied to the plates, but the low-rate spring now is a part of the bolt instead of a part of the bolted assembly as in Fig. 8A. We, therefore, have a very elastic bolt tightened against relatively very rigid members. When an external load equal to the initial bolt tension is applied to the plates, the load between the plates will be reduced to zero but the bolt will experience no appreciable change in load. The stress range will be $1 - 1 = 0$; that is, the load will be static and fatigue failure can not occur (see Fig. 4).

■ Practical Elastic Considerations

While the arrangements shown in Fig. 8 are rather fanciful, we have assemblies in actual practice that are somewhat similar. Fig. 9 shows a condition frequently met in practice in which a gasket is clamped between two mating plates. The gasket is compressed locally opposite the bolt, thus bending the plates until they serve as the spring shown in Fig. 8A. When the external load is applied, the bolt will feel the elastic load of the bent plates plus the external load, a condition that invites trouble.

Also in such assemblies the bolt or stud is usually short



■ Fig. 8 – *A*: spring is part of abutments – bolt is weak; *B*: spring is part of bolt – bolt is strong

nd relatively inelastic and we, therefore, have all the conditions necessary for large stress range and resulting tigue failure. When gaskets are necessary, the gasket rea should be large in the vicinity of the bolt and relaively smaller in proportion to the distance from the bolt, n order that the unit pressure on the gasket may be mainained with minimum bending of the bolted surfaces. The asket should be as inelastic as possible and still serve its urpose.

A practical assembly having the characteristics of the ssembly in Fig. 8B is shown in Fig. 10, in which a spring asher is clamped between the bolt head or nut and the olted assembly. When such a spring washer is dimenioned to support the required external load or the initial olt tension, whichever is the greater, within the elastic ange of the washer, the effective bolt elasticity is increased nd, therefore, the bolt fatigue strength is increased.

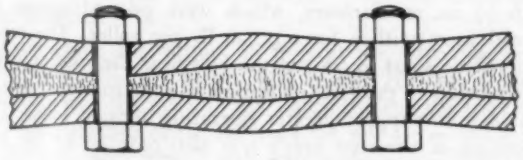
Likewise any other expedient that will increase the igidity of the bolted assembly or that will increase the lasticity of the bolt will increase the fatigue strength of he bolt.

■ Tightness Must Be Maintained

Although properly tightened bolts are stronger as the igidity of the bolted assembly is increased and as the lasticity of the bolt or stud is increased, a bolt or stud hat is incapable of maintaining an initial tension equal to he external tension load is quite likely to fail in severe ervice. Thus, we find relatively greater mortality among hort bolts and studs than among long ones because of heir lesser elastic yield.

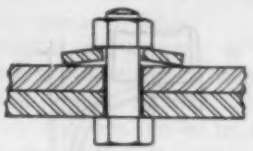
Consider a short stud such as is often used to attach ngine cylinders to the crankcase. The thickness of the ylinder flange plus the thickness of the washer may be ess than ½ in. When the nut is tightened, the stud is lastically elongated through the length from near the ottom of the nut to near the last thread engaging the rankcase. Assuming that this distance is ½ in. and that he nut, at proper tightness, stresses the stud to 120,000 psi, he stud will then be elastically elongated 0.002 in. If dur- ng operation, the bolted assembly is reduced in thickness y 0.001 in. through wear, corrosion, embedding, or by lisplacement of material such as soft plating, the stud will ose one-half of the required tension, and fatigue failure vill follow.

Greater safety is assured when longer studs are used. For example, when under the same conditions as described bove, a stud of 2-in. effective length is used, the loss of ension will be only one-eighth of the initial tension; that s, a loss of 0.001 in. from a total elastic elongation of



■ Fig. 9 – Bolts subject to fatigue – elastically deformed plates and gasket increase maximum and cyclic bolt loads

■ Fig. 10 – Spring augments bolt elasticity – bolt life is increased



0.008 in. This would be serious only when other factors, such as initial tension, are near the low limit.

It will, therefore, be seen that among the major bolt and stud hazards, particularly when they are short, must be included loss of thickness of the bolted assembly as a result of surfaces that can embed, soft-plated coatings that can be displaced, gaskets that can be plastically compressed, and materials that can yield at elevated operating temperatures.

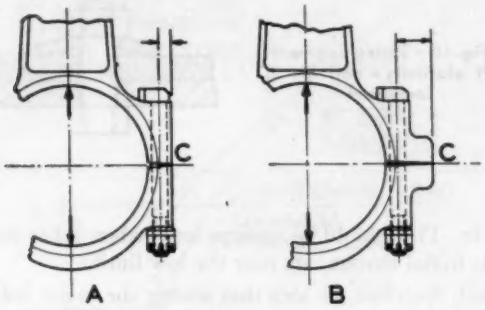
■ Yield Point May Be Exceeded

In tightening nuts, it is far better to stress the bolt above the yield point of the material than to risk undertightness. Since for adequate size bolts, the load is practically static, there can be no harm in yield provided that the plastic deformation is not so great as to reduce the static strength of the bolt. The amount of yield that can be tolerated will depend on the length of the bolt, the design of the bolt, and the characteristics of the material. When the body of the bolt or stud is equal in diameter to the outside diameter of the threads, yield will be concentrated at the roots of the threads whether the bolt be long or short. In such cases, little yield, as measured by bolt elongation, can be tolerated. For bolts or studs in which the body diameter is equal to or less than the thread root diameter, greater yield, as measured by bolt extension, can be tolerated since yield will occur over the entire body length.

It is not intended to recommend that all bolts should be tightened above the yield point but only to show that, where necessary, the practice may be followed provided proper caution is used. The necessity for tightening above the yield point would occur in such cases as when the bolt yield strength is only slightly greater than the maximum external tension load or when cotters are used in combination with castle nuts. In the latter case, it is often necessary to advance the nut beyond the required tightness, in order to admit the cotter since the nut should not be slackened to meet the cotter pin hole. For short bolts, two cotter holes should be provided at such angles as to register with the notches in the nut at 30-deg intervals, thus reducing the overtightness in critical cases.

■ Bending Loads

In addition to tension loads, a large percentage of bolts and studs are subjected to bending loads of various magnitudes. Stress changes from repeated bending loads are generally of smaller magnitude than stress changes from tension loads but since these occur simultaneously, the stress change in the bolt will usually be their sum. As in the case of repeated tension loads, the repeated bending loads, in normal designs, will be reduced as the nut tightness is increased and for the same reason. Also as in the case of tension loads, there will always be a small change in bending load regardless of nut tightness because of



■ Fig. 11 – Connecting-rod design

changes in the elastic deformation of the bolted members. When these changes in elastic deformation are not symmetrical with respect to the bolt axis, bending will occur.

The effective bending stress range can be increased or decreased by design, independently of nut tightness. For extreme cases, it is possible to design the bolted assembly so as to throw the bending stress change out of phase with the tension stress change. This can be done by elastically bending the bolt in a direction opposite to the bending induced by the external load as, for example, by drilling the holes in the abutments out of alignment by such amount that the bolt is straight when the tension load is greatest. The effective stress change would then be that of the tension component or the bending component only, whichever is the greater.

For ordinary design it is sufficient to avoid practices that may severely overload the bolt such as is illustrated in Fig. 11A. In this case the bolt may be excessively stressed in tension and in bending because the distance from the bolt to the point C is small. Since the bolted parts tend to bend about C as a fulcrum, the tension and bending loads on the bolt are great. Fig. 11B illustrates an improved design in which the fulcrum point C is farther removed from the bolt and, therefore, the tension and bending loads are reduced. This not only reduces the stress magnitude but also the stress range assuming, of course, that the nut is properly tightened.

■ Locknuts and Lock Washers

Although locking devices have apparently little to do with bolt strength, the question of their usefulness, in view of the tightness requirements for the class of bolted assemblies under discussion, may be of interest. The purpose of a nut lock is to prevent the nut from loosening, but a nut that is properly tightened does not require a lock because friction is ample to keep it in place. An improperly tightened nut or a nut that has loosened in operation does not need a lock because it is no longer performing its function and it is, therefore, immaterial whether or not the nut remains on the thread. When a free nut or bolt can damage other parts or when it is desired to save the parts as replacements, a lock may be helpful.

For the class of bolts under discussion, not only are locking devices of little value for the purpose for which they are intended but they often constitute serious hazards. Although locks of the washer type may sometimes add to the effective elasticity of the bolt, which is beneficial, they more often add to the danger of embedding and thus threaten the strength of the bolt, as has been discussed in connection with maintaining initial nut tightness. Locks of the friction type add to the difficulty of attaining proper nut tightness, as will be described. Cotters and wires are the least offensive of the commonly used locking devices and generally only their utility can be questioned.

■ Secondary Strength Factors

While considerations of bolt design, material, and processing techniques are secondary to many of the factors that have been discussed, they are not negligible. Since most bolts fail by fatigue, the design and processing should recognize this fact and precautions to reduce stress-raisers should be taken.

The most severe stress-raisers are the deep and often sharp notches formed by the threads. The stress at the thread roots can be reduced by several expedients which distribute the bolt load more uniformly among the threads engaging the nut. These expedients will not be discussed here because they have been adequately covered in several recent publications such as that by Arnold[1] except to warn that these publications discuss factors that are of low order of importance as compared to the importance of proper nut tightening in highly stressed clamping bolts.

As to the relative merits of rolled, ground, and cut threads, experience has shown that rolled threads are usually better than ground threads and that ground threads are usually better than cut threads, but this is by no means always true. Other factors than the process by which the threads are formed must be considered, such as the sharpness of the root groove and the state of residual stress at the root after the thread is formed. Perhaps the rolling process offers the best assurance of maintaining smooth radii because the die surfaces can be accurately formed and they can be readily inspected and because the process does not involve cutting, tearing, or variable wheel dressing. The state of residual stress, after forming, favors the rolling provided the bolt is not heat-treated, except perhaps a low-temperature draw, after rolling.

■ Thread-Forming Practices

The fatigue durability of ground, cut, and sometimes rolled threads can be increased by compressively stressing the material at the thread roots by a superficial rolling operation. Among the bolt fatigue tests recorded in Fig. 4 are five bolts tested with rolled threads. Three of these were tested at a stress range of 0.85 and the other two at a stress range of 0.215. It will be seen that the average durability of these rolled threads was about two times the average durability of the normal bolts at the same nut tightness.

The rolling operation applied to these bolts consisted of the application of a hand tool equipped with three rollers, each ½ in. in diameter, which were pressed against the thread roots with a force of 20 lb per roller. The rollers were made so as to contact the bolt near the root only, the object being to prestress compressively the material so as to enable it to support greater operating tension loads. The sharpness of the root notch was also reduced by the pre-

[1] See *Mechanical Engineering*, Vol. 65, July, 1943, pp. 497-506, "Effect of Screw Threads on Fatigue – A Review of the Literature," by S. M. Arnold.

ressing roller, but this effect was probably slight as compared to the compressive prestress.

Although the rolling operation increased the fatigue resistance of these bolts in the ratio of two to one, it will be seen from Fig. 4 that the rolling was only the equivalent of a slight change in stress range as would be obtained by greater initial bolt tension. Specifically, the unrolled bolts tested at stress range 0.846 would have had the same average durability as the rolled bolts had the nuts been turned 15 deg tighter. The unrolled bolts tested at stress range 0.215, which more nearly represents normal practice, would have had the same average durability as the rolled bolts had the nuts been turned 4 deg tighter. This comparison is not made to discourage improved fabrication practice, but only to emphasize the importance of proper tightening.

Commercial rolled threads can and often do take advantage of compressive prestressing intentionally or otherwise. If rolled threaded bolts are heat-treated after rolling, the prestress is lost. To retain the prestress, rolled threads should not be heated sufficiently to relieve the stress set up by the rolling or the bolts should be given a final rolling after heat-treating.

Stressing the thread roots beyond the yield point probably has an effect similar to overstressing springs, which is known to be beneficial in increasing their fatigue durability. Overloading should have other effects because it would result in better distribution of the load among the threads engaging the nut as well as reducing local stress concentration in the most highly loaded thread.

The details of design and the finish of the shank of bolts are perhaps of the same order of importance as the design and finish of the threads. The diameter of the shank should be made as small as possible consistent with the required strength in order to increase the elasticity of the bolt both in elongation and in bending. Generous fillets should be provided at all section changes and the reduced diameter should be carried up to the threaded section. Additional gain may be had by compressively prestressing the shank surface as by rolling or by shot peening.

Nut Tightening Practices

The secondary specifications of bolts and studs, such as chemical composition, heat-treatment, thread dimensions, body dimensions, and finish, are carefully controlled, but the really important consideration, nut tightness, is very loosely specified or entirely ignored. Among the reasons for this lack of control is the failure of engineers and metallurgists to appreciate the vital importance of nut tightness and the inadequacy of available means of measurement.

The most popular means of specifying nut tightness is through the use of torque wrenches, but it is no exaggeration to say that a good mechanic who has developed wrench "feel" is more reliable than the most elaborate torque wrench.

By specifying nut torque, all nuts of the same size are assumed to give the same bolt tension. However, the friction between the nut and the abutment, and the friction between the threads of the bolt and the nut are so variable that the bolt tension will vary over wide limits. Fig. 12 shows that bolt tension may vary as much as ten to one under extreme conditions of lubrication as from degreased steel to copper plate as a lubricant. The variation in practice, however, will rarely be more than three to one or less than two to one. This is, of course, still so great as to condemn nut torque measurement as a specification for so critical an operation as preloading severely stressed bolts. It would mean much to industry if means can be found for reducing the variability of nut friction to tolerable limits and, thus admit the use of the torque wrench as an accurate tool.

A recent innovation in nut tightening technique is the use of the nut as a rough micrometer. The nut is first tightened sufficiently to seat firmly all contacting surfaces, after which it is fully released and retightened "finger tight" which is taken as zero setting. From this setting, the nut is turned through a specified angle to produce the desired bolt or stud tension. The results from this procedure are more accurate than from nut torque specifications, but it is very cumbersome. The angle through which the nut should be turned must be varied with bolt size, bolt length, number of threads per inch, kind of bolted material, area of bolted material, and other variables. The technique is probably justified in spite of the complications in original assembly and the greater complications in servicing shops because of the greater assurance of tightness that is obtainable. It serves, also, to emphasize the desperate need for nut-tightening accuracy.

In the cases where both ends of the bolt are accessible, good accuracy may be assured by measuring bolt elongation by micrometer or by indicator. This practice is followed by most aircraft-engine builders for accessible bolts with satisfactory results. At least one manufacturer tightens connecting-rod bolts to a micrometer extension so great as to exceed the yield point for the bolts lying near the low limit of hardness. No trouble has been experienced since this practice was adopted.

Utilizing Spring Members

It is probable that in many instances spring members such as are shown in Fig. 10 can be profitably used. This would add to the bolt elasticity, thus providing (1) greater safety against loss of tension by embedding, corrosion, yielding, and so on, and (2) greater safety by reducing the stress range experienced by the bolt. The deflection of the spring member could probably be used as a measure of the bolt tension with reasonable accuracy since the deflection

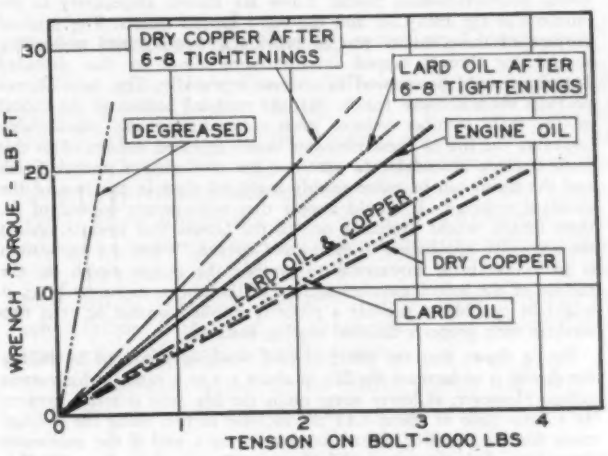


Fig. 12 – Effect of lubricants on bolt tension

could be relatively great and, therefore, easily measured. It would be necessary to design these spring members to support the desired bolt tension within the elastic range of the spring and to avoid clamping them solidly against the abutments. They could be made in the form of disc springs or as beams depending upon the available space and only one size would be required for each bolt size.

■ Conclusions

1. The strength of dynamically loaded, highly stressed bolts and studs is determined by the man with the wrench. Other considerations, such as design, material, and processing, are of relatively minor importance.

2. The elasticity of bolts and studs should be as great as possible.

3. The bolted members should be as rigid as possible.

4. Loss of dimension of the bolted members is particularly hazardous to the strength of short bolts and studs.

5. No adequate means are available for determining nut tightness except when the elongation of the bolt can be measured.

DISCUSSION

Suggestions for Control Of Initial Bolt Tension

—R. R. Moore
Naval Aircraft Factory

The setting-up tightness or initial tension in such fastenings as bolts, studs, and capscrews, is of critical importance in all applications where these elements may be subjected to repeated stresses. Its importance is particularly recognized in aircraft-engine construction and maintenance in adjusting connecting-rod bolts and cylinder hold-down studs, for the reasons so thoroughly covered in Mr. Almen's paper. The problem of obtaining sufficient tightness in actual practice is difficult and fraught with danger, as witnessed by the numerous cases of twisted-off bolts. The introduction of the torque wrench was an attempt to control initial bolt tension, but its shortcomings due to the variations in fit, finish of internal and external threads, and finish of nut face and bolted member are recognized.

It is suggested that condition of internal thread finish, either in nuts or tapped holes, might be controlled by the use of inserts that can be readily prepared with a smooth uniform bearing surface to receive the male thread. One type already on the market is made from a specially shaped wire coiled to fit various thread sizes and is inserted in the hole. These inserts are made in two types, one from a diamond-shaped wire to receive the standard v-thread and the other in a wire shaped to a "v" with a rounded back which receives a special rounded-bottom thread. These are known respectively in the industry as the Heli-Coil and the Aero-Thread inserts. The original purpose of these inserts was to provide a wearing and nongalling surface for threads tapped in aluminum alloys so that threaded fastenings could be screwed in and out repeatedly. The Aero-Thread provides the additional feature that the rounded bottom of the thread on the male member reduces stress concentration and considerably improves the life of these elements under repeated stresses. Also this form is more amendable to processes for producing a smooth finish and the finish can be more readily inspected than in the case of the standard v-thread. It would appear that with proper control of fit, these inserts would overcome one of the factors that operate against the successful application of the torque wrench. When the application is such that it is impractical to provide the proper finish on the surface of the bolted member or face of the nut or capscrew head, it might be possible to provide a properly finished washer or even two washers with properly finished mating surfaces.

Fig. 4 shows that the effect of cold working produced by rolling the thread is to increase the life in about a 2 to 1 ratio, at high stress ratios. However, at lower stress ratios the life ratio is much greater. At a stress ratio of about 0.12 the increase in life, using the average curve for the unrolled thread, is about 4 to 1 and if the minimum curve is used the increase in life due to rolling is about 25 to 1. It is to be noted that the stress ratio endurance curves diverge as the life is extended. The rolled-thread curve is rapidly becoming parallel to the life axis, indicating that at a stress ratio of about 0.10 a very long life (about 100 million) is to be expected. The unrolled thread curve is still going down at a rapid rate and for a life of 100 million cycles the fluctuating stress will have to be practically zero. Obviously rolling the thread is more effective in prolonging the life of low-stressed bolts (long life) than it is for high-stressed (short life) bolts.

Until some method of controlling initial bolt tension is developed to the point where it is a dependable method of reducing the range of fluctuating dynamic stresses, it is expedient to employ the means of reducing service failures that are immediately available. In addition to the rolled thread, the rounded-thread-root employed in the Aero-Thread system also considerably increases resistance to fatigue, and studs of this type have given satisfactory service in aircraft engines. Fatigue tests made by the writer on studs of this type show that the repeated load sustained by a 5/16 stud, for extremely long life, can be doubled by substituting an Aero-Thread stud for a standard national coarse thread stud. Improvements of this type have advantages over the bolt tensioning methods in that they are (a) permanent "built in improvements"; (b) not influenced by lack of knowledge of actual stresses developed in service which robs tension loading of much of its effectiveness; (c) not subject to variations in practice during installation or during overhaul or maintenance performed under difficult working conditions; (d) do not require special additional tools; (e) do not require special instructions or limits for each size.

■ ■ ■

What Do Fleet Operators Want in Post-War Trucks?

continued from p. 150

fancy treatment of the units, so that restrictions to its use should not be missed.

■ Trucks and Public Relations

Before drawing this paper to conclusion, it might be appropriate to say a few words at this time on "trucks and public relations." As we raise our sights to new horizons for future truck operations, we cannot ignore the importance of greater public acceptance to the truck on the streets and highways. The first objectionable item from this standpoint is noise; and I am not referring to the vehicle signal system.

I think it would do well for the designers to consider methods for reducing the noises from exhaust, gears, wheels with tires, and so forth, that are so prevalent on the present-day truck. May we not ask ourselves the question, "Are not some of these negative conditions as attached to the trucking industry one of the underlying causes of closed routes, no Sunday operation, and so on?" No doubt the general public reactions to the inconsiderate tion of the trucking industry are natural and growing. might cite the introduction of the new type of street cars which have made their appearance in many of our cities as a good example of what can be done by design in reducing noise plus, of course, greater speed, safety, and comfort, and which have certainly gained public acceptance and better public relationship for the traction operation I believe that the matter warrants real consideration by the entire trucking industry and a challenge that will surely be accepted and met.

Note: An appendix to Mr. Laurie's paper giving extensive quotations from the questionnaires used in its production is available in mimeograph form, copies of which will be sent on request.

AIR CARGO PLANES from the ENGINEERING VIEWPOINT

by JAMES B. KENDRICK

Chief of Aerodynamics and Flight Test
Lockheed Aircraft Corp., Factory A

THE use of air cargo planes is a new science which has mushroomed since Pearl Harbor. There was no time for a modest trial period – the steady southward advance of the Japanese invaders called for heroic defense, which required that supplies be sent by air to far-flung bases which were inaccessible to American ships. When the need for cargo planes became apparent, dozens of airline transports were diverted to war use, given a coat of camouflage paint overnight, and started for the South Pacific. Since then, this branch of the service has grown into an efficient, world-girdling organization, which has often been praised for its important role in the winning of the war. When the experiences of these air cargo operators are collected, they will provide new and valuable information for the improvement of domestic airlines and for the establishment of a real air cargo service in America.

The statements of many previous writers serve to indicate the magnitude of future air commerce, and its effect on the nation and world of tomorrow. It is not my purpose to amplify nor detract from these predictions, but rather to show a few ways in which improved engineering, and better cooperation between engineering and operations personnel, may serve to reduce the air cargo rates, thus attracting a greater volume of potential business, and increasing the utility of the cargo plane as a means of transportation.

This investigation of air cargo planes consists of the following general topics:

1. Some fundamentals of cargo plane design and economics, to permit an evaluation of the relative importance of various factors.

2. The engineering aspects of some of the problems of operation that must be considered by the maintenance and operations personnel.

Air and Surface Transportation Compared

What about the inherent differences between air and surface transportation? It has been stated that, whereas the efficiency factor of the modern airplane is 20:1, as indicated by its lift:drag ratio, corresponding factors for the railway train and steamship are 100:1 and 500:1 respectively. These differences affect the fuel consumption, primarily, or from the operator's standpoint the cost per ton-mile, chargeable to fuel only. This is a small part of the total cost of operation, but nevertheless must be offset by other factors before the air cargo plane can compete economically with surface transportation.

The following aerodynamic considerations will serve to show that the inherent difference in fuel cost is sufficiently

[This paper was presented at the SAE West Coast Transportation and Maintenance Meeting, San Francisco, Aug. 20, 1943.]

MR. KENDRICK here shows ways in which improved engineering and better cooperation between engineering and operations personnel may serve to reduce the air cargo rates.

This investigation of air cargo planes consists of the following general topics:

1. Some fundamentals of cargo plane design and economics, to permit an evaluation of the relative importance of various factors.

2. The engineering aspects of some of the problems of operation that must be considered by the maintenance and operations personnel.

The first step toward providing good service at low cost will be accomplished automatically by the establishment of regular cargo service. The next step is to bring about greater cooperation between the operators and engineers, to inoculate the engineers with the importance of convenience in handling and maintenance. Finally, improvements in aerodynamic refinement and structural weight, as well as increased airplane size to meet the needs, will result in further reduction in ton-mile cost of operation.

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small that the airplane's advantage of speed and flexibility of service may be expected to compensate partly for its reduced efficiency. It is doubtful whether the total ton-mile cost of air transportation may ever be reduced as low as surface methods. However, constant progress may be made in this direction by the use of improved equipment, by careful attention to more efficient methods of operation, and by the greater volume of business which will naturally accompany regular service, when it is established.

As a means to understand better the engineering aspects of air cargo plane operation, a few simple formulas for estimating the cruising speed, horsepower required, and fuel costs will aid in a comparison of various design factors. These formulas are developed for the optimum cruising speed to result in maximum range, based on the assumption that the cargo plane will operate almost entirely at these conditions for best economy. Many other factors enter into the cargo plane design, but these represent the most important from an aerodynamic standpoint.

$$\text{Cruising Speed, mph} = 19.8 \frac{1}{(C_{DP})^{\frac{1}{4}} (e\pi R)^{\frac{1}{4}}} \left(\frac{W}{S}\right)^{\frac{1}{2}}$$

$$\text{Thrust Horsepower} = 0.105 \frac{(C_{DP})^{\frac{1}{4}}}{(e\pi R)^{\frac{3}{4}}} \left(\frac{W}{S}\right)^{\frac{1}{2}} W$$

$$\text{Miles per Gallon} = \frac{1100\ (e\pi R)^{\frac{1}{2}}}{(C_{DP})^{\frac{1}{2}}} \frac{\eta}{SFC} \frac{1}{W}$$

$$\text{Fuel Cost, ¢ per ton-mile} = \frac{1.78}{k} \frac{¢}{\text{gal}} \frac{(C_{DP})^{\frac{1}{2}}}{(e\pi R)^{\frac{1}{2}}} \frac{SFC}{\eta}$$

Where:

W/S = Wing loading, gross weight per wing area

W = Gross weight

eR = Effective aspect ratio

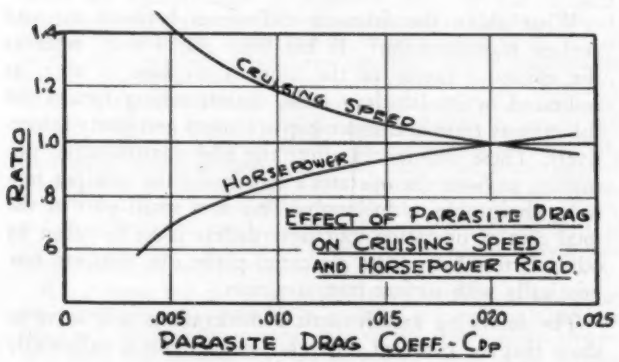
SFC = Specific fuel consumption, lb per bhp-hr

C_{DP} = Parasite drag coefficient

η = Propeller efficiency

k = Payload per gross weight

High cruising speed and low power-required are desirable characteristics, and these factors may be improved considerably over present-day standards by aerodynamic refinement. (See Fig. 1.)



■ Fig. 1 – Effect of parasite drag on cruising speed and horsepower required

The wing loading has an important effect upon cruising speed and power-required, as well as upon the take-off and landing speed of the airplane; hence, this factor must be carefully considered in the design of the airplane.

Airplane weight is a factor of prime importance in airline operation, since excess weight means reduced pay load. An extra pound, just one pound, of airplane weight-empty will amount to $100 to $200 or more profit lost to the operator over the normal life of the cargo plane. Not only must the original design be conservative of weight in order to increase the allowable pay load, but also the maintenance and service groups must avoid unjustifiable increases in weight which sometimes cause serious reductions in operating efficiency.

■ Wing Design

One of the major features of the efficient load-carrying airplane is a long, slender wing, which we call high aspect ratio. This is important because the best economy conditions of an airplane occur at low speeds, where the frictional drag is nominal. The wing may be considered as a planing surface which deflects a strip or cylinder of air downward, the reaction from this air equaling the weight of the airplane. It is desirable to have a large span in order to react upon a wider strip of air. This is especially true at the low air speeds for best cruising economy, where the forward speed of the airplane permits the wing to traverse a limited length of the strip in a given time. In this way high aspect ratio permits the airplane to fly efficiently at low speeds.

■ Low Parasite Drag

Another factor of great importance is to have low frictional or *parasite* drag, which is measured by the smoothness of the airplane and the freedom from poor intersections and improperly shaped bodies which cause turbulence in the airstream. Fig. 1 shows the effect of parasite drag on cruising speed and horsepower required. Present-day good designs have a parasite drag coefficient of approximately 0.025, whereas the ultimate we should expect, based on present knowledge, might be 0.005, thus permitting higher cruising speeds, with even less power than at present.

Parasite drag is a factor of equal importance to the aspect ratio in the determination of cruising speed and fuel cost. It offers great possibilities for improvement to the designer of future cargo planes. For example, a 50% reduction in the parasite drag of current airplanes would save enough fuel in five years of operation to equal the initial cost of the airplane. The artist's conception of the airliner of tomorrow as a flying wing has a sound engineering basis for in this way large reductions in parasite drag may be effected.

Fig. 2 shows the effect of gross weight and parasite drag on the mpg, for a family of airplanes having the same aspect ratio and wing loading. The gross weights vary from 100,000 to 1,000,000 lb. The effect of parasite drag on cruising speed and fuel cost per ton-mile will be noted, these factors remaining constant regardless of gross weight (although, of course, the horsepower increases as a function of weight). The magnitude of fuel cost per ton-mile is nominal, even for the high parasite drag airplanes, and the

ultimate for efficient designs approaches ½¢ per ton-mile. It is interesting to note that the fuel cost per ton-mile is independent of the size of the airplane. This may be an important factor, suggesting the use of larger numbers of airplanes of smaller size, to increase the frequency and flexibility of service.

What is the relationship between fuel costs and total cost of cargo plane operation? The foregoing analysis of the fuel costs involved shows this to be a sizeable factor in economy of operation. However, because of the fact that present cargo planes will average 1-2¢ per ton-mile fuel cost, as compared to 15¢ per ton-mile total operating cost, it is apparent that other factors are of much greater importance and need first consideration in a program to reduce air freight rates. Among these are personnel requirements, maintenance, handling time, depreciation, administrative costs, and so forth. These matters depart somewhat from normal engineering considerations, but in some cases require the combined efforts of engineering and operational endeavors to result in further reductions in costs of operation. Examples of such combined problems are as follows:

1. Reduction in ground handling time.
2. Reduction in service, maintenance, and replacements.
3. Simplification of operation.
4. Methods of loading.

■ Ground Handling Time

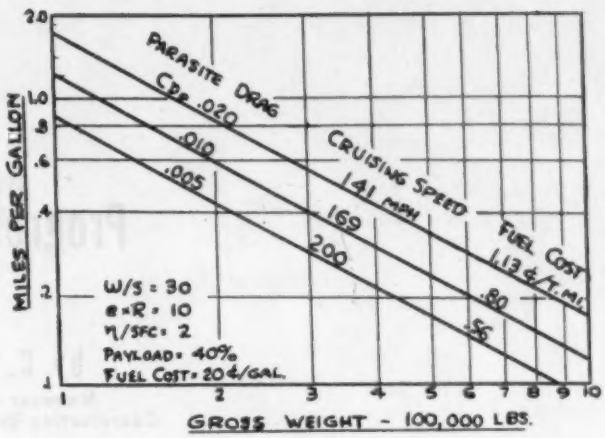
Ground handling time is an extremely important item in cargo plane operation, this factor affecting the airplane utility in the same manner as cruising speed and fuel economy. The operating schedule of the cargo plane, including taxiing time, loading and unloading of cargo, and the ground service functions required, may be minimized more easily by a reduction in ground time than by an increase in performance, especially on those airplanes of short range between stops. For example, with a distance of 100 miles between stops:

Cruising Speed, mph	200	150	110
Air Time, min	30	40	55
Ground Time, min	30	20	5
Total, min	60	60	60

the 200-mph airplane with 30-min stopovers is no better than the 110-mph airplane with only 5-min stops. Careful design may easily account for many minutes saved in loading and servicing time, to the benefit of the cargo plane's utility. Improved taxiing characteristics of the plane, and better airport design for convenient traffic handling, are likewise important factors.

■ Maintenance

Maintenance for airworthiness is different from that of surface fleet operation, primarily because of the serious consequence of failure of any one of a number of components on the airplane which might cause a forced landing or crash, whereas with a land vehicle, a similar failure might result in only a roadside stop. In general, the seriousness of failure depends upon the speed of the vehicle, in which case high-speed trucks and buses are approaching the same hazards as face the cargo airplane. The attention paid to maintenance and inspection must



■ Fig. 2 – Effect of gross weight and parasite drag on fuel consumption

therefore increase with the speed of the vehicle, both as to the frequency of inspection and the time allotted for overhaul and replacement of questionable parts.

In order to minimize the effect of this additional maintenance on ton-mile costs, great improvements are needed in airplanes to simplify the accessibility of critical parts for inspection and maintenance. Slight increases in weight may be involved, but such compromises are usually beneficial to the overall utility of the airplane. A good example of this is the use of quick-change engine installations, permitting the removal and replacement of a complete powerplant installation in one hour or less, and greatly improving the time delay involved in engine overhauls. Similar care is needed in the design of landing gear parts for quick replacement and servicing. Likewise, control systems, hydraulic systems, deicing equipment, engine and flight instruments, all of which require periodical inspection, need to be designed to favor both the time required and the thoroughness of inspection.

The reliability of aircraft powerplants is a factor deserving of the most careful attention. Modern engines are truly miracles of engineering, but they *will* sometimes fail. The weight factor usually enters into this picture to the disadvantage of the airplane engine. To safeguard against engine failures, methods are now in use for the extensive preflight testing of aircraft-engine installations, and these methods should be constantly improved to increase the aircraft-engine reliability. Likewise, other parts of the airplane should be subjected to thorough life tests for the improvement of serviceability. Landing gears, hydraulic systems, control systems, and other working parts should be given extensive laboratory and flight test to accomplish this. Other parts should be subjected to vibration test to determine best design and material for long life. Tests of this kind, together with careful inspection and maintenance during the operation of the cargo plane, will lead to reduced operating costs and service interruptions.

■ Simplicity of Operation

Simplicity of operation offers a field of advancement which deserves careful consideration by engineering and operations personnel. The number of controls, instruments,

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MOBILIZED RE

Progress Report of Coordinating Researc

by C. B. VEAL

Manager and secretary
Coordinating Research Council, Inc.

BORN during the aftermath of World War I, the nation's outstanding cooperative petroleum research organization has again enlisted in the service of the Army, Navy, and Government civilian agencies. Although both the type and the urgency of war time problems are unlike those of peace, the mechanism of the Coordinating Research Council, supported equally by the Society of Automotive Engineers and the American Petroleum Institute, has been geared to the task of solving the new problems with unprecedented speed.

More than 150 technical working groups, scores of the world's finest petroleum laboratories, and the research facilities of the Government have been coordinated on problems requested of the CRC by the Armed Services.

Intensive coordinated research in fuels and lubricants is demanded by the strenuous use of combat airplanes, military vehicles, and war vessels, wartime shipment and storage of fuels and lubricants and, shortages of certain critical petroleum products.

Because of the urgency of the problems, high-speed technical education has been required to acquaint novices with the intricacies of testing procedures and laboratory techniques.

This invigorated program, besides performing tasks of immediate necessity, is calculated to strengthen the organization so that it will be able to continue to serve the automotive and petroleum industries and the American public in the post-war era.

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THE AUTHOR: C. B. VEAL (M '12) was SAE research manager and secretary of the CFR Committee for 16 years prior to his appointment this year as manager and secretary of the Coordinating Research Council. During and after World War I, associated with Charles M. Manly, he was with the Curtiss Aeroplane & Motor Co. in a consulting capacity. Mr. Veal was head of the Machine Design Department of Purdue University for several years immediately preceding World War I, and throughout those years had an extensive consulting practice.

THESE are days when men bar the nonessentials. Especially is this true of automotive and petroleum technologists upon whose anxious minds rest the decisions on vital questions affecting the fuels, lubricants, and equipment of a supermechanized war.

A signal tribute to the service being rendered by the Coordinating Research Council in assisting in the making of such decisions is the growth in its activity during the past year. This growth has been marked in both intensity and scope. Veterans of the 22-year-old Coordinating Fuel Research have capitalized to an increasing extent upon their experience in cooperative effort. The broader fields opened up by the Coordinating Lubricants and Equipment Committees have called out an imposing body of new recruits. The War Advisory Committee has been the directing agency bringing men from all services to the Council's conference tables. Research on questions affecting the common problems of the petroleum and automotive industries has mobilized.

A permissible avocation in these strenuous days is the conception of uses to which we will put, in peace, the weapons forged in war. When the stress of war has eased, we envision our enlarged intensified cooperative effort directed not solely toward problems created by immediate emergencies, but, more constructively, toward finding ways of avoiding such problems, and of more effectively utilizing our petroleum resources in mobile land, air and water transportation.

■ General Nature of CRC War Activity

Members of the Society of Automotive Engineers form collectively one-half of the partnership sponsoring the Council, the other half being the American Petroleum Institute. The Council is a body of 12 men, directing and administering the business affairs of 200 technical working groups so classified and supervised by committee organization as to prevent duplication of programs and to ensure maximum utilization of effort. The objective of this impressive collective endeavor is the solution of problems of fuels, lubricants, and equipment common to both automotive and petroleum industries. The Council recently has been incorporated under the name "Coordinating Research

[This paper was presented at the SAE War Engineering Annual Meeting, Detroit, Mich., Jan. 14, 1944.]

RESEARCH

Council Shows Rapid Wartime Growth

Council, Inc.," thus taking a final step toward the independent entity required in dealing with outside agencies, such as the Army, Navy, and Governmental bureaus.

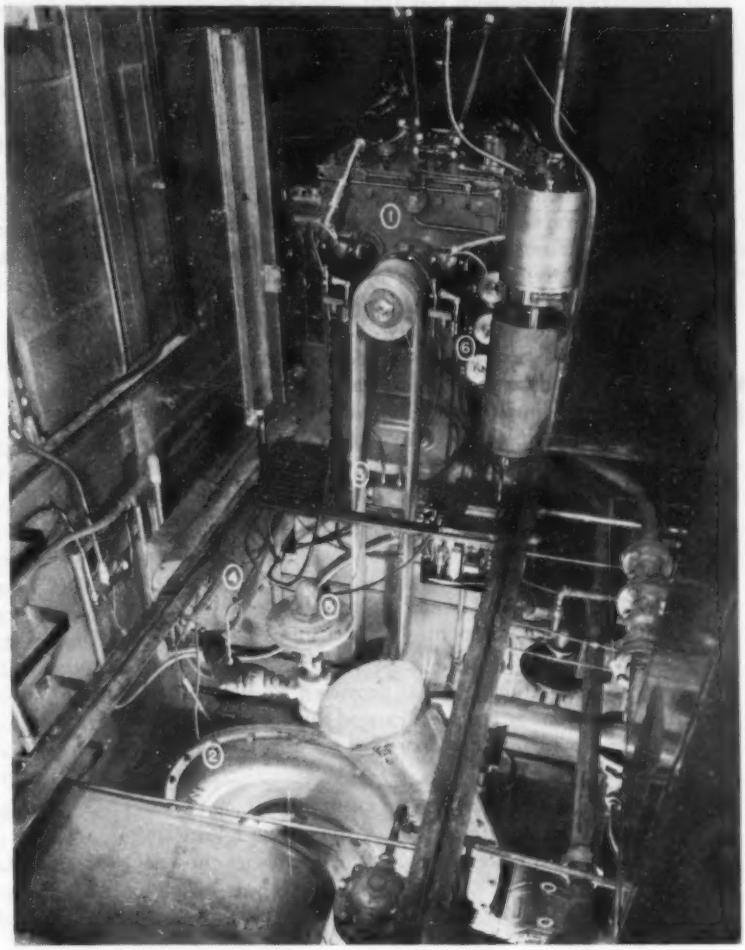
During the past year all of the working groups have concentrated on projects undertaken at the specific request of the Armed Services. Any reference to such work must be suitably guarded, because of military secrecy.

The aim of this paper is to illustrate the flow of the Council's effort into war channels by citing examples of research in each of five types of special war problems.

The types of problems are those occasioned by war's: (1) strenuous operation; (2) shipment and storage conditions; (3) shortage of critical materials; (4) simplification and standardization requirements; and (5) necessity for high speed, intensive technical education.

■ War Problems – Lubricants

This global war requires two kinds of endurance: resistance to long periods of strenuous usage, and to extremes of climate. An ambitious pre-war motorist, of the present A-card holder class, required from his car routine indi-



Cooperative Universal Engine and its Accessory Equipment

CU Engine development was started by the SAE Aviation Crankcase Oil Research Committee. The work has been continued by the CFR and the CLR Groups testing fuels and lubricants.

1. CU Engine – single-cylinder engine using current types of full-scale engine cylinders
2. Buffalo 7E blower
3. V-belt drive
4. Combustion air surge tank
5. Manifold pressure control valve
6. Cuno oil filters

vidual work-day transportation with a few weeks of intensive vacation use. During his vacation he might successfully nurse his car through several days of hot arid weather to see the wonders of our great Southwest; or with suitable precautions, make his way through the high altitudes and cold temperatures of our Rocky Mountains. The same car owner, inducted into the service, depends for his safety and the success of his military mission on long periods of unfailing, strenuous performance; he must have in the desert a gasoline that will not vapor lock, a lubricant that will not break down; in Iceland and Alaska, a fuel that will start an engine at subzero temperatures, and a grease that will permit his gears to turn.

War is a photographic enlargement; it reveals in stark crudity imperfections that are overlooked in peacetime scrutiny. The Council's mission is to anticipate and remedy in advance such imperfections.

For instance, one of the groups working under our Lubricants Research Committee collects and analyzes service information on military land vehicles for coordination with data obtained in the several laboratory test projects on motor oils. An investigation is also under way on the effect of oil additives on bearing surfaces operating under local or general high oil temperatures. Laboratory tests are being carried out, and field data collected to determine whether actual failures will occur in service. Field tests of representative types of heavy-duty engine oils are being conducted with military land vehicles at Camp Seeley, Calif.; and members of our technical civilian Army have worked through the scorching desert heat to help provide their brothers-in-arms with oils that will serve unfailingly in similar climatic conditions in Africa or Southern Italy. Similar important fuel research will be referred to later.

Dropping to the lower end of the thermometer, lubricant research is concerned with developing test conditions that will adequately predict the performance of oils under low temperatures, both in starting and under longer conditions of service. Another project in this class is the development of information on the cold-starting characteristics of gear oils and the loss of diluent in operation. Laboratory inspections and analyses have been made on 289 samples of gear oil returned from the field. Cold-room tests are being made on representative lubricants. Data on military cold-weather testing of gear oils have been made available to the Group for analysis and comment. Fifteen laboratories are participating in the development of a satisfactory method for determining the low-temperature characteristics of greases. A final project of the Coordinating Lubricants Research Committee dealing with temperature effects was a review of literature on the effect of temperature on the volume of hydraulic fluids.

Information peculiarly of interest in this war concerns the water resistance and water compatibility of greases used in amphibious equipment. This is a subject of another study.

Reflecting the need for special provision to meet severe operating conditions is a project carried out by the Coordinating Lubricants and the Coordinating Equipment Research Committees.

In developing requirements to ensure satisfactory extreme-pressure gear lubricants for Army ground vehicles, a dearth of performance data and inadequacies in test methods were revealed. To fill this lack, comprehensive field and laboratory tests of typical lubricants are being carried out. Field tests with Army equipment, at both Aberdeen Proving Ground and Camp Seeley, include high-torque, high-speed, and endurance runs.

Further, to assist the Armed Services in meeting weather operating hazards, the Coordinating Equipment Research Committee has organized a group to accumulate available information and experience on the problems of cold-weather operation sludge and its prevention. Another group has provided technical direction and observation of Army ground vehicle tests in Tioga Pass designed to throw light on the relation of load to performance at altitude.

■ War Problems – Fuels

Turning to the Coordinating Fuel Research Committee, a continuing volume of research has been devoted to knock characteristics. The improvement of knock test instrumentation has been carried out during the war emergency to meet the need for more accurate and expeditious rating of fuels. Corrected guide tables also have been prepared, specifically for knock ratings at high altitude. A comprehensive study has been conducted of spark advance, speed, and detonation relationships in liquid and air-cooled engines in military ground equipment. This is directed not to audible detonation intensity, but to the effects of borderline detonation on the engine's mechanical structure and maintenance. Extensive testing in cooperation with the equipment manufacturers has been involved, as well as full consideration of fuel availability.

The vapor-locking tendency of present and proposed military ground vehicles must be adequately investigated to forestall this difficulty in service. CRC has cooperated with Ordnance Department representatives in the planning and conduct of tests at Camp Seeley which started in July, 1943, and are now completed. Another group is making an intensive study of all available data with a view to developing refinements in technique for correlating distillation characteristics of fuels with vapor-lock and vapor handling capacity.

Turning again to low-temperature problems, to assist in establishing cold-starting requirements to be met by a single universal motor fuel for Army ground vehicles, a procedure was evolved for testing the starting and warming-up characteristics of motor fuels. The effect of fuel volatility on engine starting also was investigated in tests on 12 medium tanks employing five test fuels. In addition to the main objective, the effect on starting of throttle position, booster coil, and various priming factors was also determined.

Knocking and vapor-locking characteristics have also occupied a large part of the attention of the CFR Aviation Fuels Division. The Multicylinder Engine and Analysis Groups provide an analysis and interpretation of full-scale aircraft engine detonation results on current military fuels under military operating conditions. A wide range of aviation fuels of military interest have also been tested in a group of laboratories. A number of test methods and engine variables have been investigated to improve the reproducibility of a laboratory procedure giving results in agreement with those of the Full-Scale Analysis Group.

Among the vapor-lock projects has been a survey of aviation gasoline characteristics as related to vapor lock, with data on 30 commercial aviation gasolines. A study has been made of changes taking place in fuels during flight under a variety of conditions and their effects on the

occurrence of vapor lock. In the laboratories of the National Bureau of Standards, the Aviation Fuels Division has worked with the Army Air Forces in determining fuel evaporation loss during high-altitude flights and in studying the control of such losses both through engine design and suitable gasoline characteristics. Cooperating also with the Navy Bureau of Aeronautics, the Aviation Fuels Division has studied at the Bureau of Standards optimum fuel line size, heat transfer through fuel tank walls, and fuel pump efficiency. Aircraft manufacturers have assisted in tests on the vapor-handling capacity of carburetors. Arrangements have also been completed to investigate the extent of evaporation losses occurring during extensive long-range flight tests.

The effects of varying fuel volatility on the performance of aircraft engines have been of primary importance to the military, engine builders, fuel suppliers, and commercial aircraft operators for some time. Tests to obtain qualitative and quantitative data have been under way for the last several years, and a re-analysis of the tremendous amount of flight test data thus made available has been completed. This has constituted an accurate appraisal of the limitations of present information and the further work required to cover the lower temperature range of military operation. In cooperation with the Army Air Forces, flight studies of the effect of fuel volatility on engine performance have also been made over a wide range of carburetor air temperatures.

Cooling problems for liquid-cooled aircraft engines have assumed increasing importance with the more stringent requirements of military usage, and all phases have been studied. Fifteen potential coolant system design projects are under consideration. The physical properties of ethylene glycol and their change with use, its corrosive effects, and the effects on engine performance of concentrations of it in water mixtures, coolant temperatures and flow rate are being investigated. Still another group is considering the desirable characteristics and sources for new coolant materials.

Turning to automotive diesel fuels, CRC groups have been assisting the Ordnance Department in establishing the requirements of a single universal diesel fuel, since such a fuel is desirable to simplify the supply problem in combat areas. An important item supplementing this assignment was the establishment of the minimum cetane number requirements of diesel-powered military equipment, and both laboratory and field tests have been carried out on this subject. Cold weather, mountainous operation has been studied in Tioga Pass, and hot, dry climate requirements at Camp Seeley, Calif. Experience having indicated that a combination of high altitude and low temperature represents the most critical operating condition for diesel engines, using fuel of low cetane number, a program of cold-room tests in which throttled intake is used to simulate altitude is being carried out. Starting tests are being conducted at extremely low temperatures. Information has also been developed on the relative value of various organic compounds which may be used as starting aids.

Shipment and Storage Problems

In the second class of problems having special angles peculiar to wartime usage of automotive vehicles, storage, and shipment are involved. In peace, the time of shipment, the duration and place of storage may be selected to afford the most favorable conditions. In war, not only are these handling phases governed by necessity, they can neither be predicted, nor provided for properly.

One storage problem placed before the Coordinating Lubricants Research Committee required obtaining technical data as a basis for a pour-point requirement for SAE Grade 10 oils which will provide maximum fluidity at low temperature and eliminate the development in oils, in storage or standing in engine crankcases at fluctuating temperatures, of pour points higher than obtained on these oils by ASTM procedure. Storage experience at extremely low temperatures was found meager. A procedure for reliably predicting pour-point stability has been formulated.

A major consideration in storage is rust prevention. The Lubricants Committee was called upon to work out, for the Ordnance Department, requirements for rust preventive oil which should protect automotive gasoline or diesel engines during storage and shipment, and permit, without damage to the engine, such limited operation as may be required during shipment and storage. By investigation, the necessity of special treatment for stored engines was established, a procedure for the application of rust preventive oil was developed and field-tested, the performance of a rust preventive oil from the operating rather than the rust preventing standpoint is being studied, and laboratory tests for rust preventives are being developed. A rust preventive oil for civilian vehicles, which does not require critical materials, has also been recommended.

Storage problems of complete vehicles have been considered. The SAE Committee on Storage of Motor Vehicles, in formulating a recommendation to the Office of Defense Transportation on corrosion prevention in stored vehicles, requested assistance with regard to the lubrication of transmissions. A report was submitted to the SAE Committee and, as an outgrowth, suggestions were offered for precautions to be taken with regard to gear oils for Army vehicles during storage. Recommendations have been made, on the basis of available information, for rust inhibiting greases for stored vehicles. Suitable methods for determining storage stability of lubricating greases are also being investigated. A further contribution to the SAE Committee was an outline of good practice in preparing hydraulic brakes for storage and return to service. Similar recommendations were submitted for military vehicles.

A study of the stability requirements of motor fuel during shipment and storage was requested by the Ordnance Department. The group organized for this undertaking has met approximately once each month, and five working panels have been required to secure the necessary data. Temperatures likely to be encountered in shipment and storage, the effects of containers and coatings, laboratory methods of predicting storage stability, and antioxidant effects on storage life were some of the topics studied. Storage tests under desert conditions are in progress at Camp Seeley, Calif., on the 435 samples required to cover the full range of fuels and conditions of storage. Simultaneously, accelerated storage tests are being carried out in laboratories on samples of the fuels stored in the desert.

The Automotive Diesel Fuels Division has been working on the development of a run-out fuel which will give maximum protection against corrosion and deposits in

lines, pumps, and injector equipment. A tentative set of requirements was drafted; tests to check these requirements are nearing completion. A volunteer group has carried out oxidation stability and corrosion tests on four fuels especially selected for this program; and injector parts have been placed in storage in fuels meeting the proposed requirements to check the corrosion and gum or lacquer formation under simulated field conditions. The limitations on a suitable flushing oil to be used in diesel-engine injectors in boxed vehicles and spare engines will also be recommended.

■ Critical-Material Substitutions

A third class of problem made acute by war instigates the search for substitutes for critical materials limited in supply. The necessity for using a minimum amount of critical chemicals and heavy-oil stocks enforced revised procedure for qualification of engine oils. Accordingly, at the request of the Ordnance Department, five separate procedures were formulated. Another project had for objective conservation of oil additives and the development of new sources for heavy-duty oils to release materials needed for aviation lubricants. For this engine tests were made on eight SAE 50 oils, six types of detergent oils, and two straight mineral oils. The detergent-type oils contained 2/3 the amount of additive carried in the SAE 30 grade of each particular oil. Seventeen laboratories participated.

The substitution of ferrous for nonferrous metals in Ordnance equipment is causing certain lubrication difficulties, which a CLR group is considering.

Two reports have been submitted on problems caused by the substitution of fatty acids for fats in cup greases, the replacement of fats having become necessary to meet the volume demand. Developing satisfactory substitutes for castor oil, which must be conserved, is a joint undertaking of the CLR Hydraulic Fluids Groups and representatives of the shock absorber and hydraulic brake manufacturers organized under the Coordinating Equipment Research. The latter are particularly interested since corrosion and the effect on rubber connections and fittings are involved.

■ War-Time Simplification

Simplification and standardization, and the extent to which these may be carried without deleterious effect on equipment or performance constitute the fourth type of war problem. A summary of grease specifications and lubrication instructions now in use is being made, as a first step toward a reduction of the number of the greases required by the Air Forces and aircraft operators under various climatic conditions. A complete summary and analysis of current Government grease specifications has been prepared in response to a request from the military services which, at the suggestion of the Working Committee of the Army-Navy Aeronautical Board, are cooperating with the CLR Grease Advisory Group in attempting to simplify the requirements. A group is endeavoring to simplify shock absorber fluid requirements. A review of available data on the numerous hydraulic jack fluids employed by 15 manufacturers of hydraulic jacks supplied to the military forces clearly indicated that, with one exception, the jacks could be operated satisfactorily on one grade of mineral oil.

■ Technical Education for War

A fifth type of problem made urgent by war is the necessity of speeding up and spreading technical education. Because of the vastness of the engineering job to be done, a large number of novices must be introduced into fields of operation, maintenance, and testing. Partaking of the nature both of standardization and education continued and concentrated attention has been given during the last year to the test procedures formulated by the CRC. These procedures have been carefully revised both technically and editorially. They have been amplified, clarified, and, as far as possible, brought to a standardized form. There are 24 such test procedures, copies of which may be obtained from the CRC headquarters. As part of the editorial activity must be mentioned the preparation of the 1944 edition of the CFR Handbook, which embodies test procedures applicable to fuels, a brief indication of the development of the CFR, and information resulting from some of its research. Indicating the growth of the CFR, the new edition will be almost twice the size of the first handbook issued in 1941.

A project carried out continuously by the CRC which, with the other major benefits, is largely educational, is the cooperative testing of fuel samples in laboratories throughout the country. For aviation and motor fuels, the quality tested is octane number; for diesel fuels, cetane number. In all cases, certain laboratories make up the group which tests identical samples each month, using specified CRC procedures, and periodically all owners of test equipment are permitted to participate. The National Bureau of Standards receives and analyzes the test results and issues reports covering the data. These exchange tests set up standards of precision in testing, indicate to each individual laboratory the extent to which its results conform to the average, and furnish the basis for a statistical study of the reproducibility of the procedures and the trends in fuel characteristics.

Two more specific instances of the CRC might be cited. The first is a set of instructions for the lubrication of wheel bearings, drawn up at the request of the Ordnance Department. The other is concerned with laboratory knock testing of aviation fuels. In accordance with the request of the Office of Petroleum Administration for War, supported by the Army and Navy, arrangements have been made for the training of operators of the aviation super-charge knock test unit as experience has indicated that it is extremely difficult to secure reliable results with this equipment without thorough instruction and training. Four regional forums for engine operators have also been held, the total attendance being 200. Information developed in individual laboratories on methods of operating and maintaining these units is circulated to all owners of the test equipment through the CRC headquarters. This continuing educational program has been found so beneficial that a similar undertaking has been organized for laboratories using the CFR Research and Motor Methods.

These items of the work of the CRC cited above indicate its scope. Some figures taken at random will indicate the degree of its activity. Of five procedures for engine oil qualifications, 975 copies have been requested and distributed, and 6850 copies of the report forms, involving 96,000 sheets. Eighty thousand copies of two report forms for use with an aviation fuel knock test procedure were sold during one month. To users of this procedure, more than 85

pieces of informational literature were sent during four months. The mailing list now includes 95 names and requires 275 sets of material for each periodic mailing. One fuels group prepared a total of more than 100,000 gal in special batches of test fuels. The CRC embraces more than 175 active projects, which have occasioned 200 meetings during the past year; the reproduction of minutes and reports alone required the cutting of 6000 stencils.

Future of CRC

The current great activity of the CRC is the fruit of the past cooperative research of the automotive and petroleum industries; it bears within it the seeds of its future. The CRC aptly exemplifies the adage that the child is father to the man. Its illustrious progenitor, the Cooperative Fuel Research, was conceived in the first World War, when the results of cooperative effort – the Liberty engine and the Class B Military motor truck – inspired Henry L. Doherty and Harry L. Horning with the possibilities of such common endeavor extended to times of peace. Organized for war and its aftermath, the CFR adapted itself to peacetime industry, and rooted in the soil of serviceability, has been prolific in fruits of usefulness. Similarly, the CRC, in part at least, has owed its formulation to war. Similarly, provided that three important conditions are met, its peacetime progress will duplicate, or rather better, that of the CFR, being based on the same historical logic, and in addition, on the experience of the CFR.

First of the conditions to be met is that the CRC will continue to honor two fundamental principles governing the CFR flexibility and voluntary effort. This organization never has been, and never should become, static. Organization and procedure, as servants, not masters of cooperative endeavor, must remain adaptable to changing conditions and to new issues. CRC must draw only on voluntary service; only when men give freely do they give their best, and only in the voluntary cooperation of industry with government are the principles of democracy truly observed. All our important war activity has seen no single instance of Government levy on industry. The Government has stated its needs; industry has made available the free advice of its best qualified technicians, the facilities of its finely equipped laboratories, and the services of their personnel. Lines of company or industry demarcation have ceased to exist; all have pooled the required resources in a vast mobilization of research. This mobilization will serve peace as well as war, provided that the second and third conditions before mentioned are fulfilled.

The second condition is that need will exist for such mobilized research; CRC, like CFR, never will prolong its existence beyond its usefulness. As long as problems exist in the adaptation of fuels and lubricants to equipment, powered by internal-combustion engines and such equipment to fuels and lubricants, CRC has a vital mission. The extent and urgency of such problems are indicated in any collection of SAE or API meeting papers; the program of the present meeting bristles with challenges to the automotive and petroleum technician. The urgency and the difficulties of these problems will be increased because of the depleted state of petroleum resources due to war drains; the war-flattened national pocketbook and the extravagantly optimistic anticipations of the public concerning post-war fuels and equipment, founded on irresponsible and unwise lay conclusions from generally conservative and well-reasoned technical expositions.

The third condition for a successful future for CRC is a continued selection of projects peculiarly suited to its resources. Such problems have two outstanding characteristics: (1) they must require the united effort of large sections of the automotive or petroleum industry or of both of them; (2) they must result in findings capable of widespread and repeated application. Illustrations are test procedures, such as octane-number determination, or requirements for classes of material or equipment, as for rust-preventive engine oils for storage purposes.

The CFR through all its years has perhaps done more than its share of trouble-shooting. Called in after an emergency has arisen, it has prescribed practical remedies such as could be evolved and applied quickly. The undertaking of long-range problems in time to avoid anticipated trouble, that is, preventive research is to be desired and striven for in the future; but we are fearful that, human nature being what it is, all too frequently no request will be made to CRC until really serious trouble exists. Admittedly, the results have been more appreciated because they have met emergencies by curing existing troubles, but industry and interested Government agencies would profit even more by supporting programs anticipating avoidable trouble.

No dissertation on fuel and lubricants research and its relationship to automotive equipment is complete without some prognostication of possible post-war developments, and among these the most discussed presently is high-octane fuels. The effective utilization of such fuels brings the engine designer many knotty problems, such as those growing out of the peak combustion pressures produced by these fuels, practically double those due to current motor gasoline. Such pressures enormously increase engine stresses, affecting the design, construction, and life of engine parts, bearings, pistons, piston rings, crankshafts, spark plugs, and other vital engine elements. A small high-octane prototype aircraft engine for automotive use, built with endurance to withstand such loads would cost, as does the present aircraft engine, probably ten times as much per pound to build as do current automotive mass-production engines suitable for fuels of the prevalent octane number. Such an engine would be intolerably noisy, and even if all structural and maintenance problems were solved, the knowledge and experience are not available to ensure the smooth quietness which the American motorist demands. Fuels of much higher octane number appear to offer few immediate compensating advantages for the arduous efforts required to utilize them. The public will be interested in lighter, smaller engines, but because of the higher stresses, it is doubtful that the high-octane automotive engines will be of less weight, unless novel materials of greater specific strength become available.

From the viewpoint of the fuel producer, also, high-octane fuels for general automotive use offer no enticing picture. The present high-octane values in aviation fuels are produced by blending with special chemical materials which are made available for war emergency uses at costs so high as to be prohibitive for ordinary peacetime motor gasolines. In addition, the transformation of molecules necessary for the production of high-octane fuels greatly reduces the yield per barrel of crude oil. Should the engine designer succeed in development of a powerplant for average automotive use capable of satisfactory operation on

high-octane fuel, he would force the petroleum refiner to adopt expensive refining methods entailing uneconomical use of the available crude.

A more cheerful prospect for post-war improvement in automotive performance is revealed if it is sought in other directions. Another discusser of this subject has pointed out that although the present passenger car obtains only 20 or 30 ton-miles per gal, a bus obtains from 75 to 100 ton-miles per gal, and this with little sacrifice of serviceable speed and performance as compared with the passenger car. The reason lies in the design and use of the bus transmission, making it possible for its engine to operate more nearly at its optimum loading over a greater percentage of its operating time than does the passenger-car engine. These considerations indicate that the automotive designer of the near future would profit by turning from the fallacious glamor of high-octane fuels toward the more solid attractions of a practicable, light-weight, automatic, preferably continuously variable transmission.

For the fuel refiner, the objective of research should be to determine the type of fuel that, taking into consideration availability, refining costs, and first and maintenance cost of equipment, will result for the ultimate user, not in the greatest number of miles per gallon of gasoline, but rather in the maximum ton-miles per barrel of crude oil, or putting it more simply the optimum return in satisfactory transportation per dollar expended.

What of the future in lubricants? Apparently, the advertising and publicity specialists have failed to develop the glamorous lingo for lubricants that has been so persuasively and pervadingly applied to high-octane fuel consequently the public is not lubrication conscious. Actually, the public has more to look forward to immediately in improvement of lubricants than of motor fuels, and if the rapid growth in the work of the Lubricants Research Committee is a criterion we may expect extensive CRC activity in this field throughout the next several years. History repeats itself. It was not until individual technologists and their companies had a considerable backlog of fundamental experience and knowledge on fuels that cooperative work was possible. Perhaps the present activity in lubricants research can be traced not alone to the war requirements but quite as much to a sufficient reduction in the factor of ignorance among the technologists of the two industries to make them unashamed to discuss questions of improvement in open committee meetings. That such knowledge does exist is evidenced by the rapid strides and outstanding success in meeting military demands. As more came to be known about fuels, the experts grew more willing to contribute to joint research; just so it will be with lubricants.

For fuel the CRC will be repeating, indeed, the history of the CFR, whose first project was this very problem of conservation of natural resources now confronting us, securing the maximum satisfactory mileage from a barrel of crude oil, but with this difference: the initial CFR project was on volatility rather than knocking characteristics. It is anticipated that the CRC in dealing with fuels, lubricants, and equipment, backed by the logic and experience of history, will more than match its past in a serviceable and profitable future.

Air Cargo Planes from the Engineering Viewpoint

continued from p. 161

instructions, and procedures has become appallingly complicated during the past few years, with the result that highly trained personnel are required to operate such aircraft safely and efficiently. In many cases, extra crew members are needed to relieve the pilot of attention to so many details. Through the use of automatic controls for many of the functions, rather than manual, and the use of warning lights instead of indicators, the task of piloting a large airplane may be simplified to the extent that the number of operators, and the skill required, may be reduced. The number of pilots and copilots necessary to operate a fleet of airplanes capable of handling a certain volume of cargo is in sharp contrast to the small number of operators on a railway train. Such a comparison tends to indicate the need for larger airplanes, with fewer numbers of operators, as a logical step toward reduced costs. This factor must, however, be balanced against the need for frequency of operation and flexibility, which dictate the use of smaller units in greater quantity.

■ Load Distribution

More attention must be paid to balance and load distribution on the cargo plane than is generally needed on a land vehicle. Not only must the load be controlled to avoid excessive gross weight from the safety standpoint, but the distribution must be such as to avoid exceeding the center-of-gravity-limitation characteristics of the airplane, which limitation is necessary to ensure satisfactory control and stability in flight. Furthermore, the air bumps encountered in a flight require careful tie-down of all cargo to avoid shifting and consequent damage to cargo or aircraft structure. Great progress has been made along these lines by the extensive use of air cargo in military operation, where the variety of size and shape of cargo packages is quite extreme. To improve this, more or less standardized packaging methods have been established, as well as methods of tie-down to make maximum use of the limited cargo storage volume that exists in most airplanes.

Because of the variety of types and sizes of cargo planes required to give adequate service, the maintenance and operation of the various classes will provide many new and diverse problems to the operating personnel, yet the problem common to all will be to provide good service at low cost. The first step toward reduced costs will be accomplished automatically by the establishment of regular air cargo service. The next step is to bring about greater cooperation between the operators and engineers, to inoculate the engineers with the importance of convenience in handling and maintenance. And finally, improvements in aerodynamic refinement and structural weight, as well as increased airplane size to meet the needs, will result in further reduction in ton-mile costs of operation.

Effect of WARTIME DEVELOPMENTS on FUTURE STEELS

by W. P. EDDY, JR.*
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THE war has intensified and accelerated certain types of scientific work, particularly those which help to translate fundamental knowledge into usable tools for engineers. Steels are participating in this intensification, and engineering practices are being improved thereby.

An important aid in these accelerated studies has been the ignoring of normal peacetime competitive barriers, with resultant free interchange of information.[1]

Some of the more important wartime developments pertaining to steels have been made along the following lines:

A. Fatigue endurance.
B. Heat-treatment.
C. Castings.
D. Welding.
E. Alloy evaluation.
F. Hardenability.
G. Special addition agents.
H. NE steels.

Most of these developments were under way before the war began, but wartime progress in all these phases of steel metallurgy has been so rapid as to justify classing them as wartime developments.

Many other advances in fabrication of steels have also been made, but, since they are not strictly metallurgical in nature, they are not included here.

This paper will briefly summarize and discuss the above developments, primarily from the standpoint of the engineer. The very fact that this is a review of some metallurgical accomplishments signifies that it will not be instructive to metallurgists, most of whom have contributed to and are well aware of these advances. Predictions (guesses) will be made as to how our future practices with respect to steels will be affected. We must remember in predicting, however, that the war is far from finished, that developments which we discuss are not yet complete, and that, therefore, the influence of each development on post-war steels will depend upon (a) future progress made during the war, (b) status of each development during the period in which we return to what we shall then call normal economic conditions, and (c) the nature of those post-war "normal" conditions.

The reader should have no trouble in distinguishing between fact and fancy. Statements concerning past and present situations are, unless qualified, believed to be fact. Discussions of future events are, of course, merely conjecture, and should be classed as one man's present opinions.

■ Fatigue Endurance

For a long time, many of us have naively made stress calculations on the basis of published fatigue endurance limits based on rotating-beam tests of smooth and uniform bars, and even on the basis of arbitrary percentages of ultimate tensile strengths. We have saved face by picking large factors of safety out of the air, and tacking them on our formulas. Engineers can't be blamed for this – they

THE needs of our war machines have made it necessary to accelerate many old projects and inaugurate some new ones in the field of steel metallurgy.

Mr. Eddy discusses these developments and gives his predictions of how future practices with respect to steels will be affected by these metallurgical accomplishments.

This discussion of present accomplishments and future trends is made by Mr. Eddy along the following lines: fatigue endurance, heat-treatment, castings, welding, alloy evaluation, hardenability as a primary requirement, special addition agents, and NE steels.

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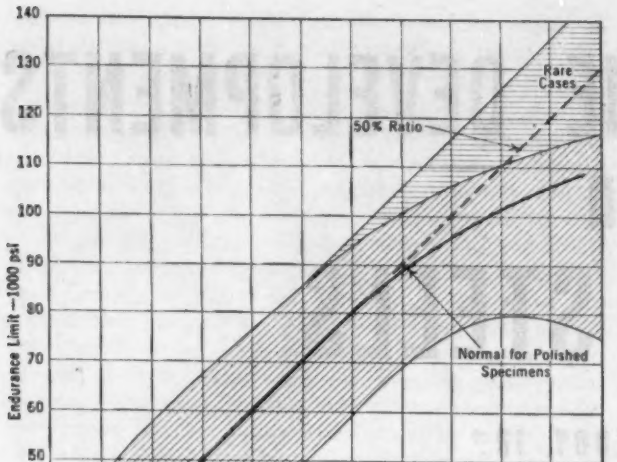
* Paper written while author was with GMC Truck & Coach Division, General Motors Corp.

[1] See *Metals and Alloys*, Vol. 18, October, 1943, pp. 764-767: "The Bank of Metallurgical Engineering 'Know How'," by Zay Jeffries.



■ Fig. 1 – Diagram indicating general behavior in fatigue of steel of polished, notched, or corroding specimens, in relation to tensile strength – applicable to ordinary corrodible steels

have been doing the best they could with the scanty information on behavior of metals furnished to them by metallurgists and mechanical researchers. As a result, we must rely on performance, in simulated or actual service, for our only reliable guidance.

However, some light is beginning to glimmer through the fog. Certain trends and developments indicate progress is being made as result of long and intelligent studies of both fundamentals and specific applications.

It has been found, in certain parts free or practically free from stress concentrators, that fatigue endurance can be improved by increasing hardness way beyond limits previously thought practicable because of toughness limitations. In order to do this, at least one of two conceptions had to be revised in each case:

1. *Toughness Is Essential* – It might be difficult to call to mind a part which has been rejected solely because it was too tough. By negative inference, we have concluded that toughness is necessary in nearly every part. It is nevertheless true that some parts need little or no toughness in the steel. Whenever this is recognized, higher elastic limit and ultimate strength, at least near surfaces, and greater fatigue endurance may be utilized with resultant improved performance or weight reduction.

2. *Parts Must Be Hardened Throughout* – We formerly specified (and still do, too frequently) high-alloy steels for highly stressed parts, hardened them throughout, and tempered them to as high hardnesses as were consistent with required toughnesses. If toughness really is required, it is often possible, particularly if the part is to be subjected primarily to torsional or bending stresses, to use lower alloy or carbon steel, so as to retain a tough, shock-resisting interior, and to harden only the outer highly stressed portions of the part to higher hardness than would be possible with deep-hardening steel, thus improving fatigue life at no sacrifice of overall toughness. Contrary to an idea held rather widely, the highest hardenability steel available may not be the best (as has been repeatedly demonstrated in many production automotive and other applications, notably by Ford, Chrysler, and Timken-Detroit Axle).

However, all parts cannot be handled in this way. Some really must be hardened throughout. Unless range of repeated stress is reduced thereby, increasing hardness and strength beyond certain limits *throughout* such parts of irregular section may actually lower fatigue endurance due to notch effect. As indicated in Fig. 1,[2] through-hardened parts having stress-raisers (or subject to corrosion) can sometimes be improved in fatigue endurance by going in the opposite direction. First, the designer should try to remove sources of stress concentration. In cases where that has been impossible, it has frequently been found helpful to lower hardness of the part, in order to permit some stress equalization through plastic flow. In this connection, greater knowledge concerning relative notch sensitivity of materials is needed.

Another means of improving fatigue resistance is to improve surface finish. This is merely one way of eliminating stress-raisers, and represents action which can be taken by engineer and production man. The value of smooth surfaces has long been appreciated, but cost considerations often prevent adoption except in the more efficient utilizations of materials, such as in many aircraft and a few automotive parts.

It has been asserted by Almen[3] that fatigue failures occur only in tension. To increase fatigue endurance on this basis, initial compressive stress may be imparted to outer portions of parts to be highly and repeatedly stressed in tension. Such stresses can be introduced by shot peening, rolling, cold-drawing, nitriding, some other types of case-hardening, and so on. Use of one of these processes, especially shot peening, may also eliminate necessity for expensive finishing operations, some of which, incidentally, may be imparting to surfaces of parts tensile stresses which actually tend to offset intended benefits of high surface finish. Also, a roughened surface, such as produced by shot peening, may in some cases improve performance because of its better retention of lubricant than that of a smooth surface.

Some future effects of these recent developments in improving fatigue endurance may be:

1. Reduced alloy content in steels for many heat-treated parts in which deep hardening is found unnecessary or undesirable.

2. More intense fight by designers against stress concentrators in highly stressed parts.

3. Great increase in use of shot peening and other applications of compressive stress to surfaces, and lowered cost of finishing many highly stressed parts.

4. Further study of notch sensitivity with relation to composition and structure, on which more precise data are needed.

■ Heat-Treatment

Determination and study of transformation characteristics (during cooling from above the critical temperature)

[2] See "Prevention of the Failure of Metals under Repeated Stress," Battelle Memorial Institute, John Wiley & Sons, 1941, p. 78.

[3] See SAE Transactions, Vol. 51, July, 1943, pp. 248-268: "Shot Blasting to Increase Fatigue Resistance," by J. O. Almen.

of steels of various compositions have paid big dividends in revealing possibilities for improved heat-treating processes. For example, principles and rules of annealing have been postulated much more precisely than before.[4] Annealing has become, in some cases, a more complex procedure than merely heating and slow cooling. The most common cycle for precision annealing of alloy steel is to heat to above the critical temperature, hold at temperature for a short measured time, cool rapidly to a certain subcritical temperature, hold at that temperature for a specified time (usually not more than a very few hours), and cool as rapidly as desired to room temperature. Temperatures and times depend on composition, and are selected by studying transformation characteristics of the steel in question. A modern cycle-annealer is shown in Fig. 2.

Precise annealing has helped to make possible the use of steel for cartridge cases,[5] an application formerly considered impracticable. Examples of alloy-steel parts, machinability of which has been improved by cycle-annealing, are axle shafts, gears, tank-track pins and end connectors, as well as tools of many kinds.

Results of establishment of precise and best annealing cycles will be lower annealing cost, shorter annealing time, more uniform hardness and structure, and better formability or machinability (with longer tool life and smoother finishes).

In hardening and tempering, need in many war products for unusual combinations of mechanical properties, without use of high-alloy steels, has resulted in unprecedented precision control in heat-treatment. Examples are tank armor plate, in which combination of high ballistic properties and good weldability is obtained, and armor-piercing shot. Die-quenching of armor,[6] with resultant savings in straightening, is a notable contribution to the art of controlling distortion in heat-treatment.

A process which permits application of thin hard cases to ordinary carbon and alloy steels without necessity of quenching and, therefore, with little distortion is carbonitriding or dry cyaniding.[7, 8] This process has been applied to such parts as steel cylinder liners, oil pump shafts, valve parts, valve push-rod ends, harmonic balancer pins, and (with oil quenching) high-carbon steel transmission gears.

Effects of these and other advances in heat-treating techniques will be lower manufacturing costs, better and more uniform products, and ability to use less-critical and cheaper steels, both for war matériel made during the next period of peace and for industrial products.

■ Castings

Scientific study of foundry problems was started years ago by several large and progressive steel foundries. Developments have been broadened and accelerated during the war, largely because of demand for dependable cast

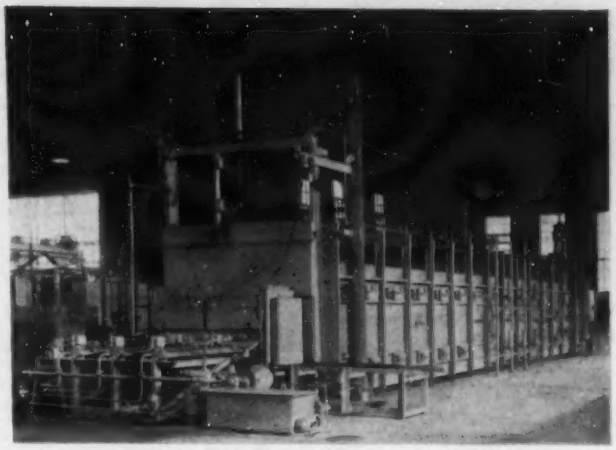
[4] See *Iron Age*, Vol. 151, June 24, 1943, pp. 44-47, 130; Vol. 152, July 1, 1943, pp. 48-54; Vol. 152, July 8, 1943, pp. 74-77; Vol. 152, July 15, 1943, pp. 70-77; Vol. 152, July 22, 1943, pp. 60-67: "The Annealing of Steel," by Peter Payson.

[5] See SAE Transactions, Vol. 51, July, 1943, pp. 229-233: "Steel Cartridge Cases," by R. B. Schenck.

[6] See *Iron Age*, Vol. 151, April 29, 1943, p. 54: "Quench-Straightening of Armor Plate."

[7] See ASM Transactions, Vol. 26, September, 1938, pp. 766-787: "The Process of Dry Cyaniding," by R. J. Cowan and J. T. Bryce.

[8] See *Heat-Treating and Forging*, Vol. 29, June, 1943, pp. 295-299: "Dry Cyaniding in the Production Heat-Treat," by L. L. Clark and C. H. Leland.



Courtesy, Surface Combustion Corp.

■ **Fig. 2 – Large, continuous, pusher-type, cycle annealing furnace – inside dimensions: 50 x 8 ft; gas fired, radiant tube burners, above and below hearth; five automatically controlled zones, operable as any number of zones up to five; second zone equipped with cold-air pipes as well as burners, permitting its use as rapid cooling zone, if desired; capacity: 3500-4000 lb per hr**

armor and other important parts of military vehicles and artillery, where properties closely comparable with those of wrought steels were demanded.

Improvements have been made by more thorough application of known principles of steel melting and deoxidation, flow of molten metals, and cooling phenomena. All types of castings – static, centrifugal, and pressure – have participated in the development.

In development and control of quality, use of radiographic techniques has expanded greatly and played an important part.

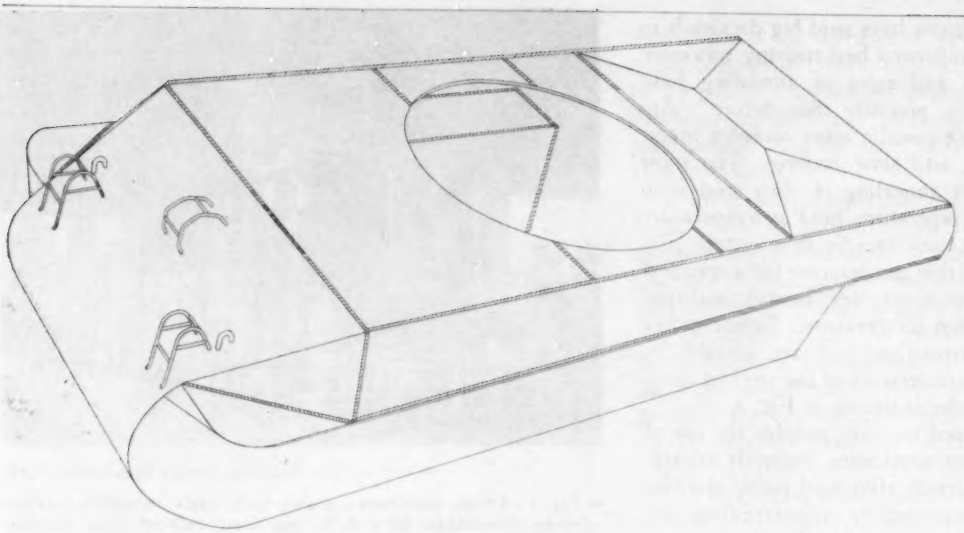
In spite of attempts made to distribute information, the less progressive foundries are falling behind the procession of quality improvement. The fact still exists that each casting is a problem in itself, so that, in addition to the general foundry problem of melting good and suitable steel, it is usually necessary for capable men to study each job painstakingly in order to maintain reputations as consistent producers of sound and dependable castings.

Effects of improvements in steel castings will be, of course, more and diversified applications, especially for high-production parts of which expense for detailed study is warranted, with resultant decreases in amounts of metal required and in fabricating costs. Greater use of X-ray and gamma-ray inspection techniques will undoubtedly be made after the war than before. It may also be found that small jobbing foundries will experience greater difficulty in competing in production of high-quality castings, because of lack of men, facilities, and money to permit detailed studies required to "work out the bugs" in production of intricate castings.

■ Welding

Great strides have been made during this war in the progress of welding from an art to a science.

Two of the outstanding advances are spot-welding and arc-welding of armor and other highly stressed parts of carbon and alloy steels previously considered unweldable, at least for primary structures (Fig. 3). Intensive industry-wide cooperative studies of basic principles, designing and build-



■ Fig. 3 – Sketch of hull of a military combat vehicle – example of all-welded armor plate

ing new types of equipment, X-ray examination, ballistic tests, careful training of personnel, close supervision, and above all the ruling idea, "it must be done," have been responsible for these developments.

■ Arc-Welding Equipment

New automatic arc-welding equipment,[9] including submerged-arc type, has been devised to weld carbon and alloy steels with a high degree of uniformity and at far faster rates than obtainable by manual welding.

In both automatic and manual arc-welding of hardenable (medium and high carbon, and alloy) steels, use of austenitic welding rod[10, 11] was first developed. Steel in those electrodes contained a total of 30% or more of alloying elements nickel, chromium, manganese, molybdenum, and others; such rod was for a time the only known type for welding plate up to 1 in. in thickness without preheating or postheating. More recent work at GMC Truck & Coach has resulted in development, for automatic welding, of a low-alloy steel electrode containing only one-tenth of the total alloy content formerly considered necessary for such welding. Welding hardenable alloy-steel parts with this low-alloy electrode has been in regular production for the past 16 months. A similarly low-alloy rod for manual welding has just recently been developed under a National Research Council project.

Several new electrodes have been developed for better welding of sheet steel (Fig. 4) in all positions, with a-c as well as d-c equipment. New materials for welding cast iron[12] and for hard-surfacing have widened the scopes of those operations. Improvements in arc-welding machines have also been noteworthy.

Gas torch applications with respect to welding have also broadened and improved. Flame-cutting of steel has so increased in accuracy that many intricate parts are now being cut so nicely that no subsequent machining is required.[13] Flame-descaling of steel plate[14] and flame-gouging of weld roots[15] are important aids to high-speed production.

Resistance-welding equipment and technique have advanced remarkably. By pulsating technique, spot-welding of heavy-gage sections has been made possible.[15] Stored-energy equipment of several types has been developed; with this, rapid spot-welding of aluminum and other metals as well as steel[16] is possible without the large power demands involved in using conventional a-c machines. Use of synchronous timing equipment has made possible extremely accurate performance, with one stroke of the spot-welder, of a sequence of operations consisting of weld, grain refine, quench, and temper.[17, 18] Electrode tip refrigeration has speeded spot-welding, stabilized tip temperature, and increased tip life.[19] New automatic devices have improved control and uniformity. Other inventions and studies have resulted in great expansion of seam welding, in flash- and spot-welding of dissimilar metals (steels to nonferrous alloys), and in spot-welding of hardenable alloy steels and even of case-hardened parts.

■ Evaluation of Weldability

Progress has been made in evaluation of weldability of steels. Many different tests have been devised and are in use to control weldability for specific applications. The Committee on Standard Tests for Welds, American Welding Society, has published, for comment and criticism, proposed tentative tests for arc weldability with intended

[9] See *Welding Journal*, Vol. 16, October, 1937, pp. 22-32: "Welding in Tanker Construction," by T. M. Jackson.

[10] See *Welding Journal*, Vol. 21, August, 1943, pp. 513-514: "Proposed Specifications for Stainless Steel Arc-Welding Electrodes for Welding Steels of High Hardenability," American Welding Society Filler Metal Specifications Committee.

[11] See *Iron Age*, Vol. 151, Jan. 21, 1943, pp. 46-50: "Working Out Techniques for Welding Armor Plate," by E. G. Biederman.

[12] See *Welding Journal*, Vol. 21, October, 1942, pp. 698-699: "Development in Cast-Iron Welding Rods and Electrodes," by R. J. Franklin.

[13] See *Welding Journal*, Vol. 20, February, 1941, pp. 91-101: "Flame Machining – Control and Applications," by G. L. Walker and W. G. Sylvester.

[14] See *Welding Journal*, Vol. 20, April, 1941, pp. 231-234: "Flame Cleaning of Structural Steel," by F. H. Dill.

[15] See "Electronic Control of Resistance Welding," by George M. Chute, McGraw-Hill Book Co., 1943, pp. 106-114.

[16] See *Welding Journal*, Vol. 22, October, 1943, pp. 484s-493s: "Stored-Energy Welding of Mild Steel," by J. M. Diebold.

[17] See *Welding Journal*, Vol. 21, April, 1942, pp. 203s-208s; May 1942, pp. 241s-244s: "Spot-Welding of N-A-X High-Tensile Steel," by W. F. Hess and C. R. Schroder.

[18] See *Welding Journal*, Vol. 21, September, 1942, pp. 421s-426s: "Progress Report on the Use of Flash-Welding as a Means of Fabricating Aircraft Structural Parts from X-4130 Steel," by R. P. Della Vedowa.

[19] See *Welding Journal*, Vol. 21, December, 1942, pp. 583s-590s: "Rrefrigerant-Cooled Spot-Welding Electrodes," by F. R. Henzel, E. L. Larsen, E. F. Holt.

road applicability.[20] Weldability has been correlated with hardenability (examples will be given later).

The expected effects of these and other welding developments on industry may be summed up in a few words: Mostly increased use of welding, especially arc and resistance, to replace more expensive assembling methods. All-welded ships,[21, 22] tanks, and many other types of war matériel will lead the way to more welded road and rail transportation equipment as well as machinery and structures of countless types.

In order to bring this about, greater attention will be paid to "design for welding." It will not be sufficient merely to make sure welding operations are correctly specified; welding engineers must be allowed to influence thought of designers from the first conception of an assembly, long before detailing is begun.

Evaluation of Alloying Elements

The effect on steel common to virtually all alloying elements – and by far the most important effect obtained from alloying steel – is now expressed as increased hardenability.

Absolute and relative effects of the several commonly used elements have been evaluated by several investigators; the method devised by Grossmann[23] is at present the most widely accepted. It is now realized that effect on hardenability of a single element is often exceeded by the cumulative effect of the same total percentage of several elements, each present in small amounts.

In past years, steels were often selected for certain parts because of one or more of the following:

1. Sales efforts of alloy manufacturers.
2. Preferences and recommendations of steel suppliers.
3. Users' customs and prejudices (familiarity with certain steel types).
4. Lack of good heat-treating equipment, and need for "foolproof" steels.
5. Willingness of steel mill to furnish "residual" alloying elements, not specified or publicly acknowledged and not directly paid for.

[20] See *Welding Journal*, Vol. 22, June, 1943, pp. 433-438: "Proposed Tentative Tests for Metal Arc Weldability of Steels."

[21] See *Welding Journal*, Vol. 20, March, 1941, pp. 158-163: "The First All-Welded Cargo Ship Built under Maritime Commission Specifications," by A. C. Leigh.

[22] See *Welding Journal*, Vol. 20, June, 1941, pp. 358-365: "All-Welded Ships," by L. W. Delhi and M. N. Maltseff.

[23] See American Institute of Mining and Metallurgical Engineers, Technical Publication No. 1437, February, 1942: "Hardenability Calculated from Chemical Composition," by M. A. Grossmann.

Table 1 – Suggested Mechanical-Property Specification Ranges

Class	Hardness			Tensile Strength, psi
	Rockwell	Brinell Diameter	Bhn	
1	B93–99	4.40–4.00	187–229	92,000–110,000
2	B96–102	4.20–3.85	207–248	100,000–118,000
3	C18–26	4.10–3.75	217–262	103,000–125,000
4	C20–28	4.00–3.70	229–269	110,000–128,000
5	C23–30	3.90–3.60	241–285	115,000–135,000
6	C25–32	3.80–3.50	255–302	121,000–143,000
7	C28–35	3.70–3.40	269–321	128,000–152,000
8	C30–38	3.60–3.35	285–331	135,000–157,000
9	C32–38	3.50–3.25	302–352	143,000–168,000
10	C35–40	3.40–3.15	321–375	152,000–180,000
11	C37–42	3.25–3.05	352–401	167,000–196,000
12	C40–45	3.15–2.95	375–429	180,000–214,000
13	C42–46	3.05–2.90	401–444	196,000–223,000
14	C45–49	2.95–2.80	429–477	214,000–247,000
15	C48–52	2.85–2.70	461–514	235,000–275,000
16	C50–54	2.75–2.65	495–534	261,000–291,000
17	C52–56	2.70–2.60	514–555	
18	C55–59	2.60–2.50	555–601	
19	C57–61	2.55–2.45	578–627	
20	C60–63			
21	C62–65			

Much debunking and self-education has already been accomplished. We realize that many low-alloy steels of equivalent carbon contents are interchangeable, and that carbon may sometimes be substituted for alloying elements, provided heat-treating characteristics are understood. Efforts are being made to express such interchangeability in tables and charts. Recommended practices have been established[24] for selection and substitution of British En steels for deep-hardening parts, on basis of usable sections for heat-treatment to various physical properties, although those recommendations may be classed by some as ultra-conservative. A group of American metallurgists (headed by F. C. Young), working as a subcommittee of SAE War Engineering Board Iron and Steel Committee, is attempting to classify steels on the basis of performance requirements; progress is slow, because one of the chief products of discussion so far is realization that certain data which are considered essential have never been compiled.

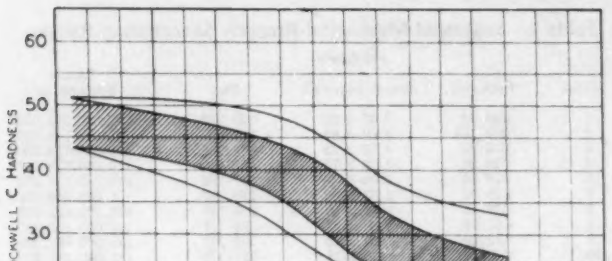
Future effects of more precise evaluation of alloying elements would appear to be:

1. *Use of Less Alloy* – Gradual reduction in total amount of alloying elements used, and possible increase in complexity of alloy steels, is an expected trend.

2. *More Intelligent Specification of Materials by Engineers* – The trend may be toward stipulation of mechanical requirements, with no mention of steel to be used, leaving steel selection to the metallurgist. As an example, for parts to be hardened and tempered, a suggested list of mechanical-property ranges is shown in Table 1. The ranges given

[24] See British Standards Institution, War Emergency British Standard Schedule 970:1942, Wrought Steels for General Engineering Purposes.

Fig. 4 – The "Duck" – 2½ ton, 6 x 6 amphibian truck; entire hull a spot- and arc-welded assembly of sheet steel



Courtesy, E. O. Mann, Chevrolet Motor Division

■ **Fig. 5 – Summary of end-quench hardenabilities of production open-hearth heats of NE 8620 steel, purchased for use in truck transmission gears – spread in 30 heats shown by lowest and highest curves – shaded band indicates range considered desirable for satisfactorily uniform results in these applications**

cover all requirements from soft to hard steels; there are probably more ranges than would be required by any one group of designers, but each may be classed as "commercial" with present-day standard steels and heat-treating equipment of average precision.

3. *More Intelligent and Economical Selection of Steels by Metallurgists* – Selection will be based more upon physical and mechanical requirements than directly upon chemical composition. However, for some applications, relative susceptibilities of hardenability and maximum hardness to prior treatment and structure[25] may be considered. A frank and intelligent discussion of steel selection has been made by Gillette and others in a War Metallurgy Committee report.[26]

■ Hardenability as a Primary Requirement

Hardenability is one metallurgical term which should be understandable even to the most aloof engineer. It is not a foreign-language word, nor coined in honor of someone; it means simply the ability to harden (by heating and quenching).

Steels must of course be compounded by adding chemical ingredients. In structural steels to be heat-treated, however, chemical compositions are but means to ends, the most important end usually being proper hardening. Unfortunately, composition as ordinarily determined is not the sole criterion of hardenability; grain size and other factors, some difficult to evaluate, are also influential. After years of working on several of the factors, metallurgists finally began to realize that it might be simpler and more satisfactory to measure what was wanted – hardenability – than for users to continue to try to control a set of factors growing gradually more complicated and reaching far back into the steel's history.

Hardenability has been measured in many ways, to control steels and heat-treatments for specific parts. The procedure which is most generally accepted is the end-quench or so-called Jominy method, developed by Jominy and Boegehold.[27] The method consists essentially of quenching with a jet of water one end only of a uniformly heated test bar until the entire bar is cold, then measuring hardness along the side of the bar and plotting hardness readings; all details of the method have been carefully standardized.[28] Hardness adjacent to quenched end is indicative of maximum surface hardness obtainable, and shape of curve evaluates hardenability of the piece of steel tested. In general, hardenability of a well-made heat of steel is reasonably uniform throughout the heat.

Hardenability of conventionally made steels can be calculated from composition and grain size, using methods devised by Grossmann[28] and Field.[29] Results appear fairly accurate for steels of medium hardenability, somewhat less so for steels of very low and very high hardenability.

In 1942, suggestions were made by some steel producers that users specify hardenabilities and general types of steel desired, permitting producers to vary compositions beyond standard limits, provided hardenability requirements were met. We were outgrowing the conception that hardenability of a standard steel type could be represented by a single-line curve, but bands representing spreads of hardenability for standard composition ranges had not been established. Therefore, no one knew (for publication, at least) what ranges in hardenability steelmakers could reasonably be expected to meet.

Spread in hardenability within a given steel type is determined by:

1. Variations in contents of specified elements.
2. Variations in contents of residual or unspecified elements.
3. Grain size and normality (influenced by steelmaking practice and by contents of obscure elements).
4. Presence of "special addition agents" if any.

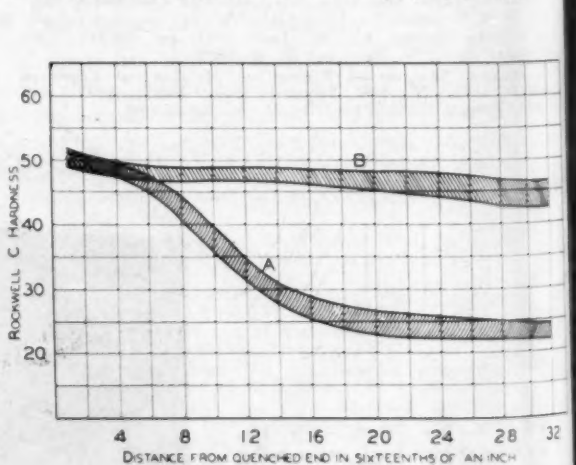
[25] See "Effect of Time, Temperature, and Prior Structure on the Hardenability of Several Alloy Steels," by J. Welchner, E. S. Rowland, and J. E. Ubben, American Society for Metals, 1943 Convention Preprint.

[26] See "Preliminary Report on the Selection, Evaluation, and Specification of Metallic Materials," War Metallurgy Committee of the National Academy of Sciences, National Research Council, July 7, 1943.

[27] See ASM Transactions, Vol. 26, June, 1938, pp. 574-606: "A Hardenability Test for Carburized Steel," by W. E. Jominy and A. L. Boegehold.

[28] See "Method of Determining Hardenability," 1943 SAE Handbook, pp. 314-324.

[29] See *Metal Progress*, Vol. 43, March, 1943, pp. 402-405: "Calculation of Jominy End-Quench Curve from Analysis," by J. Field.



■ **Fig. 6 – End-quench hardenabilities of steels listed in Table 2 – lower band indicates range of hardenability of 3 heats treated with special addition agent *A*; upper band indicates range of 4 heats treated with agent *B***

To establish fair specification ranges, we must know how much spread in hardenability users can tolerate. As an initial step, a joint AISI-SAE committee is now trying to ascertain how much spread has been involved in the past, in steels of present standard composition ranges. Of interest in these studies are such cases as shown in Fig. 5, which shows total spread in hardenability found by Chevrolet in 30 heats of NE 8620 steel, compared with the spread Chevrolet considers desirable (in this or any other steel) for use in certain truck transmission gears.

For many parts, it is necessary only to specify minimum required hardenability. This problem is relatively simple, and considerable steel is already being purchased on that basis.

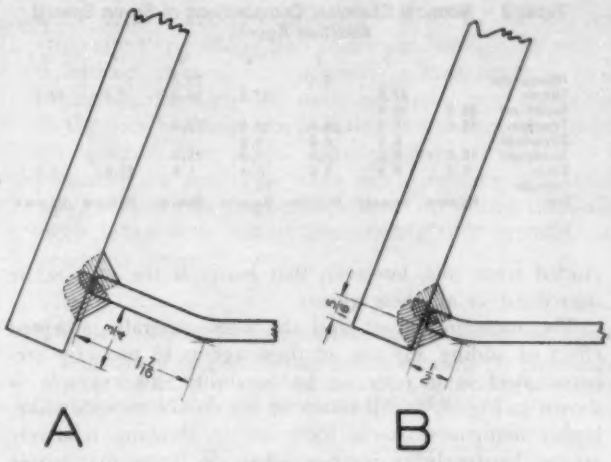
Other applications will require specifying definite ranges of hardenability – minimums to assure hardening of sections involved; and maximums to minimize quench-cracking, to prevent hardening too deeply or to too high hardness, or to control weldability. As an example, Table 2 lists seven heats of steel of nearly identical chemical

Table 2 – Plate Steels

Chemical Composition, %			Special Addition Agent
Carbon	Manganese	Molybdenum	
0.24	1.56	0.33	A
0.26	1.59	0.22	A
0.28	1.66	0.27	A
0.24	1.78	0.35	B
0.25	1.73	0.31	B
0.25	1.75	0.28	B
0.29	1.77	0.28	B
	All Heats: Nickel	0.04-0.07%	
	Chromium	0.01-0.04%	
	Grain Size	7-8	

compositions, one group of three treated with one special addition agent, and the other group of four with a different, more potent agent. These steels were rolled into ½-in. thick plate, fabricated into a certain part, hardened and tempered to Bhn 341-388, and assembled by automatic arc-welding. Hardenabilities of the two groups of steels are shown in Fig. 6. All these heats had sufficient hardenability to respond as desired to heat-treatment. The higher hardenability group, without exception, cracked very badly along the overhanging plate edge after being welded into the joint shown in Fig. 7A. No cracking or other difficulty was experienced with the lower hardenability group of heats during welding into the same design joint by the same process. It should be noted that near the left ends of the hardenability curves, indicating suitability for hardening in ½-in. plate sections, the two groups are practically identical. The significant difference from the standpoint of cracking near welds is farther to the right, in the slower cooling zone. It was possible in this instance to modify the admittedly critical joint (Fig. 7B) so as to permit using the higher hardenability steel; but this incident is illustrative of need for controlling hardenability within a range.

It now appears unquestionable that hardenability specifications are coming in the not distant future. A few large users are already purchasing some of their steels to hardenability specifications. Such users will continue and enlarge this practice as experience accumulates on certain parts. At present, steel producers are apparently not in a position to control hardenability consistently within the narrow ranges some users will specify; hence outlets for "off heats" must be available, either to the same or another user who can utilize hardenabilities slightly above or below the range



■ Fig. 7 – *A*: original, *B*: revised designs of welded joint – steels listed in Table 2 treated with special addition agent "A" welded satisfactorily by automatic arc process with ferritic electrode in joint *A*, but steels treated with agent "B" cracked under the same welding conditions; either type of steel welded satisfactorily by same process in joint *B*

aimed at, or to small users or warehouses buying only to chemical composition specifications. Therefore, general use of hardenability specifications may at first be confined to widely used standard steel types.

More extensive adoption of hardenability specifications will depend on speed of acquisition of knowledge by steel users, who must decide what they want. It will also depend on ability of steelmakers to improve controls so as to meet desired ranges in a high percentage of heats. More general adoption may conceivably await the next depression or at least full release of competition in industry, as the present seller's market and lack of competitive stimulus are not conducive to progress of this type.

■ Special Addition Agent Steels

Work on "special addition agent," "needled," "additive," "treated," or "intensified" steels (we still haven't a good name for them) started nearly 10 years ago. Use of and knowledge concerning the addition agents developed slowly, with a few large users (notably Buick and Timken-Detroit Axle) pioneering in experimental and production applications.

Progress was somewhat retarded during the early part of our engagement in the war, due to the tendency to standardize steels and to the necessity for experienced users of additive steels to abandon manufacture of their normal products in which those steels had been used. However, during 1943, study became stimulated because of desires to conserve normal alloying elements, all of which were scarce.

Needled steels are made, usually in open-hearth furnaces, by normal practices, each heat first being thoroughly deoxidized by conventional means, and a minute percentage of one of the special agents then added, in ladle or ingot mold. The agents are made and sold by various alloy manufacturers; compositions of some of those which have been or are available are listed in Table 3. They have one feature in common – all contain boron. The greater the percentage of boron present in an agent, the smaller the proportion of agent added to steel; it should not be con-

Table 3 – Nominal Chemical Compositions of Seven Special Addition Agents

	1	2	3	4	5	6	7
Manganese	....	....	8.0	....	20.0	....	....
Silicon	....	37.5	....	37.5	25.0	3.0	42.5
Vanadium	25.0	10.0	....	....	....	....	....
Titanium	15.0	10.0	20.0	10.0	15.0	....	....
Zirconium	....	5.0	4.0	4.0	....	....	....
Aluminum	10.0	6.0	13.0	7.0	15.0	....	....
Boron	0.2	0.5	0.5	0.5	1.5	11.0	3.5
Calcium	....	....	....	10.0	....	....	....
Iron	Balance	Balance	Balance	Balance	Balance	Balance	Balance

cluded from this, however, that boron is the only active ingredient in all these agents.

The most important and the only generally accepted effect of adding any one of these agents to properly prepared steel is increase in hardenability; an example is shown in Fig. 8.[30] All compositions do not respond alike, higher manganese steels, for example, showing relatively greater hardenability increases than do lower manganese steels of otherwise similar compositions.

Other effects of these intensifiers which have been reported or claimed by some, but which are not as yet generally accepted by all metallurgists are:

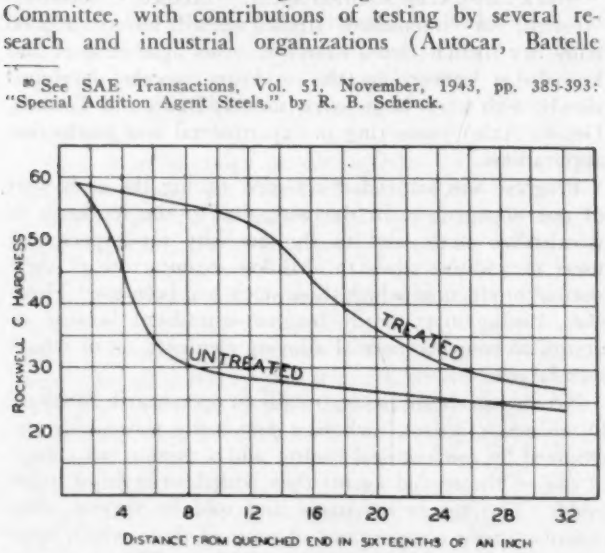
1. *Increased Toughness at High Hardness* – If this proves to be true consistently, it will be of utmost importance, as it will increase maximum safe strength for steels in many applications.

2. *Gradual rather than Abrupt Diminution of Hardness in Interiors of Shallow-Hardened Parts* – This would be an important property for parts stressed in bending or in torsion.

3. *Lowered Notch-Sensitivity in Fatigue* – The value of such an effect, if it actually exists, is so obvious that discussion would be superfluous.

Experimental projects, which will add to the store of information on special addition agents, are under way. They are sponsored and being handled by U. S. Army Ordnance (under direction of Col. J. H. Frye), a special committee (John Mitchell, chairman) of American Iron and Steel Institute, and a subcommittee (headed by R. B. Schenck) of SAE War Engineering Board Iron and Steel Committee, with contributions of testing by several research and industrial organizations (Autocar, Battelle Memorial Institute, Buick, Carnegie-Illinois, Caterpillar Tractor Co., Chevrolet, Chrysler, Deere, Eaton, Ford, Frost, GMC Truck & Coach, International Harvester, Mack, National Bureau of Standards, Republic, Saginaw Steering Gear, Spicer, Timken-Detroit Axle, Warner Gear, White, Willys-Overland, Wright Field, Youngstown Sheet & Tube, and possibly others).

[30] See SAE Transactions, Vol. 51, November, 1943, pp. 385-393: "Special Addition Agent Steels," by R. B. Schenck.



■ Fig. 8 – Increase in end-quench hardenability due to special addition agent – tests from bars rolled from two ingots of same heat; treated ingot analysis: C, 0.43%; Mn, 1.49%; agent added to steel in ingot mold; untreated ingot analysis: C, 0.45%; Mn, 1.52%

Additive steels have been and are used in large quantities in many production applications, including aircraft and ordnance armor, axle shafts, rear-axle gears, steering knuckles, and steering arms. Some of the savings made in alloying elements are actually startling, although the following examples do not necessarily imply that original and replacement steels have equal hardenabilities (symbol "T" in middle of steel number is used to indicate "treated" steel):

1. 10T40 for 3240 and 4340 in axle shafts.
2. 40T23 for 4620 in rear-axle drive gears.
3. 10T24 for 4815 in rear-axle drive pinions.

From specification standpoint, U. S. Army Ordnance has led the way with tentative specification AXS-930,[31] based on hardenability and strength-toughness relationships, and covering five types of needled steels. It is probable that the future will see additional user specifications for such steels, based primarily on physical test requirements.

The experimental projects mentioned above should assist in quantitatively evaluating effects of several agents with respect to steelmaking practices used, and should produce practical results in replacement of alloy steels with carbon or lower alloy "treated" steels for a considerable number of high-production parts used in military equipment.

It should follow that, as knowledge and experience increase, use of special addition agent steels will continue to expand, in both war matériel and civilian products. Some proponents of these additives believe that eventually they will be used in all alloy steels, thereby lowering amounts of conventional alloys required. Growth of needled steels will probably be influenced by two factors:

1. Adoption of hardenability specifications, which may promote voluntary use of special additives by steel producers.

2. Economics and price structure (will be discussed later).

■ NE Steels

Formulation and adoption of NE alloy steels resulted, of course, solely and directly from the war, because of scarcity of alloying elements. Compositions were worked out with three concepts in mind:

1. *Hardenability* of each composition should be closely comparable to that of a long-established, well-known and widely used alloy steel, if possible of equivalent carbon content.

2. *Minimum additions of virgin alloying elements* must be required, and no residual alloying element must be wasted.

3. *Mechanical properties of all alloy steels, heat-treated to identical structures and hardnesses, are alike.* That this is true for all hardness ranges is not generally admitted. It appears a safe assumption up to about Bhn 400; above that figure, toughnesses of some steels seem higher than

[31] See *Iron Age*, Vol. 152, Nov. 18, 1943, pp. 67, 168: "Needled Steel Specifications."

Table 4 – **Nominal Compositions of Present Standard NE Steels**

Series	C	Mn	Si	Ni	Cr	Mo
1300	0.30–0.50	1.75	0.28	...	...	...
*4700	0.19	0.60	0.28	1.05	0.45	0.25
*5200	1.02	0.35	0.28	...	0.50–1.45	...
8600	0.14–0.50	0.80–0.88	0.28	0.55	0.50	0.20
8700	0.20	0.80	0.28	0.55	0.50	0.25
9200	0.55–0.60	0.85	2.00	...	0–0.32	...
**9300	0.10–0.17	0.50	0.28	3.00	1.20	0.12
9400	0.15–0.50	0.95–1.35	0.28	0.45	0.40	0.12
9500	0.37–0.50	1.35	0.50	0.55	0.50	0.20

* For bearings only.
**For aircraft use only.

of others in proportion to hardness and strength. However, measurements of toughness and of strength of steels at high hardness are subject to greater relative error than at lower hardness; therefore the facts are still not clear.

Several changes have been made since adoption of the first set of NE compositions two years ago. The types now considered standard are listed in Table 4.

Many users were naturally reluctant, at first, to try to use the new NE steels. Active work, however, by several WPB technical advisory committees and others such as F. E. McCleary's NE Steel Testing Committee, each covering a type of product or a using industry, aided in promoting experimental work and production applications, chiefly in civilian and military products designed by manufacturers. Later, the Armed Services cooperated, and, in some cases with the help of technical committees such as the SAE War Engineering Board Iron and Steel Subcommittee, extensive replacements of high-alloy steels with NE steels were made in war matériel of both Army and Navy Ordnance design.

The NE types most widely used and now being promoted in preference to all others for automotive and other structural applications are the nickel-chromium-molybdenum or so-called three-alloy types. These steels may be briefly compared with older peacetime alloy steels in some of the more significant properties as follows:

1. *Relationship of Toughness to Hardness* – Complete data have not been obtained, but the great majority of available data indicate no appreciable difference. A few exceptions have been reported – for example, greater brittleness of NE 9420 than SAE 4320 in case-hardened transmission gear teeth[32] – so that some metallurgists have reserved judgment concerning toughness of NE steels at high hardnesses.

2. *Machinability* – Here again, in most cases no appreciable difference can be found.[33] A few instances of slightly inferior machinability of NE steels, such as spiral bevel gears after annealing and axle shaft splines after heat-treating to high hardness,[34] have been reported. Further work on annealing cycles will perhaps enable users to produce machinabilities in NE steels equal in all applications to those of comparable SAE alloy steels.

3. *Carburizing Characteristics* – Ratios of depths of cases to carburizing times for NE steels appear to be of the same order as for other alloy steels. Preliminary investigations have indicated, however, that at lower temperatures (1650 F), pack-carburizing produces appreciably higher carbon concentrations in outer portions of cases of three-alloy NE steels than in nickel-molybdenum steels (Table 5), both after direct quenching and after box cooling. At 1700 F, differences after direct quenching are so small as to be negligible, although NE steels are apparently less subject to the decarburizing action during box cooling characteristic of some other steels. This property, if shown to be consistent by results of more extensive tests now under way, may be an important reason for using three-alloy steels when slow cooling after carburizing in solid compound is desired.

[32] Private Communications from S. L. Widrig, Aug. 31, 1943, and Sept. 11, 1943.

[33] See *Metal Progress*, Vol. 43, April, 1943, pp. 543-547: "Machinability of the National Emergency Alloy Steels," by O. W. Boston.

[34] See SAE Transactions, Vol. 51, March, 1943, pp. 84-93: "Production Experience with National Emergency Steels," by R. W. Roush.

Table 5 – Carbon Concentration in Cases of Carburized Steels

Preliminary data on average carbon contents (in percentages) of outer 0.0025 in. of steel test bars after carburizing in nonburning compound for 8 hr at temperatures indicated.

	1650 F		1700 F	
	Direct Quench	Box Cool	Direct Quench	Box Cool
4620	1.06	0.81	1.07	0.94
8620, 8720, 9420	1.47	1.62	1.15	1.10

4. *Hardenability Spread* – As use of hardenability as a primary requirement grows, there should develop a trend toward selection of composition types which inherently fall within narrower hardenability ranges. Under present conditions, lack of alloy-free scrap means waste if single- or two-alloy steels are used. After the war, the situation in some steel mills may be such that single-alloy steels can be made without appreciable residual unspecified alloying elements; but other mills may, for several years at least, find it impossible to make large quantities of low-residual steels. Since we still are without sufficient test data to establish hardenability spreads authoritatively, the subject must for the present be studied by calculation. We shall use the Field method mentioned above,[20] confining our examples to medium-hardenability steels for which the method is known to be reasonably accurate, and, in the interest of simplification, eliminating for the present purpose effects of grain size (using No. 7 in all cases), of phosphorous and sulfur (which nearly cancel each other), and of residual vanadium and copper, since all these effects are the same for all steels here considered. Hardenability of each steel is calculated three ways:

A. Each element at minimum of specification range, with no unspecified nickel, chromium, or molybdenum.

B. Each element at maximum of specification range, with no unspecified nickel, chromium, or molybdenum.

C. Each element at maximum of specification range, with each unspecified element at maximum permitted by AISI standards (nickel 0.25%, chromium 0.20%, molybdenum 0.06%). In steels containing specified amounts of Ni, Cr, and Mo, calculations B and C are of course identical.

Table 6 lists compositions and calculated Grossmann critical diameters of the 0.40% carbon steels here considered. Calculated A, B, and C hardenability curves for the SAE one-alloy, SAE two-alloy, and NE three-alloy steels are shown in Figs. 9, 10, and 11, respectively. They indicate clearly that, in absence of residual unspecified elements, hardenability spreads are roughly proportional to the number of alloying elements contained, also that ranges widen in proportion to number and percentages of unspecified elements present. Of unspecified elements, the

Table 6 – Composition Limits of Some 0.40% Carbon Low-Alloy Steels

For each steel: "A" represents all elements at low limits; "B," all specified elements at high limit with no unspecified elements present; "C," all specified elements at high limit plus maximum permissible amounts of unspecified nickel, chromium, and molybdenum. "Critical diameter" means maximum diameter in which each steel will harden, with ideal quench, to 50% martensite structure at center, calculated by Grossmann's method.[23]

	Chemical Composition, %						Critical Diameter
	C	Mn	Si	Ni	Cr	Mo	
1340 A	0.38	1.60	0.20	....	....	....	1.57
B	0.43	1.90	0.35	....	....	....	2.12
C	0.43	1.90	0.35	0.25	0.20	0.06	4.03
2340 A	0.38	0.70	0.20	3.25	....	....	1.81
B	0.43	0.90	0.35	3.75	....	....	2.74
C	0.43	0.90	0.35	3.75	0.20	0.06	4.75
4040 A	0.38	0.75	0.20	....	....	0.20	1.40
B	0.43	1.00	0.35	....	....	0.30	2.45
C	0.43	1.00	0.35	0.25	0.20	0.30	3.90
5140 A	0.38	0.70	0.20	....	0.70	....	1.91
B	0.43	0.90	0.35	....	0.90	....	2.97
C	0.43	0.90	0.35	0.25	0.90	0.06	3.83
3140 A	0.38	0.70	0.20	1.10	0.55	....	2.43
B	0.43	0.90	0.35	1.40	0.75	....	4.13
C	0.43	0.90	0.35	1.40	0.75	0.06	4.92
3141 A	0.38	0.70	0.20	1.10	0.70	....	2.66
B	0.43	0.90	0.35	1.40	0.90	....	4.47
C	0.43	0.90	0.35	1.40	0.90	0.06	5.33
3240 A	0.38	0.40	0.20	1.65	0.90	....	2.38
B	0.43	0.60	0.35	2.00	1.20	....	4.38
C	0.43	0.60	0.35	2.00	1.20	0.06	5.22
4640 A	0.38	0.60	0.20	1.65	....	0.20	1.93
B	0.43	0.80	0.35	2.00	....	0.30	3.64
C	0.43	0.80	0.35	2.00	0.20	0.30	5.15
8640 A	0.38	0.75	0.20	0.40	0.40	0.15	2.72
B	0.43	1.00	0.35	0.70	0.60	0.25	6.08
C	0.43	1.00	0.35	0.70	0.60	0.25	6.08
9440 A	0.38	0.90	0.40	0.20	0.20	0.08	2.32
B	0.43	1.20	0.60	0.50	0.40	0.15	5.28
C	0.43	1.20	0.60	0.50	0.40	0.15	5.28

present maximum permissible content of chromium is by far the most potent in widening hardenability spreads, with molybdenum next. One way, impracticable at present, of controlling hardenability closely would be to keep different types of steel scrap accurately segregated, and to make single-alloy steels to present composition limits.

Having summarized some of the NE steel developments and made a few comparisons between NE and SAE alloy steels, we now consider possible future effects of NE steels on metallurgical practice, both for the duration of the war and in subsequent peacetime.

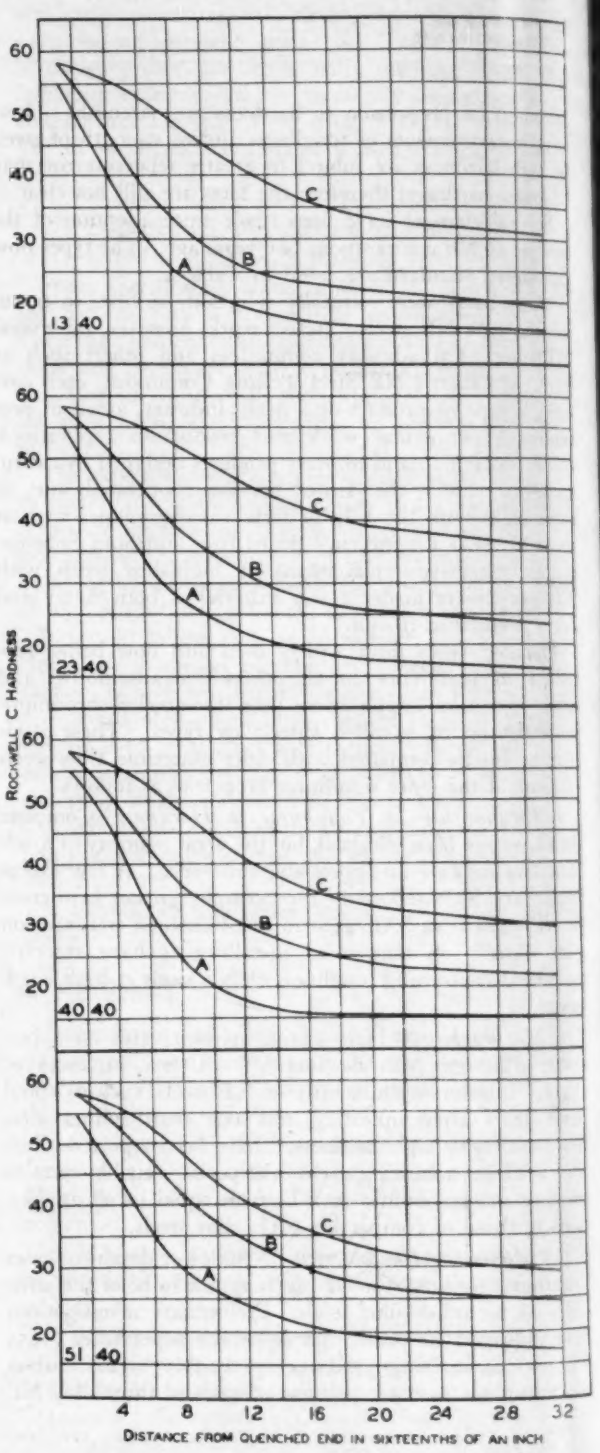
There has been for some months a definite trend toward a single NE steel type. The 1300 series manganese steels are increasingly difficult to obtain, because of scarcity of scrap free from other alloying elements; and the relatively high manganese and silicon 9400 series, although perfectly satisfactory for countless applications, was never as cordially received as had been hoped. The latest modification of the 9400 series eliminated the influence of dislike for high silicon, but the recent easing of molybdenum has removed the chief motive for establishing 9400 steels. Therefore, and because much alloy-steel scrap is too high in nickel to make 9400 steels, and because 8600 and 8700 steels are held generally in high favor, it now seems probable that the 9400 steels will die out, and that 8600 or possibly restored 8700 steels will take their place.

Permanency of NE steel compositions after the war will depend on a half-dozen factors:

1. *Technical Superiority of Various Types* – Except for a small proportion of specialized applications, and possibly except from standpoint of hardenability control, this factor is of small importance, since, as has been previously implied, all low-alloy structural steels are interchangeable for most applications.

2. *Sales Efforts* – Understandably, efforts to sell virgin alloying elements and higher alloy steels can be expected after the war. Such activities will of course militate against the use of multiple-alloy steels.

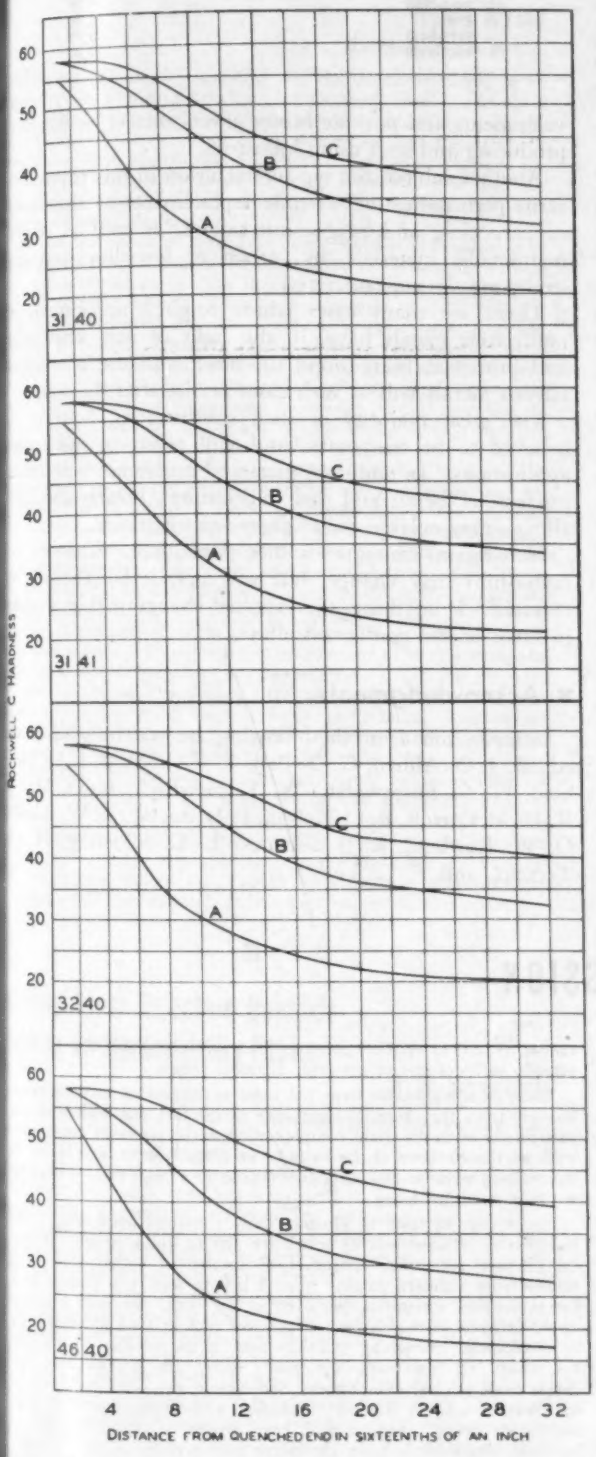
3. *Length of the War* – The longer the war lasts, the more familiar with NE steels we shall become, and the less we shall remember about our old pets. For example,



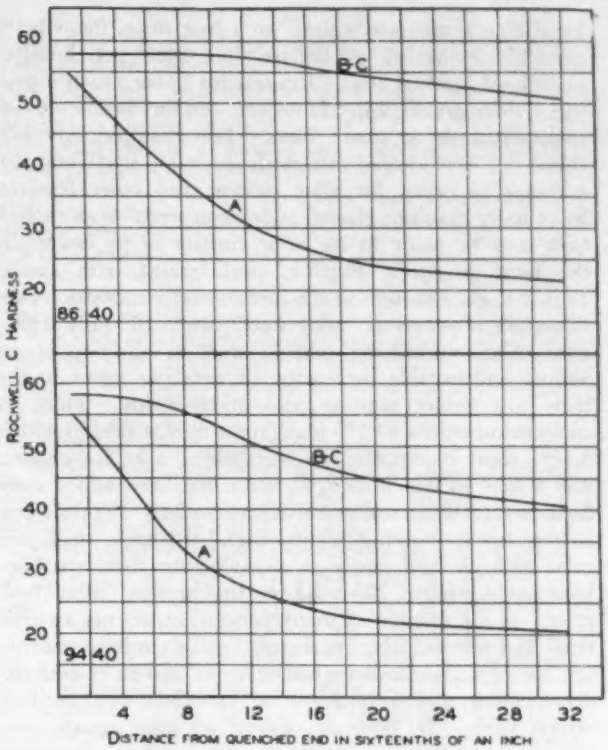
■ Fig. 9 – Hardenability curves of single-alloy SAE 1340, SAE 2340, SAE 4040, and SAE 5140 steels, calculated from compositions in Table 6

another two years' experience with an unchanging nickel-chromium-molybdenum alloy series would go a long way toward cementing permanent friendships with those steels.

4. *Regimentation after the War* – There is much talk of continuing bureaucratic control over manufacturing for an indeterminate period after the war, and many agree that abrupt abandonment of such control would be undesirable. As long as regimentation persists, there may be a tendency to continue to use NE or other multiple-alloy steels. However, the motive for close governmental control will, in the minds of many, have disappeared with the war. Pressure will be exerted upon politicians by industry and by individuals to abandon regimentation. The pressure will to some extent be resisted, but it is possible that, within a short period, the pendulum, long high on the side of governmental control, may swing far the other way. In that event, our desires to be individualists will no longer be suppressed, and their release may influence us to move away from "emergency" steels and other practices associated with wartime regimentation.



■ Fig. 10 – Hardenability curves of two-alloy SAE 3140, SAE 3141, SAE 3240, and SAE 4640 steels, calculated from compositions in Table 6



■ Fig. 11 – Hardenability curves of three-alloy NE 8640 and NE 9440 steels, calculated from compositions in Table 6

5. *Nationalistic Preparedness* – This is the first mechanized war. The next war will probably be much more mechanized. By that time, many of our natural resources will have been depleted, and all natural materials, including ferrous alloys, will be much more critical than they have been in this war. In anticipation of future needs, appreciation of the value of peacetime conservation – development of nonstrategic alloys and efficient utilization of all raw materials and residues – may constitute a reason for continued use of multiple-alloy steels. A related factor will be willingness of all the Armed Services to learn during peacetime how to build matériel for the next war with minimum use of strategic materials.

6. *Price Structure* – In normal peacetime manufacturing, the economics of any situation is usually the ruling factor. Steels are no exception; and, assuming eventual return to anything like what we call normal operations, the five factors just discussed will, in the great majority of applications, be relatively insignificant except as they influence steel economics. Present net prices of steels are computed

by adding numerous "extras" to a base price, items being generally expressed as dollars and cents per hundred pounds of finished steel. Reasons for having such a pricing system are logical. However, certain details are not understandable to many users. For example, any hot-rolled bar steel classed arbitrarily as "alloy steel" costs, in addition to extras for alloy content and other features, $0.55 more than one classed as "carbon steel," even though both may be made in the same furnace or by essentially the same practice. Further, steel treated with special additives, even though in the carbon-steel composition classification, is priced as "alloy steel," when the only difference in manufacturing practice may be addition of an almost undetectable percentage of needling agent to the ladle just before pouring into ingot molds. Thus, in comparison with a 1.50% manganese steel, a steel to which 0.25% more manganese has been added costs $0.65 more, and a steel to which 0.003% boron has been added costs $0.80 more. It is realized that, particularly in operations subsequent to pouring ingots, high-hardenable steels are more difficult and costly to manufacture than are low-hardenable steels. Nevertheless, the present "alloy base price," in the opinion of many persons, represents an artificial and unreasonable increment which tends to encourage use of carbon and high-alloy steels and to discourage development and application of low-alloy and needled carbon steels. If, however, special addition agents continue to show merit in accordance with results to date, they will force their way into the picture, and eventually should influence price structure sufficiently to make their use economical in carbon as well as alloy steels. As to standard extras for alloy content, Table 7 lists them for a few widely used standard steel types. A glance at this list reveals that certain other equally good alloy steels are priced lower than NE steels. It is probable that readjustments will be made; in fact, some modification will be required to keep NE steels alive after the war.

Table 7 – Alloy-Steel Price Extras

Steel Type	Extra for Composition (per cwt)
1330-1350	$0.10
3045	0.80
3115-3150	0.70
4023, 4027, 4032-4068	0.45
4119, 4120	0.60
4130-4150	0.55
5120, 5145, 5150	0.35
5130-5140	0.45
8613-8650	0.75
8720	0.80
9415-9440	0.75
9442-9450	0.90

■ Steels Versus Nonferrous Metals

The preceding discussions have dealt with wartime developments and possible future effects thereof, within steel producing and steel using industries.

Another subject, too big for inclusion in this paper, concerns permanence of wartime replacements of aluminum, copper, zinc, and magnesium alloys, as well as several nonmetallic materials, by carbon or low-alloy structural steels, mostly unheat-treated.

There are many cases where original applications of nonferrous metals have, in the light of new knowledge and processes, been found unsound; and use of cheaper ferrous metals will in such cases persist after the war.

The great majority of such substitutions, however, is believed to be temporary, and will revert to nonferrous applications. In addition, increased experience with many nonferrous metals and their fabrication – notably the light alloys – may enlarge their spheres of usefulness.

Here again, economics will be paramount. Economic relationships may change, but will not, it is believed, be reversed. If anything, the expected change will be slightly in favor of the nonferrous alloys.

■ Acknowledgments

Contributions from the following are gratefully acknowledged: J. O. Almen, C. O. Ball, E. V. Bennett, J. M. Diebold, G. C. Farnsworth, W. F. Herdrich, E. O. Mann, R. H. McCarroll, R. C. Nelson, F. L. Racine, R. W. Roush, G. W. Rowland, R. B. Schenck, L. L. Schiering, H. G. Taracks, and F. C. Young.

DISCUSSION

Narrow Hardenability Bands Considered Too Restrictive

– O. W. McMullan

Youngstown Sheet & Tube Co.

I WISH to make a few comments regarding Mr. Eddy's discussion of hardenability specification ranges, which is illustrated in Fig. 5. Chemical ranges now listed in SAE and AISI specifications represent no wider than reasonable working limits in steel making. Any other added specifications which further restrict these limits may, if carried far enough, result in unjustifiably large rejections of heats which under normal conditions must eventually have an influence on the price structure. Some of the desired "bands" now being suggested which include only a small portion of the hardenability range to be expected from the type of analysis may represent an ideal for which to strive, but the art or science of steelmaking has not reached a stage of development that will permit working to such narrow bands without undue hardship on steel producers when other means of disposing of heats are not available. Furthermore, it is extremely doubtful if engineering knowledge of stresses and conditions encountered in service has developed to a point where it can be proved that the chosen bands actually include all of the most desirable range; in fact, in extreme cases no evidence is available that the band actually includes any of the most desirable range.

There is a reasonable basis for some restriction of extreme ranges possible from the chemical limits due to the fact that seldom if ever, particularly with the more complex alloy steels, are heats encountered with all the elements at the extreme of either low or high limits. In the writer's opinion this justifies working to a high or low limit but not to a restricted range.

Fig. 5 may be used to illustrate how a desired band can, if used completely, be unnecessarily restrictive and in doing so serve no useful purpose for either consumer or producer. Rather than being results from standard Jominy tests, I believe Fig. 5 is derived from L-type Jominy specimens, but even in the latter case only a portion of the left end of the diagram represents cooling rates to be expected in commercial quenching practices and in the smaller section sizes for which NE 8620 steel is normally used. The desired band here covers most of the actual spread, whereas at the right end, 50% of the heats would be rejected assuming a uniform scatter of heats within the range. Most of such heats would be rejected because of hardness obtainable in sizes far larger than actually in use. Possibly there might be some minor influence on annealing and machinability, but certainly not on core hardness of the article treated. It is, therefore, suggested that there should be distance limits on hardenability chart specifications covering only the range needed for the application

avoid unnecessary rejections of steel actually possessing the de- hardenability characteristics for the parts intended.

■

operation Emphasized as Need Steel-Industry Developments

– Henry Wysor

Bethlehem Steel Co.

VERY now and then a composite analysis should be made of an industry such as that in which we are engaged. Steel is a complex terial and, for its maximum usefulness and safety, requires the hest degree of technical skill in its preparation and treatment. It imely that these observations are set forth and so emphasized for benefit of those who would be more fully informed and for ers who are forgetful. All do well to review most carefullv this presentation by Mr. Eddy.

am glad that the other discussers have emphasized the subject of gue endurance – for that is of paramount importance in present gn and manufacture of parts in military equipment, and will be ally so with the return to normal manufactures.

The relation of notch impact to fatigue is most interesting and mising for further research. The importance must not be over- ked of actual performance and, in many instances, this has proved only real means of determining endurance for a part. An example this came to our attention and somewhat to our humiliation, ut a quarter century ago when we found by actual trial that emer steel shafts stood up better than the best alloy shafts we ld furnish in the old Griffin Mill cement crusher. However, it is practicable to put the steel to work in many instances, and the test under standard or simulated conditions is very helpful. re is often a serious hazard in the application of technical method definition, and this is accentuated where inexperience is con- ed. After arriving at optimum conditions for testing material, as h, we may still be uninformed or even misled in the application ervice conditions.

believe the nearest I might come to taking issue with Mr. Eddy n his philosophy. I am aware of the benefit of competition and ld not care to be associated in any enterprise which did not have stimulus. On the other hand, I feel that cooperation, rather than petition, is the word to emphasize in technical development re the facts are common property. It did not take this war to g home to the steel industry the necessity for exchange of ideas. equipment and processing have been open to competitive in- sts for many years, and it is my feeling that future development the steel industry along technical lines will have its main impetus collaboration among producers and, as well, between producers consumers of this most essential engineering material. Further- e, our national outlook calls for the highest degree of cooperation, through our combined effort we will render the maximum ser- to our country.

■

nsiders Steel Selection Depends t Alone on Hardenability Factor

– H. B. Knowlton

International Harvester Co.

the early days of the automotive industry a great many parts ere made of plain carbon steels which were low in hardenability. that time very little attention was given to the hardenability of erent steels. Failures of steering parts were not uncommon. A en steering knuckle was sometimes considered a plausible alibi drunken driving. Gradually, alloy steels came into use for many the important automotive parts, and service performance was tly improved even though the total weight of the vehicle was ced. Sometimes steering knuckles were found badly bent after ion, but rarely ever broken. On the whole it may be said that alloy steels performed satisfactorily.

With the war came the NE steels, which were set up on a basis of enability equivalent to that of the steels they replaced. To a very degree these steels have proved satisfactory in service. There however, been some exceptions which lead to the thought that enability, while necessary, is not the only factor governing the essful performance of steel.

me time ago, we conducted a large number of torsion fatigue of axles having a 2¾-in. involute spline in the critical section. original steel used for this part was 4140, and the axles were heat-treated for a Brinell hardness range of 321-388. A number of alternate steels were tested in comparison with 4140. All but two were heat-treated for the same range of hardness. These were low carbon alloy steels and were case hardened. It was found that the steels could be divided into three general classes on the basis of torsion fatigue strength of finished axles. The two case-hardened steels had the highest fatigue strength.

A number of steels, including 4140, 8744, 8245, 9445, and 13T40 had medium fatigue strength and probably could be classified as equivalents within the range of experimental error. Particular attention is called to the fact that 13T40 (a very low alloy steel treated with addition agent) was nearly as good as the other alloy steels. However, two types of steel, 8442 and 7045, were somewhat inferior in fatigue strength. These were manganese-molybdenum steels and have since been discontinued. However, their fatigue strength was still above the service stress for these particular axles.

The hardenability of 8442 is almost identical to that of the original 4140, which was successful. The cross-sectional hardness, and consequently the cross-section strength, of an axle depends upon the hardenability of a steel. It is desirable that the strength at all points over the cross-section shall exceed the stress which occurs at each point.

When no stress-raisers are present, the stress in bending or torsion should theoretically decrease as a straight-line function from a maximum at the surface to zero at the center. With stress-raisers present, a very high stress appears in the locality of the stress-raiser and the stress decreases very rapidly below the finally reached zero at the center.

Plots of hardenability and theoretical stress indicate that all of the steels tested had sufficient hardenability to guarantee that the strength at all points on the cross-section would exceed the stress. The difference in performance, therefore, must be due to some other factor than hardenability. How then can we explain the difference in fatigue strength? What are the factors that contribute to successful performance of automotive parts? The following are some of the points which should be considered in selection of steel for automotive parts:

1. Most automotive parts fail in bending or torsion fatigue. According to Almen, this type of failure always starts as a tension break in the surface layers. We are primarily concerned, therefore, with the fatigue strength of the surface layers and any condition of stress which may affect fatigue strength.

2. There must be sufficient hardenability to assure that the surface hardens completely (100% martensitic), with a production quench which may, in many cases, have to be an oil quench. Unhardened spots (pearlitic or troostite) even of microscopic size may not affect the surface Brinell or Rockwell hardness, but may constitute weak spots where failures may start. Sometimes deep penetration of hardness is specified as an assurance that the surface will harden 100% martensitic.

3. There must be sufficient hardenability to assure that the specified surface hardness can be met with the standard production quenching and drawing treatment. In fact, over 99% of the heats of steel received should have this minimum hardenability. The heat-treater stands or falls on the basis of whether the percentage of rework in his department is ½, 1, or 2%.

Mr. Eddy has shown that all of the steels have wide hardenability bands. From the standpoints of either production or service, it is of maximum importance that a certain minimum hardenability be maintained. The narrower the range of hardenability above the required minimum the better, for high hardenability may result in poor machining or in cracking during hardening.

4. There must be sufficient toughness in the critical area to withstand the local deformations which must take place in service. What this is, depends upon design and type of service. The smaller the fillet or the sharper the notch, the greater will be the toughness required. Consequently there is always a minimum toughness for each type of part. Additional toughness above the minimum may not be of any benefit.

5. There may be opportunity for development of better techniques for the measurement of toughness. It is usually considered that the notched-bar impact test is a measure of toughness. Our fatigue tests of production parts indicate that notch fatigue strength may be more critical than notch impact strength. In other words, two steels which show identical notch impact strength after a production heat-treatment, may not show the same performance in fatigue tests of finished parts.

This seems entirely logical. In notch fatigue testing, the practice is to determine the stress producing the first crack in the notch itself. Notch impact testing determines the energy required to rupture the entire cross-section.

6. It is certainly true that all heat-treatment may produce residual stresses in the surface layers. These may be either tensile or compressive stresses and will decrease or increase the fatigue strength of the surface layers. A great deal more study should be given to this

subject. The type and degree of stress probably depends upon the following:

a. Speed of quenching.

b. Speed of transformation of the steel (decomposition of austenite).

c. Relation between the behavior of the surface and internal layers during quenching.

d. Temperature at which transformation takes place.

e. The effect of tempering or drawing.

f. Final treatment, such as cold rolling or shot blasting.

■

Specification Needed For Surface Finishes

– Roger F. Mather

Willys-Overland Motors, Inc.

FIRST, as regards surface finish, the ideal surface is one which is both smooth, so as to be free from stress-raisers, and which has a compressed surface, in order to increase the fatigue strength. Honing is a method of obtaining a smooth surface and shot blasting is a method of obtaining a compressed surface, but neither method meets both requirements. At present, vapor blasting is the only mechanical means of obtaining both conditions, but it may be that the newly developed application of this method to aircraft parts means that the matter is still too confidential to be published. Another wartime development in fatigue endurance is the setting up of standards for surface finish. This work has been promoted largely by ASME, ASTM, Watertown Arsenal, and others, and many companies are also doing work along these lines. It appears that some sort of specification for surface finish is necessary to obtain consistent results on the fatigue strength of machined parts. Instruments for surface analysis in production have been developed at the same time.

The purchase of steel on the basis of hardenability rather than chemical analysis has been shown desirable by both practical experience and calculation of the maximum variation possible. However, it has not yet been proved that more uniform hardenability can be obtained through specification on this basis. It is possible that the use of special addition agent steels will provide a means of closer control, but no data have yet been published on this point. It seems unlikely that we shall ever relinquish entirely our purchase of steel on the basis of chemical composition. The minimum limit on carbon content is required to provide sufficient surface hardness, and to ensure that the drawing temperature is sufficiently high to eliminate residual stresses; the maximum on carbon is necessary to avoid difficulties in machining, distortion and cracking on quenching, low impact strength, and so on. The minimum on manganese is required to provide sufficient deoxidation and desulfurization; the maximum is needed because manganese contents over 2% tend to give brittle steels. The maximum on phosphorus is required to avoid brittleness, namely, the sum of the carbon and phosphorus must not exceed 0.25%. The maximum on sulfur is required to avoid cracks when rolling and forging. The minimum on silicon is needed for deoxidation and grain-size control; the value of the maximum on silicon has been proved in the case of the 9400 steels, whose composition was revised partly in order to avoid the grain boundary precipitate in carburized parts. However, in the case of the alloying elements proper, namely nickel, chromium, molybdenum, vanadium, and the special alloy additions, no limits need be specified on chemical composition, except in so far as some control may be necessary to provide corrosion resistance, low temperature impact strength, machinability, and so forth. The consideration of machinability alone is sufficient to warrant some control of chemical composition in most cases.

One of the most significant things about the National Emergency steels is that the effect of two different alloying elements of equal hardening power is greater than the effect of using an equal amount of one element alone. This is something that is difficult for the engineer to understand. It can however be proved mathematically, starting from the generality that the curve of hardening factor versus alloy content is a straight line. Without this knowledge the engineer probably assumes that the effect is similar to that of mixing a cocktail, in which it is well known that small quantities of several different materials are more potent than a large quantity of a single type.

To me, one of the most important things about the SAE steels is that the additions do not effect the fabricating properties, in the ranges in which they are usually recommended. In particular, the machinability of these steels is the same as that of the untreated steel, so that there is a possibility of reducing the cost of the finished part in this way.

With regard to welding, a fundamentally new type of welding has been developed in the use of plastics for joining similar and dissimilar metals, and this may find wide application in peacetime industry.

Another group of wartime developments in metallurgy relates to methods of strain analysis and inspection. Perhaps these methods can hardly be called developments in the metallurgy of steel, yet they definitely influence our ability to get the maximum performance from finished parts. The new methods of analysis include the electric strain gage, brittle lacquers, and three-dimensional photoelastics. The use of these tools makes possible the better analysis of stresses and strains, so that design can be improved. There have also been considerable advances in methods of inspection production, notably the Magnaglo and Zyglo methods of fluorescent inspection.

During the discussion of this paper a question arose as to the maximum hardness at which NE 9400 gears may be used. We have approved for production transmission gears made from NE 9420 with hardnesses of 58 Rockwell C minimum.

■

Induced Internal Stress Can Lower Load Stresses

– G. C. Riegel

Caterpillar Tractor Co.

UNDER the heading of fatigue endurance, it seems appropriate to place greater emphasis upon the factors which metallurgists are beginning to evaluate in terms of stresses in the components of construction. For example, it is impossible to enhance the mechanical properties of any item by heat-treatment without imparting stresses of considerable magnitude. Many times, the heat-treated item is stressed so highly by this processing that it is almost ready to fail without the imposition of additional stresses. One can observe this when a highly hardened, carburized surface is checked with very light grinding. Similarly, highly hardened, induction heated journals may craze badly with the first "border-line" failure of lubrication. Many gears are so highly stressed from the efforts to promote wear resistance with increased hardness that a little, suddenly applied load will cause the gear to shell out its teeth or break through the web.

In recent months in the laboratories with which the writer is connected, stress analyses have been made of a number of gears, piston pins, track pins, and other items, which have enabled us, by selective methods of heat-treatment, to apply the stresses in such a manner as to oppose the stresses encountered in service, thus diminishing, instead of increasing, the tendency of the part to break or fail suddenly in a machine of which it was a component. This new use of peening, cold working, selective hardening, as by induction, offers unusual opportunities for overcoming certain structural and design limitations, as well as offering weight reduction opportunities in machines in which this feature is paramount.

In other words, as engineers and metallurgists, the internal stresses should be made to work for us, instead of against us.

■

Author's Comment On Points Raised

– W. P. Eddy, Jr.

Pratt & Whitney Aircraft

MR. McMullan's comments concerning the question of need for close hardenability control are thought-provoking, and lead directly to the designer. If narrow physical-property ranges are in some cases unnecessary, engineers should be the first to recognize it; however, where narrow ranges are specified, the manufacturer must attempt to conform. Mr. McMullan's comments concerning relative unimportance of right-hand portions of hardenability curves are of course in order when quenching small or medium sections is involved. These portions of the curves may, however, be of great importance when, as in the example given in the paper, hardening during slow cooling is a factor.

The author is in entire agreement with Mr. Wysor's belief that cooperation rather than competition is conducive to progress, when the facts are common property. However, the statement in the paper to which exception was taken concerned purchasing steel to hardenability specifications; the facts on which such practice will be based are not at present common property.

Discussions of Messrs. Knowlton, Mather, and Riegel are all pertinent, but seem to require no further comment. The author is appreciative of the several interesting discussions submitted, each of which adds considerably to the paper as originally presented.

THE DETERMINATION OF FUSELAGE MOMENTS

by C. E. PAPPAS

Chief, Aerodynamics and Flight Test Department, Republic Aviation Corp.

I. Introduction

The determination of the unstable moment of a body immersed in an inviscous, incompressible fluid can be made in terms of additional apparent mass coefficients. Coefficients of additional apparent mass have been calculated for a body of revolution by Lamb[1] and the unstable moment has been determined by Munk[2] in terms of these coefficients.

O'Donnell[3] has taken into account the variation of the additional apparent mass coefficients when the shape departs from a true prolate ellipsoid.

Boundary layer effects were considered by Smith[4] in the determination of the longitudinal additional apparent mass coefficient, and it was shown that the coefficient increased when the energy of the boundary layer was taken into account.

The purpose of this paper is to present theoretical coefficients of additional apparent mass for an ellipsoid of three unequal axes and to indicate the change in the theoretical coefficients when boundary layer, interference, and turbulence effects are considered. In order to determine the effect of a viscous, incompressible fluid on the additional apparent mass coefficients, wind-tunnel tests on circular fuselages alone and in combination with a wing were used to determine the extent of the variation in the theoretical coefficients.

Pitching and yawing moments can be readily determined for circular fuselages alone, and, in addition, the change in fuselage moments resulting from the interference effects of a wing can be found from the figures presented in the main body of the paper.

An attempt has been made to give the necessary theory associated with the concept of additional apparent mass so that the reader may understand better the actual variation of moment of a body moving in a viscous fluid in terms of the mass coefficients.

II. Concept of Additional Apparent Mass

The concept of the additional apparent mass of a body is based upon an inviscous fluid. A body in steady motion in such a fluid possesses no resultant body forces, that is, lift or drag. When a body moves through an inviscous fluid, however, a certain amount of energy is stored up in the streamlines of the fluid, the energy being a function of the body shape. When the body is at rest no energy field exists. In order to start the body in motion it is necessary to overcome the inertia of the body and the inertia of the fluid since the shape of the body will cause the fluid to diverge at the nose and to converge at the rear of the body. Considering the body and the fluid medium as one system, the total kinetic energy may be written as $\frac{1}{2} M U^2 + \frac{1}{2} M_a U^2$ where M is the mass of the body and M_a is the ratio of the kinetic energy of the fluid to $\frac{1}{2} U^2$, and U is the speed of the body. The quantity M_a is called the additional apparent mass of the body.

There is an important difference, however, between the mass of the body and the apparent mass of the body. The former is the same for all directions of motion but the latter is not necessarily so. This is associated with the fact that the resultant momentum of the flow is generally not parallel to the direction of motion. There are, however, at least three axes of the body, mutually at right angles to each other and designated principal axes, for which the momentum is parallel to the direction of motion if the direction of motion coincides with one of these axes.[5] Moreover, if M_a, M_b, and M_c denote the additional apparent masses when the direction of motion coincides successively with each of these axes, the total kinetic energy for any direction of motion can be written as:

$$\tfrac{1}{2}M(U^2 + V^2 + W^2) + \tfrac{1}{2}M_aU^2 + \tfrac{1}{2}M_bV^2 + \tfrac{1}{2}M_cW^2$$

where U, V, and W are the components of the velocity along the principal axes.

Considering now motion parallel to one of the principal axes, the term $\frac{1}{2} M_a U^2$ is the energy which goes into deflecting the streamlines around the body during the process of accelerating the body from rest. The energy remains in the streamlines around the body when it is moving with steady motion and is given up to the body only as the body comes to rest.

When a body is in steady motion, the forward part of the body supplies energy to the fluid and at the rear portion of the body energy is being transmitted to it by the fluid, the difference in energy in front of and at the rear of the body being zero. The energy in the streamlines remains within the streamlines as long as the body is in steady motion.

The energy stored up in the streamlines should not be confused with the pressure drag of a body moving in a viscous fluid. In such a case a drag results because of the boundary layer which exists over the body and which increases in thickness along the length of the body. It is because of this boundary layer resulting from viscosity that

[The Wright Brothers Medal for 1943 was awarded to Mr. Pappas for this paper which he presented at the SAE War Engineering Production Meeting (Annual Meeting), Detroit, Mich., Jan. 14, 1943.]

[1] See "Hydrodynamics," by H. Lamb, Cambridge University Press, 1932.

[2] See NACA Technical Report No. 213, 1928: "A Résumé of the Advances in Theoretical Aeronautics made by Max M. Munk," by Joseph S. Ames.

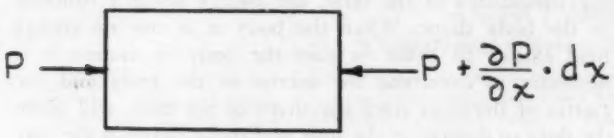
[3] See "On the Motion of Elongated Bodies of Revolution through a Perfect Fluid," by William J. O'Donnell, Catholic University of America, Washington, D. C., 1938.

[4] See *Journal of the Aeronautical Sciences*, Vol. 3, September, 1935, pp. 26-31: "Longitudinal Potential Flow about an Arbitrary Body of Revolution with Application to the Airship 'Akron,'" by Richard H. Smith.

[5] See "Fundamentals of Fluid Dynamics for Aircraft Designers," by by Max M. Munk, Ronald Press Co., 1929.

a pressure drag exists in a viscous fluid. This drag results since full atmospheric pressure recovery at the trailing edge of the body is not possible. In contradistinction to a viscous fluid, full pressure recovery is realized at the trailing edge of the body.

The kinetic energy imparted to the inviscous fluid due to the motion of a body through it will be determined by consideration of the concept of impulsive pressure. An inviscous, incompressible, and irrotational fluid having no circulation is set in motion only by the surface-body normal velocity components. The motion of the fluid can be considered as having been produced from rest by a series of impulses applied by the surface of the body. If an element of the fluid is considered:



we have $\left[P - (P + \frac{\partial P}{\partial x}\,dx)\right] \cdot dydz$ = Impulsive Force $= F \cdot dt$

where P is an impulsive pressure.

But $F \cdot dt = mdU$, where m is an elemental mass of fluid and dU is the increment in velocity of the body in the X-direction.

$mdU = mU$, assuming the body to have started from rest.

$mU = \rho dxdydz \cdot U$, where ρ is the mass density of the fluid.

Then, $\left[P - (P + \frac{\partial P}{\partial x}\,dx)\right] dydz = \rho \cdot dxdydz \cdot U$

$$-\frac{\partial P}{\partial x} = \rho \cdot U$$

For a potential flow we have $\frac{\partial \Phi}{\partial x} = U$

$$\rho \cdot \frac{\partial \Phi}{\partial x} = -\frac{\partial P}{\partial x}$$

or

$$P = -\rho \cdot \Phi$$

where ρ is the density of the fluid

Φ is a velocity potential function

The impulsive force, normal to the surface dS, acting on the fluid is $-\rho\Phi \cdot dS$. $\frac{\partial \Phi}{\partial n}$ is the velocity normal to the element of surface dS, where dn is drawn from the body into the fluid. Then, the work done by the impulse = impulse × the average of the initial and final normal velocities in the direction of the impulse:

$$-\int \rho \cdot \Phi \cdot \left(\frac{0 + \frac{\partial \Phi}{\partial n}}{2}\right) \cdot dS$$

Since the work done by the impulse is equal to the kinetic energy imparted to the inviscous fluid we have:

$$T = -\rho/2 \int \Phi \frac{\partial \Phi}{\partial n}\,dS \qquad (1)$$

and therefore

$$M_a = \frac{T}{U^2/2} = \frac{-\rho/2 \int \Phi \frac{\partial \Phi}{\partial n}\,dS}{U^2/2}$$

The additional apparent mass coefficient k is defined as the ratio of the kinetic energy of the fluid to the kinetic energy of the body, that is

$$\frac{T}{\frac{1}{2}M\ (\text{Velocity of body})^2}.$$

The longitudinal additional apparent mass coefficient for translational motion is

$$\frac{T_{x-x\ \text{axis}}}{\frac{1}{2}MU^2} \text{ or } k_1 = M_a/M.$$

The transverse coefficients are

$$\frac{T_{y-y\ \text{axis}}}{\frac{1}{2}MV^2} \text{ or } k_2 = M_b/M \text{ and } \frac{T_{z-z\ \text{axis}}}{\frac{1}{2}MW^2} \text{ or } k_3 = M_c/M,$$

respectively.

In order to illustrate the concept of additional apparent mass, let us consider a sphere. Applying the general expression (1) to the case of a sphere moving through an inviscous fluid it can be shown that

$$T = \tfrac{1}{4}(4/3\pi r^3\rho)U^2 = \tfrac{1}{4}M'U^2,$$

where M' is the mass of the fluid displaced by the sphere and r is its radius. The additional apparent mass of the sphere is, therefore,

$$\frac{T}{U^2/2} = \frac{\frac{1}{4}M'U^2}{U^2/2} \text{ or } \tfrac{1}{2}M'.$$

For the derivation of T for a sphere moving in an infinite mass of liquid which is at rest at infinity see pp. 123-124 of Reference 1. If M be the mass of the sphere, the total kinetic energy of the liquid and sphere considered as one dynamical system is

$$\tfrac{1}{2}(M + \tfrac{1}{2}M')U^2 = \tfrac{1}{2}(M + M_a)U^2 \text{ since } \tfrac{1}{2}M' = M_a \quad (2)$$

This means, therefore, that the effect of the presence of the fluid is equivalent to an addition to the inertia of the sphere of one-half of the mass of the liquid displaced.

When referring to a body having a certain mass, one should not think of a lead body or a body having a concrete mass. The body under discussion is formed by the closed streamline which results from the sink-source distribution selected to represent the particular body. The mass of the body M can then be represented as $Q \cdot \rho$, where Q is the actual volume of the body and ρ is the density of the fluid medium for an incompressible fluid. Since M' is equal to the mass of fluid displaced by the body, M is numerically equal to M' for an incompressible fluid. In order to take into account the presence of the fluid as an item separate from the body alone, the term M' will be maintained in the following discussion.

The additional apparent mass coefficient of the sphere is:

$$k = \frac{\frac{1}{4}M'U^2}{\frac{1}{2}MU^2} = \frac{\frac{1}{2}M'}{M} = 0.50$$

In like manner, the kinetic energy of the fluid due to the motion of a circular cylinder moving through it with its axis perpendicular to the direction of motion is:

$T = \frac{1}{2}M' \cdot U^2$, where $M' = \pi r^2\rho$ = mass of fluid displaced by a unit length of cylinder and r = radius of the cylinder.

In this case

$$k = \frac{\frac{1}{2}M'U^2}{\frac{1}{2}MU^2} = 1.0.$$

Let K = volume of apparent mass.

Then, for the case of the sphere, we have

$$\left(\frac{M + \frac{1}{2}M'}{\rho}\right) = K.$$

But

$$\rho = \frac{M}{Q},$$

where M = mass of the body having the same density as the fluid.

Therefore,

$$\left(\frac{M + \frac{1}{2}M'}{M}\right) Q = K$$

or

$+ k)Q = K$, since $k = \frac{1}{2}\frac{M'}{M}$ for the sphere.

The physical interpretation of this general expression is interest. If we multiply each side of the last expression ρ, we obtain:

$Q\rho + kQ\rho = K\rho$

ρ represents the mass of the body

ρ represents the additional mass which must be added the mass of the body having the same density as the d to allow for the effect of the presence of the fluid; k the additional apparent mass coefficient

ρ represents the total apparent mass of the system ich is composed of the body and the presence of the rrounding fluid

III. Fuselage Moments for a Body of Revolution

The theoretical pitching moment acting on a body imersed in an inviscous fluid can be determined from the nciple of the rate of change of moment of momentum, t is:

$$\frac{d}{dt}\sum(K\rho \cdot U \cdot l) = M$$

ere $K\rho$ = Total apparent mass of the system

U = Velocity of the body

l = Moment arm for the components of momentum

M = Pitching moment of the body

A body moving with a constant velocity U in the X-rection and an angle of attack α possesses momentum th the components $K_1\rho(U\cos\alpha)$ along the principal ngitudinal axis, and $K_2\rho\,(U\sin\alpha)$ along the principal nsverse axis, which is perpendicular to the principal ngitudinal axis and lies in a vertical plane. To determine e rate of change of moment of momentum, we choose a int in the plane of the longitudinal and transverse axes d take moments about the point.

We obtain as the moment M, assuming stalling moments sitive:

$-K_1\rho(U\cos\alpha)\,(U\sin\alpha) + K_2\rho(U\sin\alpha)\,(U\cos\alpha) = M$

$U^2\rho K_2\sin\alpha\cos\alpha - U^2\rho K_1\sin\alpha\cos\alpha = M$

$$\frac{\rho U^2}{2}(K_2 - K_1)\sin 2\alpha = M,$$

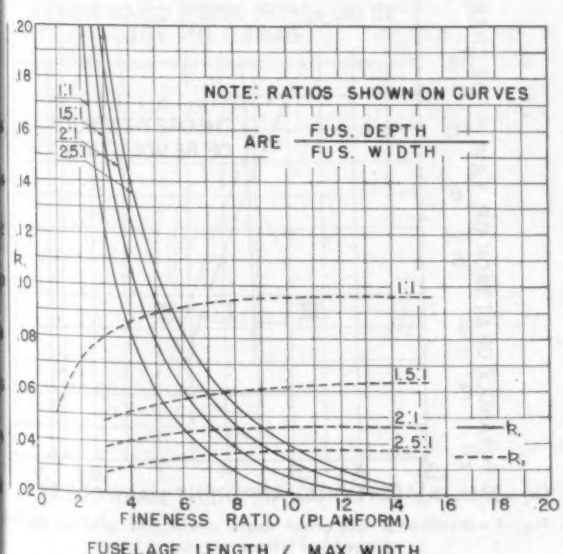


Fig. 1 – Additional apparent mass coefficients k_1 and k_2 plotted against fineness ratio for longitudinal stability

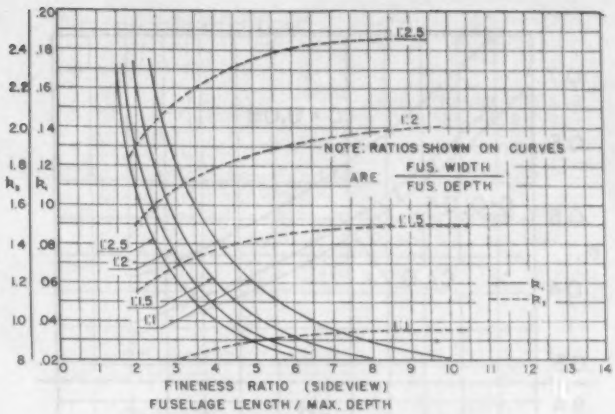


Fig. 2 – Additional apparent mass coefficients k_1 and k_3 plotted against fineness ratio for directional stability

(For a more detailed discussion of this expression see Reference 2.)

where

K_1 = Longitudinal apparent mass coefficient along the principal longitudinal axis

K_2 = Transverse apparent mass coefficient along the principal transverse axis, which is perpendicular to the longitudinal axis and lies in a vertical plane

Since

$(1 + k_1)\,Q = K_1$ and $(1 + k_2)\,Q = K_2$,

we obtain upon substituting in the above equation:

$$M = \frac{\rho U^2}{2}(k_2 - k_1)\cdot Q \cdot \sin 2\alpha \qquad (3)$$

where

k_1 = Additional apparent mass coefficient along the principal longitudinal axis of the body

k_2 = Additional apparent mass coefficient along the principal transverse axis, which is perpendicular to the principal longitudinal axis and lies in a vertical plane

Since $k_2 > k_1$ for a body moving in an inviscous fluid (See Fig. 1) the moment will be an unstable one, since positive moments are stalling moments.

In like manner, the theoretical yawing moment is:

$$N = \frac{\rho U^2}{2}(k_3 - k_1)\cdot Q \cdot \sin 2\psi \qquad (4)$$

where

k_3 = Additional apparent mass coefficient along the principal transverse axis, which is perpendicular to the principal longitudinal axis and lies in a horizontal plane

Since $k_3 > k_1$ (See Fig. 2) the moment will be unstable.

An axis of symmetry of the body is also a principal axis. Lamb[1] has derived the general energy expression of the fluid in terms of 21 coefficients when no stipulation is made in regard to the principal planes of the body. When the three principal axes of equilibrium are considered, the energy of the fluid can be expressed as:

$T = \rho/2\,(K_1U^2 + K_2V^2 + K_3W^2)$

for pure translational flow in a three-dimensional flow

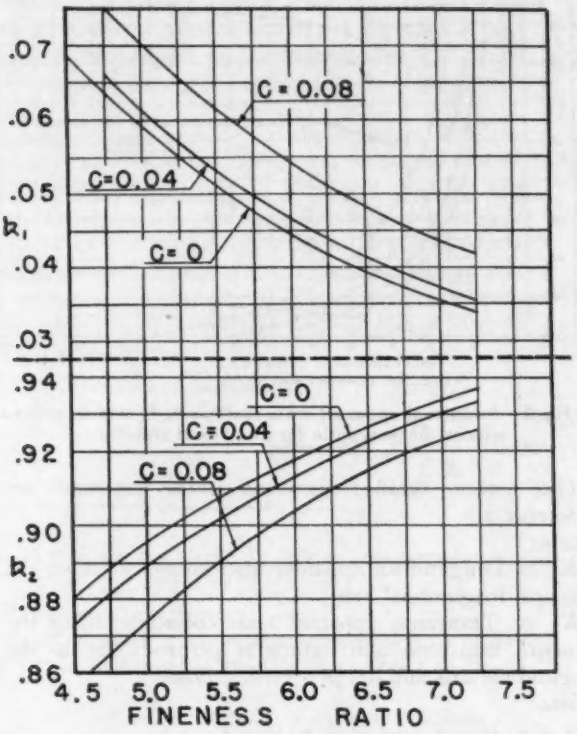
where

K_1, K_2, and K_3 = Apparent mass coefficients referred to the three mutually orthogonal principal axes of equilibrium

U = Longitudinal velocity

V = Transverse velocity in a vertical plane containing K_1 and K_2

W = Transverse velocity in a horizontal plane containing K_1 and K_3



■ Fig. 3 – Coefficients of additional apparent mass k_1 and k_2 for bodies of revolution having various conicities

For a body of revolution, $k_2 = k_3$ since an angle of attack is equivalent to an angle of yaw. Coefficients k_1 and k_2 have been computed by Lamb[1] and are shown in Figs. 1 and 2. The coefficients have been computed for a prolate ellipsoid in which the centroid of the body is located at the mid-point of the length of the body.

For shapes which depart from the prolate ellipsoid, and in particular when the nose section and/or the tail section are so modified that the centroid moves toward the nose of the body, curves of k_2 and k_1 are replotted from Reference 3 and are shown in Fig. 3. In order to define the shift in the centroid of the body from the mid-length of the body, the term "conicity"[3] is introduced and is defined as: $C =$ Distance along the axis of symmetry from the mid-point of the length to the centroid of volume divided by the length of the body.

Fig. 3 shows the effect of conicity on the additional apparent mass coefficients for $C = 0$ (prolate ellipsoid) and $C = 0.04$ and 0.08 for various fineness ratios. It is interesting to note that $(k_2 - k_1)$ for a given conicity and fineness ratio does not differ appreciably for the same fineness ratio and conicity $= 0$.

Smith[4] has shown the effect of a viscous fluid on the longitudinal additional apparent mass coefficient k_1 by taking into account the additional energy contained in the boundary layer on the airship "Akron." The coefficient k_1 increased 35% over that for an inviscous fluid, the value of k_1 for the "Akron" being 0.0534 based on an inviscous fluid and 0.0721 when viscosity was taken into account.

Circular Fuselage – Pitching Moments – To show further the effects of a viscous fluid on $(k_2 - k_1)$, the pitching moments of the "Akron" were computed according to equation 3 and were compared with the moments obtained from Reference 6 taken about the quarter-point of the body length. Table 1 shows the modified values of $(k_2 - k_1)$ determined in accordance with:

$$M = q\,(k_2 - k_1)\,Q \cdot \sin 2\alpha = (C_{M_F})\,q \cdot Q$$

where

$q = \frac{1}{2}\rho V^2$

C_{M_F} = Pitching moment coefficient of "Akron" referred to the quarter-length of the body.

$$k_2 - k_1 = \frac{C_{M_F}}{\sin 2\alpha}$$

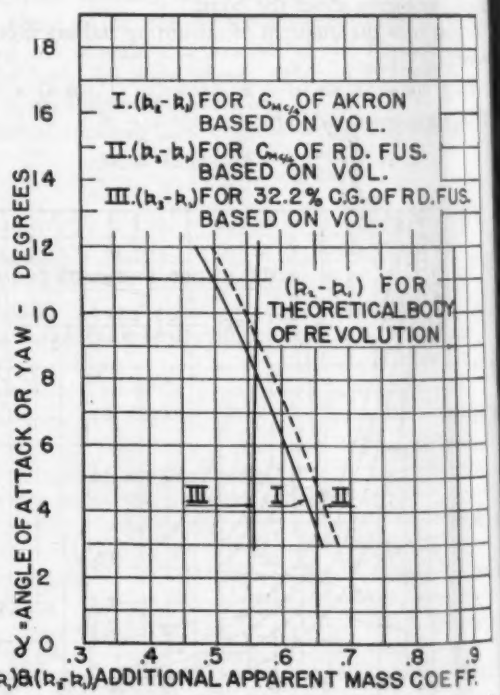
In determining $(k_2 - k_1)$ it should be noted that the angle of attack α also changes because of boundary layer effects. Experiments indicate, however, that the change in α as well as in angle of yaw ψ is quite small. On this basis the entire effect of viscosity will be taken into account in the factor $(k_2 - k_1)$.

Table 1

α	C_{M_F}	2α	$\sin 2\alpha$	$(k_2 - k_1)$
0	0.003	0	0	–
3	0.0685	6	0.1045	0.656
6	0.125	12	0.2079	0.603
9	0.165	18	0.3090	0.534
12	0.191	24	0.4067	0.470

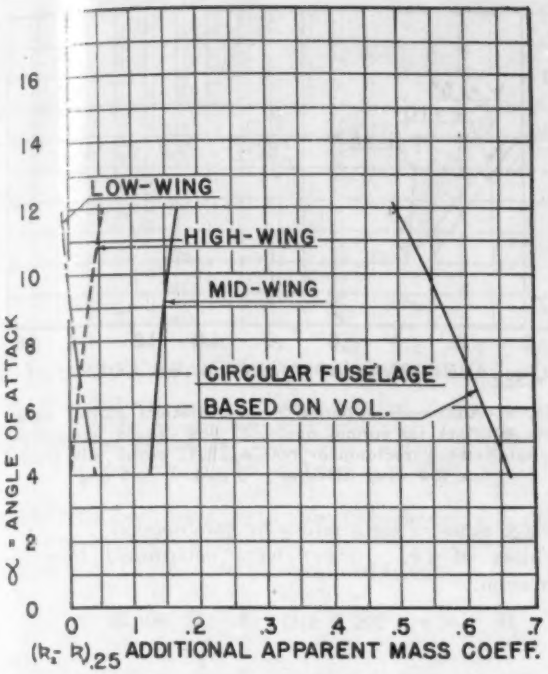
The fineness ratio of the "Akron" is 5.9 and $C = 0.36$. From Reference 3, $k_2 - k_1 = 0.912 - 0.048 = 0.864$ (See Fig. 3)

It is apparent that viscosity and the very large boundary layer effects, especially at the higher angles of attack, have a very pronounced effect on the coefficients. The effect, however, is to reduce the unstable moment of the body.



■ Fig. 4 – Additional apparent mass coefficient plotted against angle of attack or yaw

[6] See NACA Technical Report No. 432, 1932: "Force Measurements on a 1/40-Scale Model of the U. S. Airship 'Akron,'" by H. B. Freeman.



▪ Fig. 5 – Additional apparent mass coefficient plotted against angle of attack for round fuselage plus wing interference – $\delta_f = 0$ deg, tapered NACA 0018-0009 airfoil, tapered fillets

the moment being a function of $(k_2 - k_1)$. Values of $(k_2 - k_1)$ are listed in Table 1 for various angles of attack.

Values of $(k_2 - k_1)$ were also obtained from data given in Reference 7. The fuselage tested was circular in cross-section and had a fineness ratio of 5.9. Values of $(k_2 - k_1)$ against angle of attack are plotted for both the "Akron" and the circular fuselage in Fig. 4. The moments were taken about an axis 25% of the fuselage length.

Circular Fuselage – Yawing Moments – The theoretical yawing moment of a body of revolution is given by

$$N = q(k_3 - k_1) \cdot Q \cdot \sin 2\psi = q(k_2 - k_1) \cdot Q \cdot \sin 2\psi; \text{ since } k_3 = k_2$$

Theoretically, then, the pitching moment of a body of revolution should be the same as the yawing moment. To show that this is practically the case, the data from Reference 8 were used to determine values of $(k_3 - k_1)$. The results are shown in Fig. 4. Curve III of Fig. 4 is obtained by evaluation of

$$k_3 - k_1 = \frac{C_N}{\sin 2\psi}\left(\frac{S \cdot b}{Q}\right) = \frac{\psi \dfrac{dC_N}{d\psi}\left(\dfrac{S \cdot b}{Q}\right)}{\sin 2\psi}$$

where

ψ = Angle of yaw

C_N = Yawing moment coefficient $\left(\dfrac{N}{qSb}\right)$

q = Dynamic pressure $(\frac{1}{2}\rho V^2)$

S = Wing area

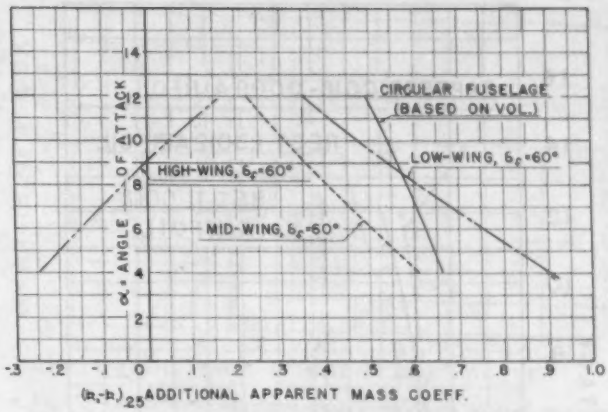
b = Wing span

Q = Fuselage volume

It is assumed that $\dfrac{dC_N}{d\psi} = 0.0006$ for an angle of attack range from 0 to 15 deg (See Fig. 8, Reference 8). This

[7] See NACA Technical Report No. 540, 1935: "Interference of Wing and Fuselage from Tests of 209 Combinations in the NACA Variable-Density Tunnel," by E. N. Jacobs and K. E. Ward.

[8] See NACA Technical Note No. 730, 1939: "Wind-Tunnel Investigation of Effect of Yaw on Lateral-Stability Characteristics. II. Rectangular NACA 23012 Wing with a Circular Fuselage and Fin," by M. J. Bamber and R. O. House.



▪ Fig. 6 – Additional apparent mass coefficient plotted against angle of attack for round fuselage plus interference – rectangular NACA 0012 airfoil, tapered fillets

assumption is not strictly correct since C_N is not a linear function of the angle of yaw throughout the yawing range. As a result, the slope of Curve III differs from the slopes of Curves I and II and as indicated by Curves I and II in Fig. 4, the slope of Curve III should change in sign and magnitude as the angle of yaw is increased. The yawing moments were obtained about an axis 32.2% of the fuselage length. The fuselage was similar to the one tested in Reference 7.

Effect of Wing-Fuselage Interference – Pitching Moments – The NACA has conducted a very thorough series of tests to determine the effects of aerodynamic interference between wings and fuselages at a large value of the Reynolds Number.[7, 9, 10, 11, 12, 13] The program included tests on circular and rectangular fuselages in combination with various wings having different airfoil sections. The quarter-chord axis of the wing was placed at several positions with respect to the quarter-point of the fuselage in the fore-and-aft direction and the vertical position of the wing was varied to simulate high-, mid-, and low-wing combinations. The vertical displacement of the wing quarter-chord axis was measured with respect to the quarter-point of the fuselage. Tests were also made with various wing-fuselage fillets and split flaps deflected 60 deg. All tests were conducted in the variable-density tunnel of the NACA.

The results show that wing-fuselage interference has a very pronounced effect on the magnitude of the fuselage unstable pitching moments. Reference to Figs. 5 to 7 show clearly the effect of interference on fuselage moments.

In determining $(k_2 - k_1)$ no allowance was made for the change in angle of attack over the fuselage because of interference effects and wing downwash. It is believed that most of the interference effects can be properly allowed for by variation in $(k_2 - k_1)$.

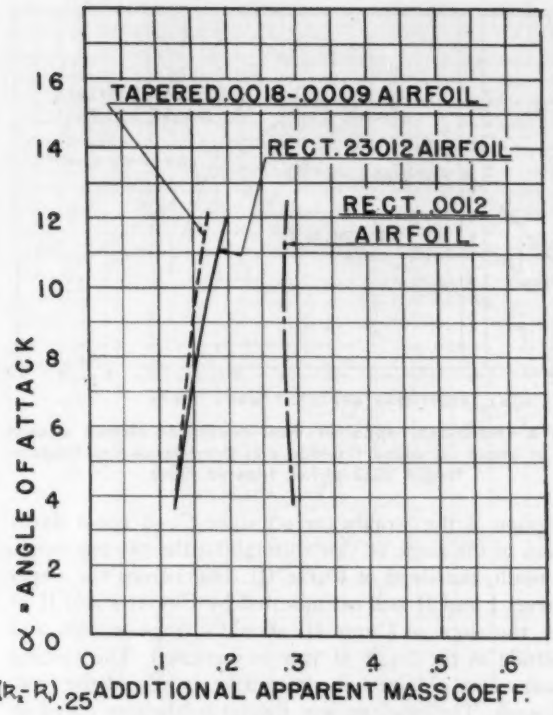
[9] See NACA Technical Report No. 575, 1936: "Interference of Wing and Fuselage from Tests of 28 Combinations in the NACA Variable-Density Tunnel," by A. Sherman.

[10] See NACA Technical Note No. 640, 1938: "Interference of Wing and Fuselage from Tests of 18 Combinations in the NACA Variable-Density Tunnel – Combinations with Split Flaps," by A. Sherman.

[11] See NACA Technical Note No. 641, 1938: "Interference of Wing and Fuselage from Tests of 17 Combinations in the NACA Variable-Density Tunnel – Combinations with Special Junctures," by A. Sherman.

[12] See NACA Technical Note No. 642, 1938: "Interference of Wing and Fuselage from Tests of Eight Combinations in the NACA Variable-Density Tunnel – Combinations with Tapered Fillets and Straight-Side Junctures," by A. Sherman.

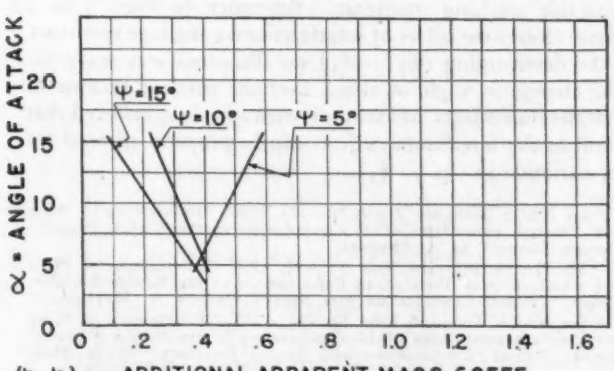
[13] See NACA Technical Report No. 678, 1939: "Interference of Tail Surfaces and Wing and Fuselage from Tests of 17 Combinations in the NACA Variable-Density Tunnel," by A. Sherman.



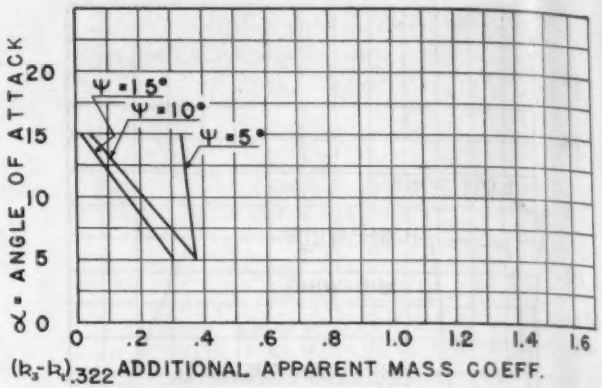
■ Fig. 7 – Effect of airfoil section on $(k_2 - k_1)$ – round fuselage, tapered fillets, mid-wing

No attempt has been made to present a comprehensive study on interference effects. For the investigation made in this paper, the wing quarter-chord axis was chosen to coincide with the one-quarter point of the fuselage in the fore-and-aft direction. Holding the fore-and-aft direction of the wing with respect to the fuselage constant, the wing was moved vertically to simulate a typical high-, mid-, and low-wing combination. The wing for each combination was set at 0-deg incidence with respect to the fuselage centerline.

Fig. 5 shows the variation of $(k_2 - k_1)$ for zero flap deflection for the high-, mid-, and low-wing positions in combination with a circular fuselage, the wing being a tapered NACA 0018-0009 airfoil. Fig. 6 indicates the effect on $(k_2 - k_1)$ when the split flap is deflected 60 deg. Fig. 7 shows the effect of different airfoil sections on the



■ Fig. 8 – Additional apparent mass coefficient plotted against angle of attack for various angles of yaw – round fuselage plus wing interference, rectangular NACA 23012 airfoil with rounded tips, mid-wing, dihedral = 5 deg, $\delta_f = 0$ deg



■ Fig. 9 – Additional apparent mass coefficient plotted against angle of attack for various angles of yaw – round fuselage plus wing interference, rectangular NACA 23012 airfoil with rounded tips, low wing, dihedral = 5 deg, $\delta_f = 0$ deg

fuselage moment for a mid-wing combination.

Values of $(k_2 - k_1)$ were determined from the expression:

$$M = \Delta C_{M_{c/4}} qS\bar{c} = q(k_2 - k_1) \cdot Q \cdot \sin 2\alpha$$

$$-(k_2 k_1) = \frac{S\bar{c}}{Q} \cdot \frac{\Delta C_{M_{c/4}}}{\sin 2\alpha}$$

where

S = Wing area

$\bar{c}$ = Mean aerodynamic chord of wing

Q = Fuselage volume, which can be determined easily by integrating mechanically the area under the curve determined by plotting cross-sectional area against fuselage length

$\Delta C_{M_{c/4}}$ = Pitching moment coefficient of fuselage plus interference of the fuselage in the wing-fuselage combinations about the quarter-chord point of the fuselage given in References 7, 9-13, inclusive, that is, the difference between the characteristics of the combination and the characteristics of the wing alone

α = Angle of attack.

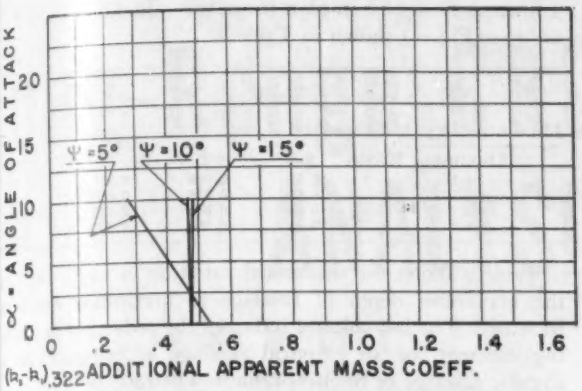
Yawing Moments – An extensive wind-tunnel investigation has been made at the NACA to determine the effect of yaw on lateral-stability characteristics including the effects of aerodynamic interference between wings and fuselages. Various wings and wings in combination with a circular fuselage have been tested.[8, 14, 15] The yawing moments were taken about an axis located 32.2% of the fuselage length from the leading edge of the fuselage. Coefficients $(k_3 - k_1)$ were computed for a circular fuselage of fineness ratio 5.9 and include wing-fuselage interference effects resulting from the 23012 rectangular wing with rounded tips installed on the circular fuselage.[8]

Values of $(k_3 - k_1)$ against angle of attack for various angles of yaw are plotted in Figs. 8 to 12, inclusive. For the zero flap deflection, a mid- and low-wing were chosen, the wing having a 5-deg dihedral. The high-wing results are not shown since the $(k_3 - k_1)$ values were quite erratic at an angle of attack of 15 deg, due possibly to wing stalling.

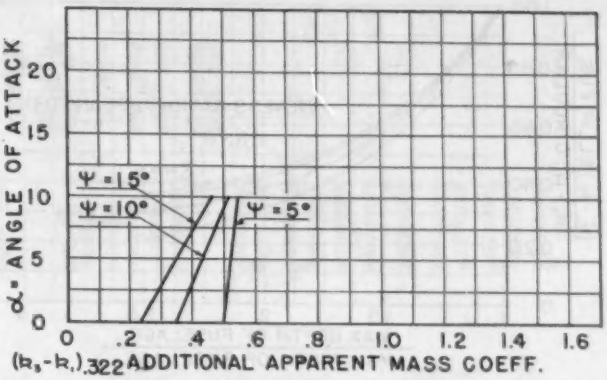
Figs. 10, 11, and 12 show $(k_3 - k_1)$ for the 60-deg split

[14] See NACA Technical Note No. 636, 1938: "The Estimation of the Rate of Change of Yawing Moment which Sideslip," by F. M. Imlay.

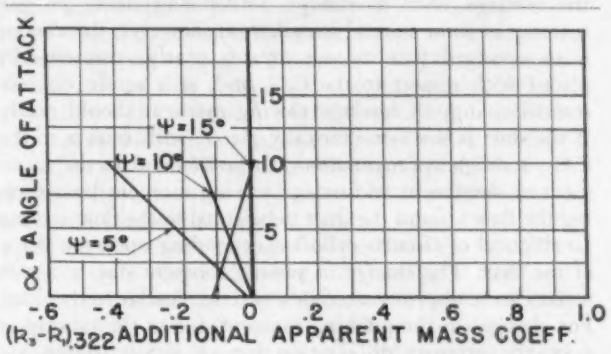
[15] See NACA Technical Note No. 703, 1939: "Wind-Tunnel Investigation of Effect of Yaw on Lateral-Stability Characteristics. I. Four NACA 23012 Wings of Various Plan Forms with and without Dihedral," by M. J. Bamber and R. O. House.



■ Fig. 10 – Additional apparent mass coefficient plotted against angle of attack for various angles of yaw – round fuselage plus wing interference, rectangular NACA 23012 airfoil with rounded tips, high wing, dihedral = 0 deg, $\delta_f = 60$ deg



■ Fig. 11 – Additional apparent mass coefficients plotted against angle of attack for various angles of yaw – round fuselage plus wing interference, rectangular NACA 23012 airfoil with rounded tips, mid-wing, dihedral = 0 deg, $\delta_f = 60$ deg



■ Fig. 12 – Additional apparent mass coefficient plotted against angle of attack for various angles of yaw – round fuselage plus wing interference, rectangular NACA 23012 airfoil with rounded tips, low wing, dihedral = 0 deg, $\delta_f = 60$ deg

flap and for a high-, mid-, and low-wing, the wing in this case having 0-deg dihedral.

Values of $(k_3 - k_1)$ were determined from:

$$(k_3 - k_1) = \frac{Sb}{Q} \cdot \frac{\Delta C_{N_{32.2\,\%}}}{\sin 2\psi}$$

where

$\Delta C_{N_{32.2\,\%}}$ = Yawing moment coefficient of fuselage plus interference of the fuselage in the wing-fuselage combinations given in Reference 8.

ψ = Angle of yaw

IV. Fuselage Moments for an Elliptical Fuselage (Ellipsoid of Three Unequal Axes)

Elliptical Fuselage (Ellipsoid of Three Unequal Axes) – Pitching Moments – The problem of an ellipsoid of three unequal axes has been treated by Lamb,[1] who reduced the problem of finding the additional apparent mass coefficients in terms of elliptic integrals. Munk[16] reduces these integrals to a form which is readily usable:

$$a_o = \frac{2abc}{\sqrt{(a^2 - c^2)^3}} \left(\frac{1}{k^2}\right) \Big[\mathrm{F}(u) - \mathrm{E}(u) \Big]$$

$$\beta_o = \frac{2abc}{\sqrt{(a^2 - c^2)^3}} \cdot \frac{1}{k^2(1-k^2)} \cdot \Big[\mathrm{E}(u) - (1-k^2)\,\mathrm{F}(u) - k^2 \sqrt{\frac{(a^2 - c^2)\,c^2}{a^2 b^2}} \Big]$$

$$\gamma_o = \frac{2abc}{\sqrt{(a^2 - c^2)^3}} \cdot \frac{1}{1 - k^2} \left[\sqrt{\frac{(a^2 - c^2)\,b^2}{a^2 c^2}} - \mathrm{E}(u) \right]$$

where

$a > b > c$ are the semi-axes of the ellipsoid

$$k = \sin \alpha = \sqrt{\frac{a^2 - b^2}{a^2 - c^2}}$$

$$\sin \phi = \sqrt{\frac{a^2 - c^2}{a^2}}$$

$\mathrm{F}(u) = \mathrm{F}(k,\phi)$ is given in Reference 17

$\mathrm{E}(u) = \mathrm{E}(k,\phi)$ is given in Reference 17

The coefficients of additional apparent mass for the three axes are:

$$k_1 = \frac{\alpha_o}{2 - \alpha_o}$$

$$k_2 = \frac{\beta_o}{2 - \beta_o}$$

$$k_3 = \frac{\gamma_o}{2 - \gamma_o}$$

Values of k_1, k_2, and k_3 have been plotted in Figs. 1 and 2. The coefficients k_1 and k_2 are plotted against fineness ratio based on the fuselage length to the maximum width of the fuselage in plan view. Curves for various ratios of maximum depth to maximum width of fuselage are shown in Fig. 1. For the yawed-flight condition, values of k_1 and k_3 are plotted against fineness ratio where the fineness ratio in this case is based on fuselage length to maximum fuselage depth in side view. Curves for ratios of maximum width to maximum depth of fuselage appear in Fig. 2.

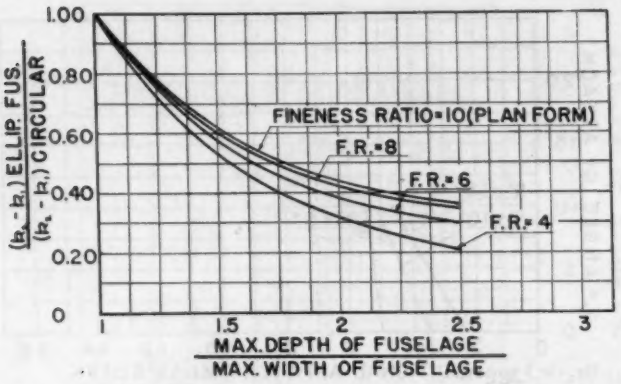
An analysis was made to determine the effect of viscosity and wing-fuselage interference on $(k_2 - k_1)$ and $(k_3 - k_1)$ for an elliptical fuselage, but because of the restricted status of the data the results can not be published.

V. General Discussion

The fuselages that have been discussed so far are rounded-nose bodies, which are commonly used in liquid-cooled engine installations. The question naturally arises as to whether the fuselage moments obtained for a rounded-nose body apply equally as well to a radial-engine installation. Examination of Table II of Reference 7 shows that the rounded-nose fuselage and the radial-engine-cowl installation fuselage moments do not vary appreciably for flow along the longitudinal axis. The radial-engine installation apparently behaves like a flat plate in that the air ahead of the engine piles up to such an extent that an equivalent rounded-nose body results, the air particles moving along a streamline which hypothetically extend the blunt radial-engine fuselage body to a rounded-nose shape.

[16] See "Aerodynamic Theory," W. F. Durand, editor, Vol. I, Division C, "Fluid Mechanics," Part II, by Max M. Munk, pp. 301-302. Julius Springer, Berlin, 1934.

[17] See "A Short Table of Integrals," by B. O. Peirce, Ginn & Co., 1929.



■ Fig. 13 – Variation of $(k_3 - k_1)$ plotted against maximum depth of fuselage to maximum width

The data presented do not allow for any appendages on the fuselage, such as canopy, underslung ducts, or gun turrets. In most normal installations, however, the canopy is so arranged that its side area is nearly symmetrically placed with respect to the C.G. and, as a result, no substantial change in fuselage yawing moment should result. If the duct is not symmetrically placed with respect to the C.G., a rough approximation can be made as to the increment or decrement in fuselage yawing moment by assuming the flow around the duct to be equal to the flow around an elliptical or circular cylinder, depending upon the shape of the duct. The change in yawing moment due to a gun turret can be approximated in a manner similar to the duct. For changes in the pitching moment due to the canopy or duct, the pressure distribution for an airfoil section approaching the appendage can be used. In the case of a gun turret near the tail, the effect on pitching moment should be negligible inasmuch as the boundary layer thickness there is appreciable. For the gun turret in the forward part of the fuselage, an estimate of the change in pitching moment of the fuselage can be made if one performs a strip analysis using data on circular or elliptical cylinders.

If Curve II in Fig. 4 is taken as the average curve of $(k_2 - k_1)$ plotted against angle of attack, it is found that the slope of the curve $\frac{d(k_2 - k_1)}{d\alpha}$ is -0.02 per deg. The experimental value of $(k_2 - k_1)$ at zero angle of attack is 0.75. This value was obtained by extrapolating Curve II to zero angle of attack, where it is believed the boundary layer effects are a minimum. The theoretical value of $(k_2 - k_1)$ for a circular fuselage of fineness ratio 5.9 is $(0.915 - 0.046) = 0.869$ (See Fig. 1 or 3). The correction to be applied to the theoretical value of $(k_2 - k_1)$ is 0.85. If the correction factor of 0.85 is multiplied by the theoretical value of $(k_2 - k_1)$ at zero angle of attack, it is then possible to construct the curve of $(k_2 - k_1)$ against angle of attack by making use of $\frac{d(k_2 - k_1)}{d\alpha} = -0.02$ per deg. On this basis it is possible to determine the pitching moment of a circular fuselage having a fineness ratio other than 5.9. The first step would be to find from Fig. 1 the value of $(k_2 - k_1)$ for the fineness ratio under consideration. Assuming a correction factor of 0.85 at zero angle of attack, the curve of $(k_2 - k_1)$ against α can be found from

$$\frac{d(k_2 - k_1)}{d\alpha} = -0.02 \text{ per deg.}$$

Examination of Figs. 1 and 2 show the marked change in the values of $(k_2 - k_1)$ and $(k_3 - k_1)$ for an ellipsoid of three unequal axes relative to a body of revolution. For a fineness ratio of 8 in plan form, the values of $(k_2 - k_1)$ are (See Fig. 1) shown in Table 2.

Table 2

Maximum Depth of Fuselage / Maximum Width	k_2	k_1	$k_2 - k_1$	$\frac{k_2 - k_1}{0.921}$
1:1	0.95	0.029	0.921	1.00
1.5:1	0.59	0.038	0.552	0.60
2:1	0.45	0.048	0.402	0.44
2.5:1	0.36	0.055	0.306	0.33

Table 2 shows the theoretical variation in $(k_2 - k_1)$ as the maximum depth of fuselage to maximum width is increased. The last column indicates the reduction in pitching moment for an elliptical fuselage as compared to a circular fuselage of fineness ratio 8. The ratio of $\frac{(k_2 - k_1) \text{ elliptical fuselage}}{(k_2 - k_1) \text{ circular fuselage}}$ against $\frac{\text{maximum depth of fuselage}}{\text{maximum width of fuselage}}$ is plotted in Fig. 13 for fineness ratios of 4, 6, 8, and 10 in plan form. For fineness ratios of 4, 6, and 10, the values of $(k_2 - k_1)$ for the circular fuselage are 0.778, 0.875, and 0.939 respectively.

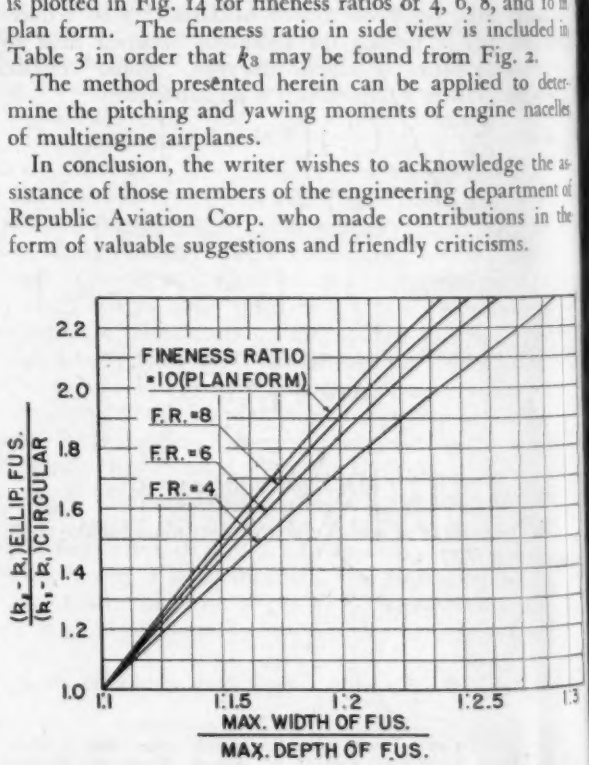
Table 3

Maximum Width of Fuselage / Maximum Depth	Fineness Ratio in Side View	k_3	k_1	$k_3 - k_1$	$\frac{k_3 - k_1}{0.921}$
1:1	8.0	0.95	0.029	0.921	1.00
1:1.5	5.3	1.43	0.038	1.392	1.51
1:2	4.0	1.79	0.048	1.742	1.89
1:2.5	3.2	2.16	0.055	2.106	2.28

Table 3 shows the theoretical variation in $(k_3 - k_1)$ and indicates the rapid increase in yawing moment of an elliptical fuselage as the depth increases. The ratio of $\frac{(k_3 - k_1) \text{ elliptical fuselage}}{(k_3 - k_1) \text{ circular fuselage}}$ against $\frac{\text{maximum width of fuselage}}{\text{maximum depth of fuselage}}$ is plotted in Fig. 14 for fineness ratios of 4, 6, 8, and 10 in plan form. The fineness ratio in side view is included in Table 3 in order that k_3 may be found from Fig. 2.

The method presented herein can be applied to determine the pitching and yawing moments of engine nacelles of multiengine airplanes.

In conclusion, the writer wishes to acknowledge the assistance of those members of the engineering department of Republic Aviation Corp. who made contributions in the form of valuable suggestions and friendly criticisms.



■ Fig. 14 – Variation of $(k_3 - k_1)$ plotted against maximum width of fuselage to maximum depth

Long-Range CRUISING ANALYSIS and CONTROL

by FRANK B. LARY
Wright Aeronautical Corp.

N aviation's rather dim past, engine operators ordinarily fell into one of two classes, high-rpm men and low-rpm en, the latter being the less popular. The way in which pilot operated the engines was largely a function of his rsonal preferences rather than the economy of the air-ane and engine. This is undoubtedly reasonable for ips being operated over short distances with light pay ads. However, like everything else in the industry, size, eight, and payloads of airplanes rose to staggering fig-es. Likewise, of necessity, the art of power control vanced to such an extent that maximum, yes even ex-eme, economy of the airplane-engine propeller combina-on is now obtainable. Even though the numbers for complishing effective power control are available, the ethods by which these numbers were selected are still rouded in mystery to all but a few, and quite unneces-rily. These obscuring shrouds are entirely unnecessary cause the processes used in establishing the operating nditions are basically simple. By definition, long-range uising is: obtaining the maximum ground miles per llon of fuel. It is simply necessary to assure that the rplane features are such that maximum miles per gallon e obtainable, analyze the characteristics of the three ements of the ship, wing, propellers, and engine, to find nditions for maximum miles per gallon, and present the nditions in such a way that the operator may use effec-e control. In all steps, miles per gallon is the keynote.

In the first place, the manufacturers of each element ust make thorough studies of the other's product to sure that the resulting combination may operate at best efficiencies of each part. The power requirements of the airplane for air speeds producing maximum lift-drag ratio must be within the economical power range of the engine. A combination of reduction gear ratio and propeller must be established so that the engine may approach the economical limits of combustion while the propeller produces maximum thrust. Also, adequate engine cooling must be provided at the lean mixtures to be used. These selections must be compromised to some extent by considerations of take-off and climb performance.

With elements thus selected we will concern ourselves with an analysis of the many variables and their effects on miles per gallon of the unit. From this, values of each variable may be defined and presented.

In the analysis it is assumed that sufficient flight tests may be made to evaluate some of the airplane and engine variables. The same study could be made with test-stand engine data and predicated airplane performance. This would, however, introduce greater quantitative errors than would an analysis based on specific flight-test data. The detailed processes in making analyses vary somewhat according to personal preferences and experience. It is believed that some merit is attached to a method which sets up calculations involving all of the efficiencies and reaches mathematical compromises among them instead of studying efficiencies of each component individually.

The flight tests required to furnish an accurate basis for analysis is organized so that only the necessary amount of data is obtained. In addition to the normal complement of flight instruments: tachometers, manifold pressure gages, air-speed indicators, altimeters, and air temperature gages – torquemeters, flowmeters, carburetor air, fuel, and cylinder temperature gages, and fuel-flow measuring tanks are

[This paper was presented at the SAE Northwest Section, Seattle, ash., Oct. 14, 1943; Oregon Section, Portland, Ore., Oct. 15, 1943; d Northern California Section, Oakland, Calif., March 14, 1944.]

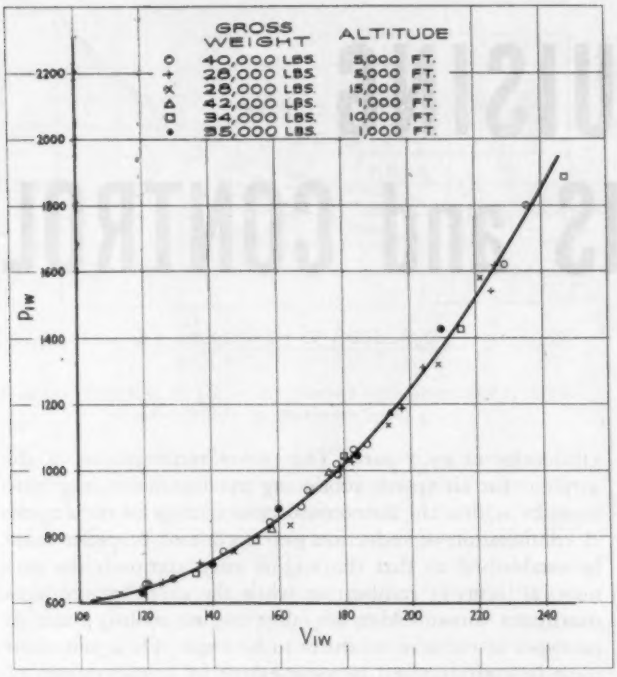
THIS paper embraces an analysis of a hypothetical airplane to determine values of indicated air speed and engine speed and manifold pressure yielding the maximum number of miles per gallon, the criterion of all range thoughts.

Amplification is attempted on the major variables influencing airplane and propeller, and especially engine economy.

A reasonable form for presentation of cruise-control information to the operator is offered with all ramifications for additional simplicity or for refinement.

■ ■ ■

THE AUTHOR: FRANK B. LARY has been with Wright Aeronautical Corp. for five years, during which time he has participated in engine testing, engine development test engineering, engine flight test and field engineering. The last involved engine installation design study in addition to cooling, fuel consumption and performance testing. Before joining Wright he was concerned with aircraft design activity at Southern Aircraft Co.



■ Fig. 1 – Airplane speed-thrust horsepower calibration – standard weight = 35,000 lb

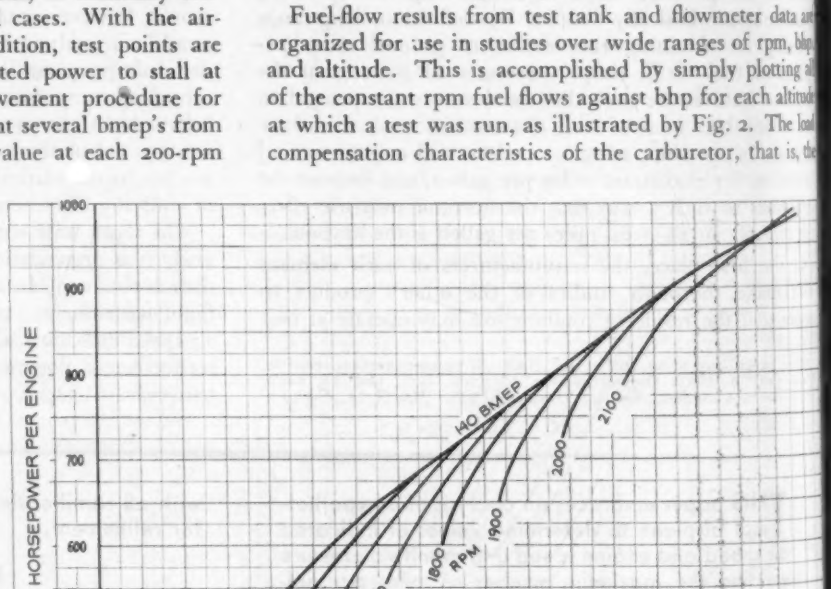
required. All instruments are thoroughly calibrated and an air-speed calibration, that is, speed course or equivalent, is made.

The flight program is arranged to furnish results for an accurate airplane-power-required curve and for cruising-lean carburetor metering characteristics, or manually controlled mixture data in some special cases. With the airplane in a representative drag condition, test points are taken over a power range from rated power to stall at representative gross weight. A convenient procedure for covering this range is to take points at several bmep's from the maximum allowable to a low value at each 200-rpm increment from stall to rated power (the low value is usually 90 to 100 bmep, as dictated by experience, and the maximum allowable is usually 140 to 150 bmep in cruising range). In this way, power-air-speed relationship at various rpm's is obtained as well as fuel-flow data over the entire operating range of rpm's. This complete program is repeated at another altitude and at another gross weight of the airplane. If the drag conditions of the airplane are changed, that is, by addition of external load, the program should again be repeated in slightly abbreviated form. As the cowl flaps act as external drag, particular note should be made of the cooling characteristics expected in the warmest climates in which the ship will operate. If partially open flaps are required to cool within limits in hot climates, this flap setting is used dur ing the above test. Further refinement, if warranted (for instance in commercial operation), may be made by ana lyzing cooling drag plotted against air temperature.

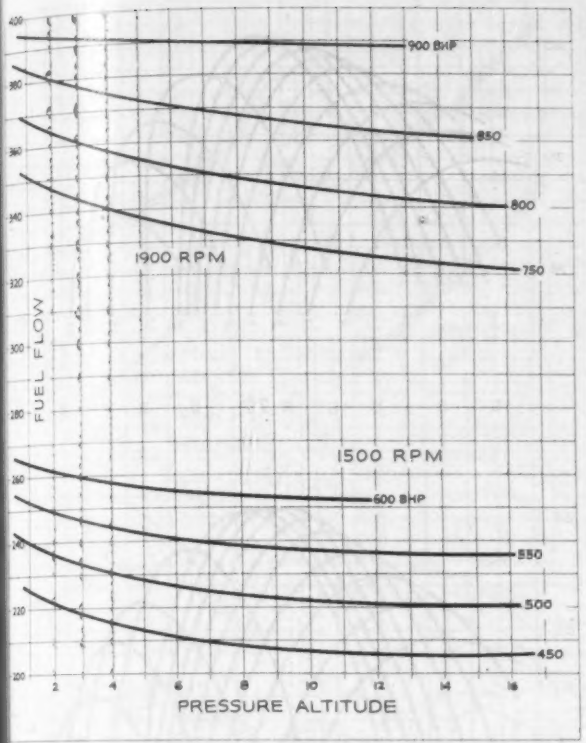
Now, with these data secured, all is in readiness to reduce it to standard conditions; the values having been corrected for instrument error and air speeds for position error. The standard conditions for the airplane are se lected as sea-level altitude and a nominal gross weight. Brake horsepowers, calculated from torquemeter and rpm observations, are reduced to thrust horsepower to allow studies of selected rpm's in the analysis. This is accom plished by computations of propeller power coefficients, C_p, and blade advance coefficients, V/ND, and obtaining propulsive efficiencies directly from a chart. Then thrust horsepower is simply the product of bhp and efficiency, η. The reduction of thrust horsepowers and air speeds to standard is accomplished by modifying both by functions of air density ratio and ratio of gross weight existing on the test at that point to the nominal standard.

The values, ordinarily referred to as V_{iw}, true air-speed weight reduced at sea level, and P_{iw}, thrust power weight reduced at sea level, are plotted to form a velocity calibration as in Fig. 1. By drawing a line through all test points the error variation in flight test air speeds is averaged out. The reduction to standard weight and altitude calculations is checked by noting that all of the points at a specific drag condition, regardless of weight or altitude, fall near the average line. Also the propeller method of reducing various bhp's at several rpm's to thrust horsepower is checked by grouping of test points on the mean line. If the propeller method were inaccurate, the variations in propeller efficiency with rpm at a given air speed would show up in displacing thp points from the mean curve.

Fuel-flow results from test tank and flowmeter data are organized for use in studies over wide ranges of rpm, bhp and altitude. This is accomplished by simply plotting all of the constant rpm fuel flows against bhp for each altitude at which a test was run, as illustrated by Fig. 2. The load compensation characteristics of the carburetor, that is, the



■ Fig. 2 – Fuel consumption at 5000-ft altitude



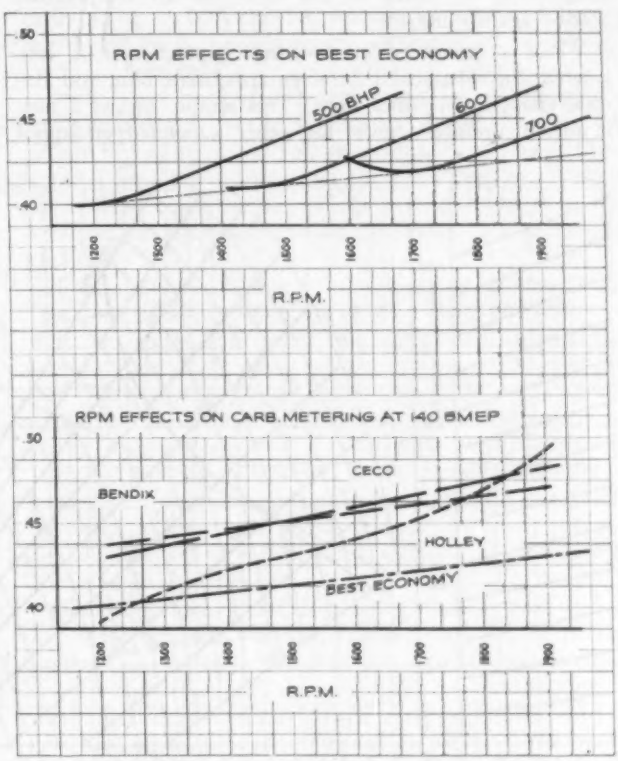
■ Fig. 3 – Fuel-flow variations with altitude at constant power and engine speed

variations in fuel flow at constant horsepower at different rpm's, sometimes change appreciably with altitude depending upon the model carburetor being used: thus the necessity for check plotting fuel-flow curves at the different altitudes. Crossplots are made of fuel-flow results against altitude at constant horsepower for each rpm in the operating range as in Fig. 3, for ease of interpolation between altitudes at which tests were conducted. This fuel-flow variation is entirely an automatic metering characteristic of the particular carburetor, Fig. 3 being an example of the most extreme case. A comparison of the load compensation of three current carburetors and their relation to the minimum specific fuel consumptions for operation of the engine are shown in Fig. 4. Specific fuel consumption, that is, the pounds of fuel consumed per horsepower per hour, is used as the ordinate in each case. The top figure represents the minimum bsfc available at several powers at various rpm's. Obviously there is an rpm range at each horsepower where the minimum bsfc is lowest, which falls at bmep's of 130 to 140. Of course, these points represent the performance we would like from the carburetor but which is not always available, of necessity. The lower part of Fig. 4 shows the minimum bsfc's at 140 bmep and the approximate bsfc's yielded by Holley, Chandler-Evans, and Bendix-Stromberg carburetors at the same bmep and at a fixed altitude. The position of the Holley line varies with altitude by moving to the right at increased altitude, thus the point at which the carburetor delivers a fuel flow at or below the minimum bsfc value for the engine occurs at higher rpm's. This is not too undesirable a characteristic when this is realized on the fuel-flow curves of Figs. 2 and 3, as it will show up as added economy in the analysis *if* the necessary rpm control is applied. Wider throttle openings at altitude and resulting drops in metering diaphragm differential bring about this particular trait in existing Holley designs.

The fuel-flow or specific fuel consumption values are probably the greatest source of quantitative error in the entire long-range analysis. The reason for this is the inevitable presence of carburetor metering tolerances which cannot be eliminated entirely because of inherent mechanical and design deficiencies. The carburetor must provide a given fuel-air ratio for all combinations of airflow into the engine and air density, making the design of the air-measuring and fuel-metering devices a compromise between high and low capacities. In addition to this, the horsepower produced by a given airflow into the engine varies somewhat according to engine life or condition, magneto timing, and valve settings. Thus, even though the fuel-air ratio produced by the carburetor might be exact, the bsfc produced may vary somewhat. Advancement in the art is constantly reducing the automatic metering tolerances. However, the only way, at present, to reduce variations appreciably from original values of fuel consumption is to provide for manual control of mixtures. Of course, this procedure is desirable also in that minimum fuel consumptions may be approached more closely but it would require accurate flowmeters and carburetor air temperature gages or torquemeters in all airplanes and the constant attention of an operator. Both are not always practicable in military aircraft.

Regardless of the fuel metering method, auto-lean positioning of the mixture control with any of the three carburetors or manual leaning to best economy values, the values used in the analysis are taken from the fuel consumption Figs. 2 and 3.

The torquemeter brake horsepower, manifold pressure, rpm, altitude, carburetor air temperature data from the



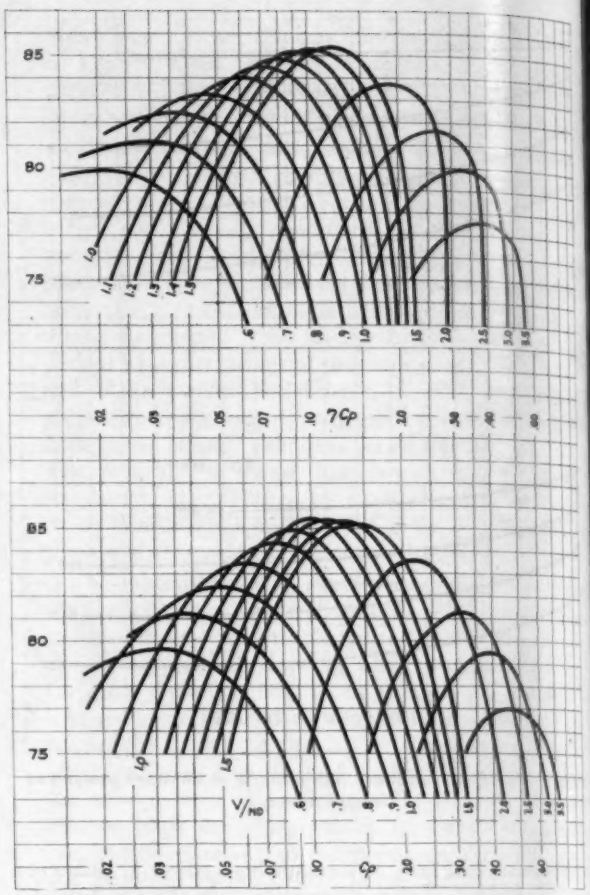
■ Fig. 4 – Comparison of engine minimum specific fuel consumption to carburetor metering

flight tests are used also to provide a flight check on the engine normal performance chart. This chart (similar to Fig. 5) shows the variations of bhp with manifold pressure and altitude (that is, exhaust back pressure), at constant rpm's, at best-power mixtures, and at standard conditions of carburetor air temperature. Best-power mixture is the F/A ratio at which the maximum power is developed at a given manifold pressure and altitude. Therefore, the manifold pressure values of the normal performance chart are raised slightly to allow for the cruise mixtures appreciably leaner than best power. This procedure is accomplished by study of mixture control curves showing the variations in air consumption and therefore manifold pressure with fuel flow at several powers. A good average value of manifold pressure increase for lean mixtures is 1 to 1½ in. of mercury.

The horsepower flight-test results are corrected to standard carburetor air temperature conditions by multiplying the bhp by the square root of the ratio of absolute carburetor air temperature to standard. On full-throttle points, the altitude also is corrected to standard conditions at the carburetor for the effects of temperature and ram on supercharger performance. Thus a cruising manifold pressure chart is drawn up for use in the analysis (Fig. 5). Also on this figure, the engine limitations are plotted against horsepower. This curve is for the limiting bmep, 140 normally, up to an rpm where 140 bmep coincides with a propeller load curve through rated power, about 2000 rpm. Above this rpm the propeller load variation of bhp and rpm are used as limiting.

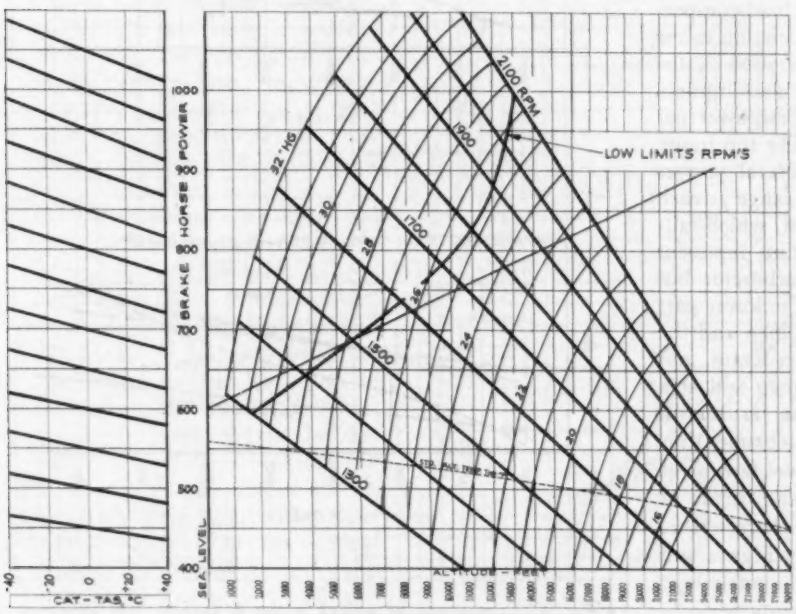
A carburetor air temperature correction chart is applied to the left side of the manifold pressure-horsepower curve so that the variations in power with carburetor air temperature difference from standard is easily obtained. Standard temperature is found on the curve of air temperature plotted against altitude.

When a two-speed or two-stage supercharger is considered, the relationship between rpm, map, bhp, and altitude changes in values only. This simply means a new manifold pressure curve for other supercharged speed ratios is drawn up which moves the full-throttle lines to higher altitudes. Lower limiting bmep's are used at the higher ratios because of the additional power absorbed in the supercharger. Use of a turbo supercharger produces an entirely different problem, inasmuch as the turbine effectively keeps the engine at an altitude or exhaust back pressure which is disassociated with airplane altitude. Therefore, an entirely separate study is made to provide a chart similar to Fig. 5, from which operating map's may be estimated.

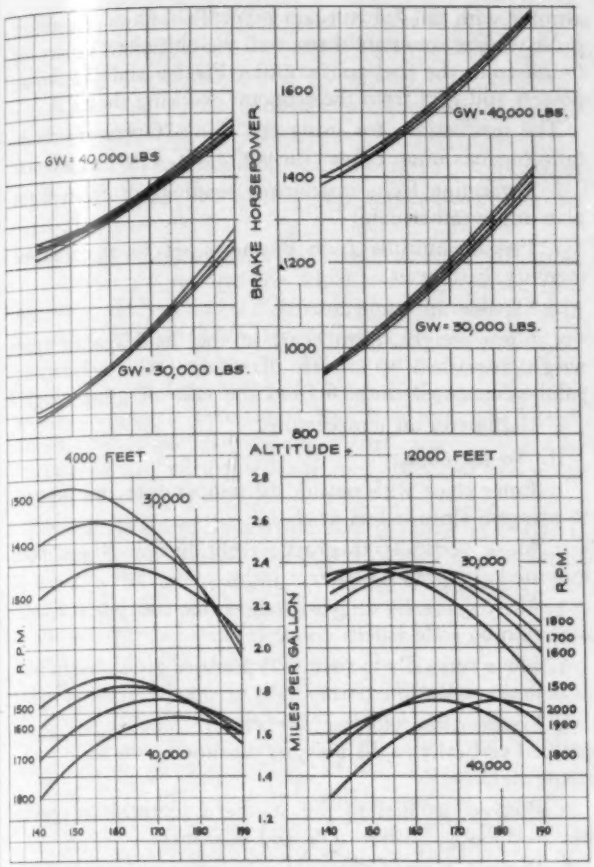


■ Fig. 6 – Propeller characteristics

For the only other item necessary to conduct a range analysis of the airplane, the propeller characteristics are plotted in the form of propulsive efficiency against power coefficient C_p at various values of advance ratio coefficient V/ND (Fig. 6). This will allow many values of thrust horsepower to be converted back to brake horsepower at any rpm altitude, and air speed. The top curve is the one used in reduction of flight-test bhp values for determination of thp, and the bottom curve is simply a conversion of power coefficient C_p to $\eta_i C_p$ so that thrust horsepowers may be used in calculations. By using the same propeller method in the analysis as was used to reduce



■ Fig. 5 – Horsepower-manifold pressure altitude calibration



■ Fig. 7 – Economy working sheet

flight-test data to usable forms of P_{iw} against V_{iw}, the mean total error in studying the effects of the propeller is held to a minimum. Therefore, errors in estimating effects of various operating conditions of the propeller are the only ones that creep into the picture. In this way, the actual values of propeller efficiency are of no consequence; only the changes in these values with different powers, air speeds, rpm's, and air densities influence the final results.

It is possible that the entire foregoing procedure, or parts thereof, may be eliminated with regard to a specific long-range study. In some cases, the aircraft manufacturer has the thrust horsepower-air-speed calibration on hand from previous tests, eliminating this testing and data reduction. However, it is essential that the same propeller method, used by the agent reducing the data as outlined, be used in the analysis based on this calibration. The propeller curves are usually obtainable from the propeller manufacturer or may be drawn up from basic 5868-9 wind tunnel data modified for individual blade characteristics and aerodynamic interference.

Also, enough carburetion flight data may be available at least to supplement, if not eliminate the necessity for specific flight tests. Test-stand results on the particular carburetor and flight-test results in other airplanes may be used to augment carburetion data on the aircraft in question. It is kept in mind, however, that such data may increase the mean total error of results because of differences in effects of airscoops on other airplanes or on the test stand or of engine condition.

At any rate, there now are assembled usable curves on the four items necessary to make a range analysis:

1. The airplane speed-thrust power calibration.
2. The engine-carburetor fuel requirements.
3. The propeller characteristics.
4. The engine power-manifold pressure calibration.

For the actual long-range analysis or study, conditions are selected to cover all conceivable altitudes and gross weights within the scope of the airplane; from sea level to cruise-service ceiling and from weight empty to maximum overload. At each combination of weight and altitude, a number (6 or 8) indicated air speeds are selected to bracket completely the anticipated most efficient air speed of the wing, that is for maximum lift-drag ratio. True air speeds are calculated and reduced to the original standard conditions of sea level and nominal weight, producing V_{iw} for each air speed assumed.

Now, with V_{iw} determined for each point to be studied, values of P_{iw}, thrust horsepower required at sea level and nominal weight, are found from the speed-power calibration, Fig. 1. These power values at standard conditions are then brought up to the particular gross weights and altitudes in question by modifying them according to the weight ratio to standard and air density ratio.

Computations are now in readiness for rpm variations to be considered and injected into the machinery of analysis. Several rpm's are selected, more or less by experience, to bracket an efficient range, say from 140 bmep to 100.

Propeller characteristics are brought into the picture through calculations of V/ND and ηC_p. With these values, the propeller chart (Fig. 6) is used to determine values of propulsive efficiency for each rpm and air speed. Then, of course, brake horsepowers are easily computed for each thrust horsepower at the various rpm.

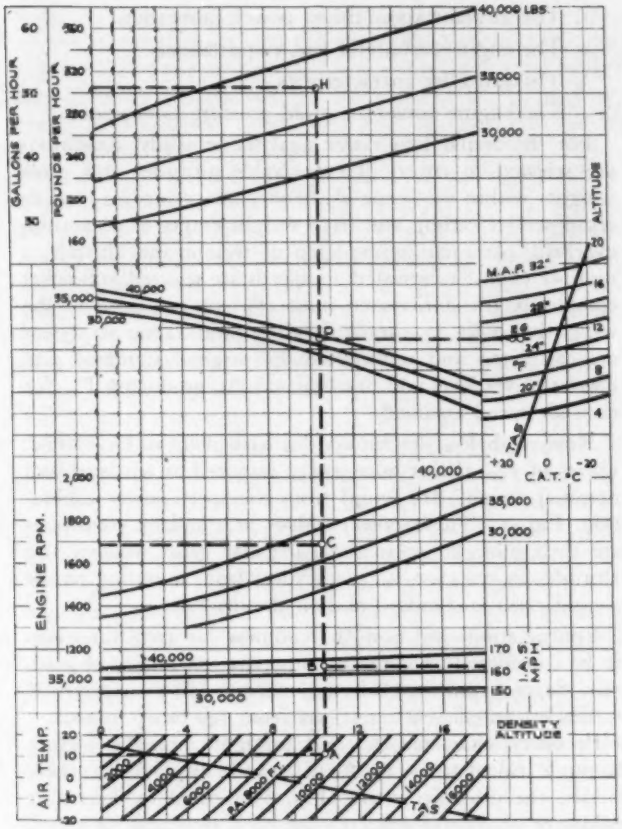
The calculation of these items, V_{iw}, thp, ηC_p, and V/ND appears to be somewhat voluminous. However, tables are easily set up to perform the necessary steps, and constants may be set up for certain groups so that effort involved is kept to a minimum.

With brake horsepowers determined for all the points being considered, the end is in sight. Fuel-flow values are simply picked from the fuel consumption curves, as Figs. 3 and 4 for the bhp, rpm, and altitude. Then by dividing the true air speed by the gallons per hour fuel flow, the criterion of all range studies, air miles per gallon, is obtained.

The results of all these computations are plotted on an "economy working sheet" (Fig. 7). Here the values of miles per gallon at constant rpm are plotted against indicated air speed. Brake horsepowers at constant rpm are also plotted against ias so that values will be readily available. This set of curves is repeated for each gross weight considered and for all altitudes selected. Determination of optimum operating conditions is now simply a question of noting the peak miles-per-gallon points with the corresponding indicated air speeds and engine rpm's and the brake horsepowers necessary to produce that air speed at that rpm in standard air.

This information, and then some, must be presented to the flight operator, pilot, or engineer, in a simple, concise form involving all of the variables he can evaluate from instruments in the airplane. Also, the full-throttle altitude limitations and limiting bmep's must be applied, and in the case of a turbo-supercharged installation, the closed waste gate and surge limits.

Obviously, Fig. 7 is not the chart to present to an operator. He would probably heave it out the window in short



■ Fig. 8 – Operating conditions for maximum-range cruising

order and fly 1800 rpm and 23 in. of mercury. It is quite easy to determine the effects of such action on the airplane economy, for instance at gross weight of 30,000 lb and altitude of 4000 ft. Here 23 in. at 1800 rpm would produce 620 bhp and 191 ias, and the resulting miles per gallon would be 1.8 to 1.9 instead of 2.75 at the peak. This is a loss of 30%. The example is not too extreme either.

Now the peak economy point is picked from Fig. 7 and the corresponding horsepower. The horsepower at the selected rpm is then checked on the manifold pressure-horsepower calibration (Fig. 5) for full throttle and limiting bmep, and the map found. In some cases the optimum rpm is such that the required bhp is not available at full throttle or closed waste gate. Then, the lowest possible is used, and the loss in miles per gallon is suffered.

Now, density altitude is selected as the basis for plotting all other variables. The choice is between density altitude and gross weight. Altitude is chosen because, in this way air temperature variations on airplane performance may be taken into account. This is accomplished by means of a simple chart of pressure altitude lines against outside air temperature (Fig. 8).

In developing the lines of Fig. 8, the air speeds found on the economy working sheet are plotted at constant gross weight for the range of altitudes at which studies were made. The rpm's at constant gross weight also are plotted after checking the power points on the manifold pressure-horsepower calibration (Fig. 5). The manifold pressures at these power points are from Fig. 5. Carburetor air temperature corrections are provided in the manifold pressure determination scale. This is accomplished again by use of a simple chart of carburetor air temperature plotted against altitude with lines of constant map. Fuel flows at constant gross weight are established at all the altitudes. Values are found from the fuel consumption Figs. 2 and 3 at horsepowers and rpm from the economy working sheet.

The instructions for using the chart "Operating conditions for maximum range cruising" (Fig. 8) are as follows:

I. Determine from instrument readings or from flight plan and weather data:

1. Pressure-altitude from altimeter set to 29.92 in. of mercury, barometer.

2. Outside air temperature.

3. Gross weight, from chart of fuel load against gross weight furnished by aircraft manufacturer or subtracting gallons-consumed times 6 from the take-off gross weight.

4. Carburetor air temperature.

II. To use chart:

1. Enter chart with outside air temperature and proceed to pressure altitude line at *A*.

2. Move vertically to gross weight line at *B* and note indicated air speed on scale at right.

3. Move vertically to gross weight line for rpm at *C* and note rpm on scale at left.

4. Move vertically to manifold pressure gross weight line at *D* and mark a line horizontally across the chart on right.

5. Locate carburetor air temperature on this line at *E*, proceed vertically to the pressure altitude on the scale at extreme right at *F*.

6. Proceed parallel to the standard temperature guide line to the original line at *G* and note map.

7. Proceed from *D* vertically to gross weight line for fuel flow at *H* and note pounds per hour or gallons per hour per engine fuel flow on scale at left.

III. To set up power condition:

1. Set governor control to rpm found above at *C*.

2. Adjust throttle to map found above at *G*.

3. Place mixture control in cruising lean position.

4. If, under stabilized conditions, the air speed, corrected for position error, is not equal to the value found above at *B* readjust the map not more than 1 in. of mercury to produce the desired air-speed.

5. Check conditions and reset power every hour of flight.

The chart and its use may be simplified to almost any extent if warranted or if required by lack of instruments, as in Fig. 9.

In the first place, the correction to airplane performance for variation in air temperature could be eliminated for a margin of accuracy. The margin would be greater for greater differences in climates of operation from standard.

Secondly, the slight variations in optimum indicated air speeds with gross weight and altitude may be ignored for a small sacrifice in range. By such action, one ias value may be quoted for all conditions or possibly two; one value for "heavy" and one for "light" weights, as in example *A* of Fig. 9.

Thirdly, the careful control of manifold pressure may be relaxed; the carburetor air temperature corrections being entirely eliminated. Simplification may be carried to the extent that either rpm or map is eliminated as a control item. One or the other would be quoted and the other adjusted at will to maintain the desired air speed.

With all simplifications applied and a table form used

rather than a chart, instructions such as Fig. 9 might result. The instructions for use would be merely:

1. Set up rpm as noted (For example *A*).

2. Adjust map to maintain air speed, do not exceed 31 in. of mercury.

3. Check fuel consumption against hourly tank checks, or

Map values may be quoted and rpm adjusted for air speed, as in example *B*.

In the diametric opposite direction from simplification, there are several refinements which may be applied to a long-range analysis and which have not been considered in this study. In some cases, torquemeters are applied to the engines and bmep gages to the instrument panel. This would allow for the power to be controlled more closely to the specific powers necessary. Also, it is sometimes desirable to provide fuel flowmeters so that mixtures could be manually controlled to predetermined values of best economy for the engine.

Perhaps the finest refinement in long-range technique is the changing of indicated air speeds with the component of wind in the line of flight. A saving is realized by this operation on particularly long flights on aircraft cruising at low air speeds. Determination of the numbers for this action is accomplished by replotting the economy working sheet for ground miles per gallon, which values are found by multiplying the air miles per gallon by one plus the wind velocity over the true air speed; the wind velocity being positive when from the tail and negative when on the head. New peak air speeds will be noticed. They will be higher for headwinds and lower for tailwinds. Thus, new operating condition charts (Fig. 8) would result; say one for a 40-mph headwind and one for a 40-mph tailwind. The effect produced is that of having a longer flight time with a tailwind, therefore, taking advantage of the wind assistance, and *vice-versa*, under headwind conditions.

The desirable degree of complexity in long-range cruise control instructions depends upon the accuracy required, the training of the operator, and the instruments available. In general, control may be more complete on commercial

A

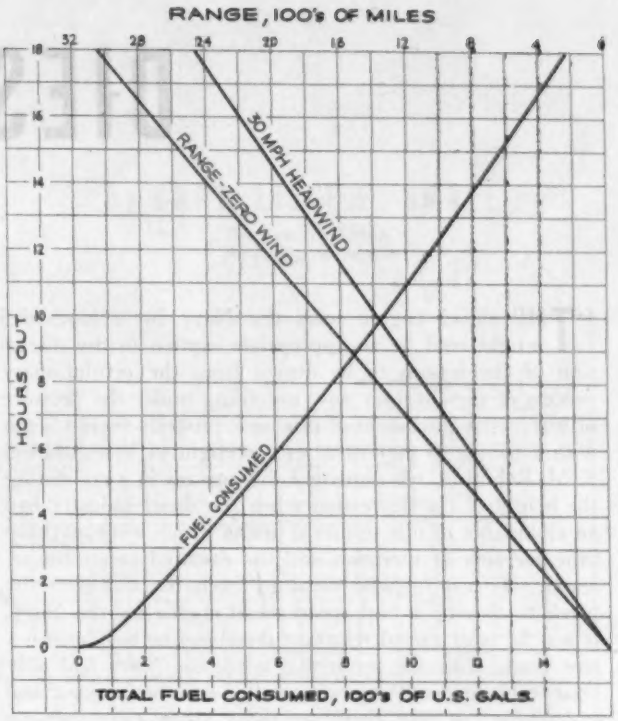
GROSS WEIGHT	LIGHT 150 MPH 28	30	32	34	HEAVY 160 MPH 36	38	40	42	IAS X1000 LBS.
SEA LEVEL	1300 29	1300 31	1320 33	1350 35	1380 37	1410 41	1440 44	1470 47	RPM GPH
5000 FT.	1300 31	1320 34	1350 37	1390 40	1430 43	1480 47	1540 51	1600 55	RPM GPH
10000 FT.	1420 35	1460 38	1500 41	1540 44	1590 47	1630 51	1670 55	1740 59	RPM GPH
15000 FT.	1600 39	1640 42	1690 45	1740 48	1800 52	1860 56	1920 61	1980 66	RPM GPH

B

USE 155 MPH I.A.S.

GROSS WEIGHT	LIGHT 29	33	HEAVY 37	41	X1000 LBS.
SEA LEVEL	28 30	28 34	29 39	29 46	IN MAP GPH
5000 FT.	26 32	26 39	27 45	27 53	IN MAP
10000 FT.	24 36	24 43	25 49	25 57	IN MAP
15000 FT.	21 40	21 47	22 55	22 64	IN MAP GPH

■ Fig. 9 – Simplified cruise control



■ Fig. 10 – Long-range cruising "how-goes-it"

airline ships than on military, whereas instructions for military ships must usually cover much wider ranges of operation types. The reasons for this situation are the high degree of training of the few commercial operators who have time to devote to accurate power control and with only two types of operation. This is compared to the wide variety of missions involving timing for military craft with operators far too busy to devote their entire time to power control.

In addition to power control instructions, there is one more bit of information to be supplied to the operator, that to allow him to check his fuel consumed and miles covered immediately. This is sometimes deemed a "how goes it" (Fig. 10). Here the total fuel consumed, or fuel yet available, is obtainable at any number of hours out of the starting base. Also, the range covered and available at various winds at a specific altitude is obtainable at a glance. This chart is repeated at several altitudes and starting loads of the ship, and for single- or three-engine operation. From all these charts, the operator can determine whether he should go ahead, or proceed to a nearer base in the event of abnormal engine functioning.

In conclusion, we note several consistent characteristics of long-range operation.

First, maximum range is obtained by operation at just about constant indicated air speed regardless of altitude. For variations in gross weight, a value on the high side of the average ias is optimum for heavy weights and on the low side for light weights.

Second, maximum range is obtained at sea level regardless of the weight. In a few special cases of light-power leading aircraft, this is not entirely true because of the fact that the powers required are in the idle enrichment range. This condition is rare.

Third, cutting out one or more engines on a multi-

concluded on page 221

DIESEL ENGINES in WA

"THE diesel engine wins the Navy by a landslide" might well be an appropriate caption to my discussion of the lessons to be drawn from the revolutionary process of dieselization now unfolding under the pressure of war. Any discussion of this topic properly should begin with a tribute to the vision and foresight of Vice-Admiral S. M. Robinson, who initiated this process in 1932 during the height of the depression when the diesel industry had an abundance of idle technical brains which were marking time for lack of direction and the essential funds for research. It was he who obtained from the Congress the funds to develop a high-speed diesel engine for the Navy. It was he who started reluctant diesel-engine builders on a new track, with the result that while the Navy had only 150,000 hp of diesel engines installed in submarines at the end of the last war, there are today about 12,000,000 hp installed in a variety of applications extending from the mighty 45,000-ton battleship Iowa to the most humble tugboat.

It truly can be said that Detroit gave the diesel engine its new lease on life. It has been my happy privilege to be in the Bureau in charge of the Internal-Combustion Engine Section at the crucial period of time when C. F. Kettering was invited by Admiral Robinson to undertake a part in this new era which eventually led to the dieselization of American railroads as well as the Navy. I look back on those days with a great deal of satisfaction because it is a historical fact that much was accomplished.

In addition to the engines which the Navy developed to suit its many needs, it inherited a large collection of all existing types of engines when substantial numbers of various kinds of merchant ships were taken over for conversion. Today we have the job of operating many slow-speed engines of the principal foreign makes as well as all domestic types of engines. This is a very interesting phase of engineering replete with complicated maintenance problems, but of necessity it must be considered beyond the scope of this discussion.

[This paper was presented at the SAE Diesel-Engine and Fuels & Lubricants Meeting, Cleveland, June 2, 1943.]

■ Two-Cycle and Four-Cycle Engines

The war has ended the old quarrel of two-cycle versus four-cycle engines. It has ended in a draw. We need both types of engines somewhere, therefore, all engine builders capable of producing reliable engines are fabricating them to the utmost of their capacity. We have hundreds of Gray marine engines (General Motors model 6-71) installed in the same type of hulls and doing the same work as other hundreds of four-cycle Hercules engines. All ships' power boats are propelled by four-cycle Buda-Lanova type engines – some 8000 have been built by Buda and the Norfolk Navy Yard jointly. All new submarines are powered by two-cycle engines. Incidentally, the Germans paid us a compliment by developing for the propulsion of their submarines a copy of the General Motors V-type engine, hundreds of which have been built right here in Cleveland.

The Bureau uses a very simple diagram, the rectangular coordinates of which are brake mean effective pressure and specific volume, that is, piston displacement per horsepower developed. On this diagram (shown in Fig. 1) are superimposed a series of hyperbolic curves representing the rpm of the engines. Any engine can be represented by a spot on this diagram which will closely define its thermo-

DIESELIZATION of the Navy has progressed rapidly under the pressures of war. At the end of the last war, the Navy had only 150,000 hp of diesel engines installed in submarines, while today about 12,000,000 hp is installed in a variety of ships both large and small.

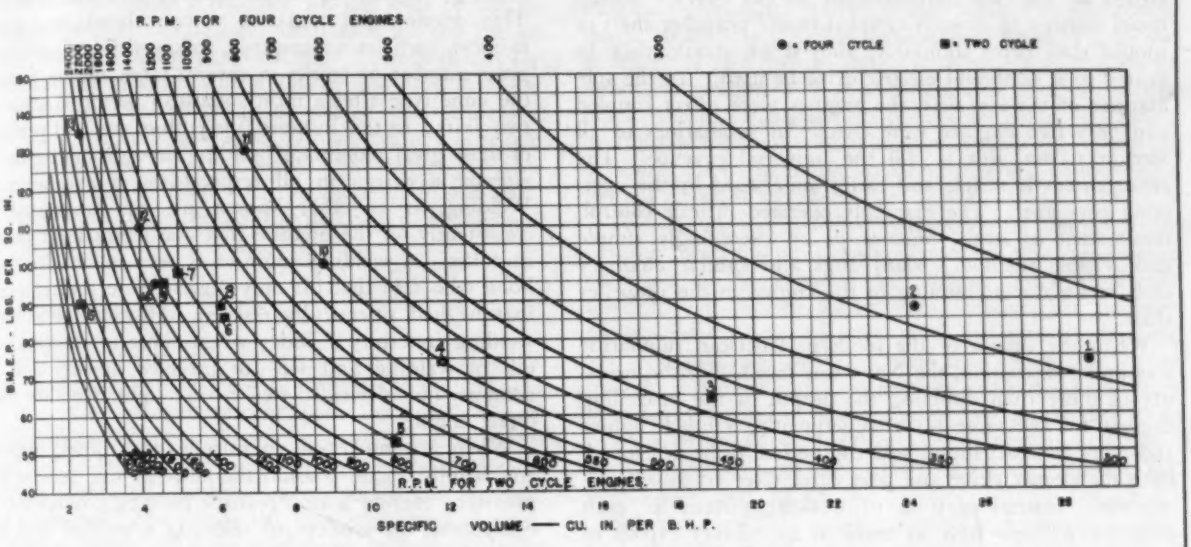
The reasons for the tremendous increase in the number of diesel engines used by the Navy are discussed by Capt. Small by means of a diagram showing how improvements have been made in 15 diesels, the earliest developed during the last war and the latest being the new pancake engine.

Capt. Small also discusses some of the problems that have been studied in the development of diesel engines, among them being: direct coupling of the propeller shaft with the engine versus the indirect drive; welded steel construction; torsional vibration of shafting; spare parts; and gas turbines.

THE AUTHOR: CAPT. LISLE F. SMALL, who is now assistant head of the Shipbuilding Division of the Bureau of Ships for machinery planning and procurement, began his career in submarines after graduating from the U. S. Naval Academy, when he served in the Battle Fleet during World War I. Shortly thereafter he graduated from the Submarine School in New London, Conn., and several years later, in 1925, received his M.S. degree in internal combustion engines from Columbia University. There followed numerous assignments with submarines which culminated in a three-year tour of duty as engineer officer, Submarine Forces. Since 1932 most of his service has been in connection with Bureau assignments and navy yards. During his last sea duty, which ended in 1940, he was engineer officer, Destroyer Force in the Battle Fleet.

WARTIME NAVY

by CAPT. LISLE F. SMALL
Bureau of Ships,
Navy Department



■ Fig. 1 – Diesel-engine development diagram

dynamic perfection by the bmep it develops to carry its load, its mechanical development by the rpm, and its weight and space requirements by the piston displacement per horsepower. In other words, this diagram indicates very closely the evolution of the diesel engine. I have plotted on this diagram some of the historically important engines which cover the period between the last war and today. I hope the diagram demonstrates the fundamental reasons for the dieselization of so large a part of the Navy. I think you will agree that it supplies the reason for the decision that argument of two-cycle versus four-cycle is a draw. Certainly it demonstrates that the engineer is a rational being somewhat devoid of sentiment, but in the final analysis responsive to the promptings of reason.

Engine No. 1 propelled the bulk of American submarines during the last war; it developed a bmep of 74 psi at 380 rpm and required 28.75 cu in. of piston displacement. It weighed 68.5 lb per hp. It is still on the firing line and doing a pretty good job. Engine No. 2 represents the highest development of World War I German submarine engines. It developed a bmep of 87.5 psi at 380 rpm, required 24.7 cu in. piston displacement and weighed 53.3 lb per hp. Incidentally, certain prototypes of this engine are doing their bit today in American submarines. So far as we can ascertain no German U-boats have them. Both of these engines were the four-cycle air injection type. No. 3 is the main engine of our first fleet submarine built shortly after the war – two-cycle, bmep of 63.7 psi at 325 rpm, requiring 19 cu in. displacement and weighing 63.7 lb per hp. No. 4 represents the main engine of the German pocket battleship – two-cycle double-acting, rated at 74.4 psi bmep (excluding scavenging blowers) at 450 rpm. Here the specific volume drops to 11.8 cu in. and the weight is 28 lb per hp. This is the first example of the welded steel frame. No. 5 is one of the German warship engines transplanted to this country under license. No. 6 is the original General Motors model 201 engine produced by George Codrington in Cleveland, which became the prototype of a number of engines now working in the Navy and on U. S. railroads, its latest development being No. 7.

No. 8 is the original and No. 9 is the present state of another two-cycle, opposed piston type of engine by Fairbanks Morse now used extensively in the Navy. No. 10 represents another product of the George Codrington dynamic personality. It is a four-cycle engine designed to do its utmost for the type. Later it was equipped with a Buchi turbo charger and performed as No. 11. No. 12 represents a recently completed four-cycle engine with a mechanically driven supercharger. No. 13 is another engine with the type of supercharger which represents the highest level of development, a considerable number of which are installed in special power boats. No. 14 is one of the engines used in great quantity in the Navy ships' boats. No. 15 is the famous pancake engine.

The lesson to be drawn from this brief review is self-evident. The inherent simplicity of the four-cycle engine and its ability to perform with a minimum of attention will

continue to be appreciated by the operating personnel. When equipped with a supercharger it still offers an attractive solution. On the other hand, the fact that the two-cycle engine has double the number of power strokes and, therefore, is bound to be lighter and more compact fortifies it with certain inherent advantages which account for its prominent trend of development to date.

■ Direct and Indirect Drives

Another argument of long standing is definitely being settled by the war requirements in the Navy. Should diesel engines be directly coupled to the propeller shaft or should they drive indirectly, and, if so, should they be geared to it or should electric drive be used? In the submarines of the last war, the engines were direct coupled and only two engines were used. In submarines of all foreign navies, this is still the universal practice. The arrangement is simple and easily understood by the operating personnel. The manually operated throttle control, reminiscent of steam practice, is an exceedingly simple and reliable method. Many tugs and similar auxiliary craft retain direct coupling of the engine to the propeller shaft.

When the design of the modern American submarine was under discussion, the Navy was faced with the necessity of more than doubling the output of the propelling engines for the same hull; consequently, a much lighter and more compact high-speed engine had to be developed. Both d-c electric drive and gear drive were employed successfully. Instead of 6 to 10 working pistons for each propeller we now have as many as 40. Every exploit of our submarines in enemy waters, often 5000 miles away from home bases, testifies to the reliable performance of the indirect drive used in them. There is no equivocation on that point. There is no doubt that the higher level of intelligence of our submarine crews contributes greatly to the success achieved. We are building ships to be operated by American boys who in civilian life rode in automobiles, who operated tractors and trucks, and who are not bewildered by internal-combustion engines. Without question, we enjoy a favored position in this respect.

The successful use of indirect drive in submarines supplied the impetus to make similar application to many powerful auxiliary ships. In certain instances, a-c electrical drive was used, and the results were most gratifying. The maximum number of diesel generators per shaft employed so far has been four. There is no reason why that number cannot be increased widely, maintaining comparable simplicity. Multiple-engine geared drive has been used very successfully. Likewise, a combination of electric and gear drive has given excellent results. All of these arrangements can be classified simply as a multiengine indirect drive. This application has much inherent merit, particularly in regard to flexibility. It permits one and the same type of engine to be used in propelling plants two, four, eight, and so on, times its output, resulting in real mass production of a comparatively small number of standardized units. And this much must be emphasized, a multiple diesel-engine drive provides the opportunity to maintain high fuel economy over the entire operating power range. This is extremely important in the Navy because our vessels must run at speeds determined by tactical considerations.

One of the most important contributions of the Navy to the development of the high-speed diesel engine was its firm insistence on welded steel construction. From the very beginning, the Bureau realized that the lightweight structure could be obtained either by using castings of metals of low specific weight or by the selected method of welded steel. The art of welding was in its swaddling clothes when the decision had to be made. Later events replete with the phenomena of shock have testified to the wisdom of the decision. Fortunately, when war came, welding technique and welded frame construction had been perfected to the point that production at a high rate could go forward up to the capacity of available facilities. This development was not without headaches and uneasiness, such, as for instance, followed the discovery of a great number of cracks in several large engine frames of the same type after a most successful performance of over two years. In this instance, the enthusiasm and persistence of those great pioneers of welding brought out on the next attempt a design in all respects sound and acceptable. Welding, as no other known art, has progressed most consistently on the premise that one learns most from his mistakes. A welding engineer is very much like the old hobo who said that his only regrets were the temptations that he had successfully resisted. However, I salute the welding fraternity. Today we are sure of the fundamentals of welding and we can produce quickly and with certainty out of welded steel a very light structure for the diesel engine.

When war broke out in Europe, and mines, torpedoes, and bombs began to assail the ships of our present allies, providing thereby a most realistic proving ground for their equipment, the wisdom of selecting a welded steel structure for our new diesel engines could no longer be questioned. Bitter experience proved that because of its large elongation under stress, steel is the only readily available material that can resist successfully the shocks imposed by war.

■ Vibration

In or out of war, there is one form of vibration which bedevils all of us. No one will have to guess that I mean torsional vibration of shafting. Our experts in this department ascribe nuisance value to it largely because it makes a problem out of every new combination of an engine with its load-absorbing device. Propriety doesn't permit me to speak with nautical frankness on this subject. Based on some recent experiences, I feel that nothing has ever been done so successfully before as nothing has been done in the last five years towards cleaning up this mess. Someone may like to edit the word "mess," but the dictionary defines "mess" as "a confusing and embarrassing situation," and, therefore, the word stays "put." We started out years ago to do something about torsional vibrations and good progress was made. Then the war started in Europe and we began to get production minded to the exclusion of everything else. We should never have side-tracked the subject of torsional vibrations. It is distinctly an industry matter and it cannot be handled by individual manufacturers. I beg of this Society to grab the torch because, through a combination of carelessness and "status quo" mindedness, troubles from torsional vibrations are appearing in places ripe for adverse advertising.

No current story on diesel engines would be complete without mentioning critical materials. The Navy's efforts

to overcome problems presented by the shortage of certain critical materials are very familiar, although not pleasantly so, to all who supply Naval machinery. Not so long ago, the Bureau of Ships suddenly found itself not the preferr d customer for nickel, which is so essential for armor making; aluminum, most of which is needed for aircraft production; tin, which the war has made more precious than the hoarded gold in the Kentucky hills. We are fortunate indeed that the Mesabi Iron Range is sufficiently extensive so as to provide us with steel and iron, and that the Joplin district of Missouri can supply us with lead and zinc. It is the plentiful native materials to which we must turn to supply our basic needs. Whether we will ever return to the more familiar materials of the past is for the future to show, but to call the new materials recently forced on us substitutes is professionally unfair because lead makes a very satisfactory bearing material, welded steel structures are far more attractive economically than castings, whether iron or aluminum, and God knows that women welders are easier to look at than the iron moulder of the past, who surely must have inspired the make-up of commando troops of this war.

Bottlenecks

Just a word now about a subject that is on everybody's mind – bottlenecks. War does funny things. What used to be the name of a convenient gripe for a little liquid stimulant is now used to describe a manufacturing nightmare. Shortly before, and immediately after Pearl Harbor, extensive additional facilities were authorized for the diesel industry. The expanded diesel production eventually demonstrated weak links of the industry as a whole. Number one worry, which has never receded from the foremost position, was forgings, particularly crankshafts. The forging fraternity has performed admirably, however, if that industry is going to survive, it must cut down on the excess metal in black forgings. A modern improvement in this direction will save millions of manhours, and it must be made. If certain of our leading engine manufacturers had not standardized on cast crankshafts, I am convinced that our position in the present war would be much less favorable today than it is. I mention this merely as a commendation for the vision on their part, and I allude in no sense to the merits of either cast or forged crankshafts. Thank God we had both kinds.

Today we are seriously worried about the available manufacturing capacity for bearings. The aircraft program imposes heavy demands for bearings and the automotive industry has suffered accordingly. The Bureau of Ships has been in constant touch with the bearing manufacturers and I must report their position is critical. They need a few kind words and less unpleasant invectives.

Spare Parts

The most widely discussed subject that comes to my ears today is "spare parts." The operation and maintenance of such vast numbers of engines by the Navy all over the globe presents a serious problem of providing spare parts for them. An imposing forty million dollar spare-parts center was created at Mechanicsburg, Pa., and an extensive service organization was set up there. In accordance with available-use tables, staggering quantities of spare parts were ordered. It was soon discovered that a too ambitious spare-parts program would interfere with the production of new engines and two things had to be done – the delivery of spares had to be reduced and the demand for spares lessened by sending competent trouble-shooters into the field to educate the operating personnel in the vital need for conserving spare parts. For years every garage man, every overhaul mechanic has been trained to squander spare parts. Those men now are in uniform in Africa, Iceland, and Attu. The shoe is on the other foot now and it is pinching like mischief. We are paying very dearly for the unique misconceptions created when spare parts were glibly spoken of as worth their weight in gold. They are worth their weight in gold today, but in an entirely different metaphorical sense. I hope we weather the spare-parts storm. It is here and it presents some grim problems. Two approaches to relief offer fairly quick returns. First, salvage centers are being established in conjunction with distribution centers. The operating personnel are being taught to cannibalize engines damaged beyond repair or stranded.

The second approach lies in the training courses at the various diesel-engine schools. In most of these courses emphasis has been placed on training the student how physically to replace worn parts. Seldom has he been shown real examples of worn parts that must be replaced and real examples of worn parts that should not be replaced. We must inculcate in these students the burning need for conservation of spare parts. We must reach them by some method and give them the will and the confidence to use wearing parts until they begin to impair the safe operation of the engine.

While I am on the subject of schools, I want to pay the highest tribute to the many companies who have fostered schools at their factories. Training centers have been established in many colleges throughout the country, in the plants of most engine builders – Cleveland Diesel Division boasts a very good one – in Miami, on Chicago Pier, and so on. These schools have been well equipped with engines and engine accessories. Their greatest assets are the instruction books published by the machinery manufacturers. Instruction books of the most comprehensive kind, well printed and profusely illustrated, comparing favorably with standard text books, have been furnished by most engine builders. Navy suppliers have been competing for excellency in this respect for years, and this competition has paid high dividends. A good instruction book is a public record of pride. It is a human sort of an instrument because it makes the operator feel he has been taken into the confidence of the manufacturer. Our training program would have been hopelessly impotent without these fine documents.

While training of personnel is of the utmost importance, it must be recognized that of equal importance is the process of continuously testing all the different engines installed in Naval craft under controlled conditions and expert supervision. The Bureau of Ships does this at the Engineering Experiment Station at Annapolis. Here scores of engines of all types undergo life tests to determine their endurance and any necessary improvements before they are released to the Service. The Experiment Station is fully apprised at all times of new operating conditions that the engines are expected to encounter in service. Special research on bearings, piston rings, lubricating oil, and the like, is continuously going on there to reduce wear and

concluded on page 224

Aircraft-Engine INLET and EXHAUST PO

IN the layout of a new engine or revamping performance of existing designs, the designer is confronted, among a multiplicity of other problems, with the task of selecting proper port dimensions not only to fulfill performance requirements but also to result in reasonably good durability of the parts directly affected by the porting. The porting is interpreted as meaning not only the port proper but the entrance or seating, valve, guide, guide boss, and so on.

Since these basic dimensions generally dictate in a large measure the head design, they should be known at the inception of the layout. Much depends on the choice of proper dimensions at the seating end and if some basic constant could be established, not worked out from theoretical consideration but determined from actual performance data, this phase of the problem should be simplified. It should be pointed out that no attempt is made to consider other factors that influence the breathing and exhausting of the cylinder – such as valve lift, timing, and effect of manifolds – as it is considered these are of such nature that experimental work can be carried out quite readily, whereas modification of cylinder heads to revise port dimensions is a major change, and while highly desirable it involves too many changes and in the majority of instances just is not done.

From a dimensional point-of-view it is desirable to keep the ports as small as possible to permit flexibility in obtaining proper sections to withstand thermal and hydraulic loading, to obtain maximum coolant flow (whether liquid or air), to maintain reasonable temperatures, and also to obtain best flow characteristics of the passages in the head, which control the flow of inlet and exhaust gases.

Velocity through the valve is commonly calculated using the following formula:

$$V_v = V_p \left(\frac{B^2}{D^2}\right)$$

where

V_v = Velocity through port
V_p = Piston velocity
B = Cylinder bore diameter
D = Port diameter

Since this formula considers only the basic data – such as cylinder diameter, stroke, speed, and port diameter – it is obviously reduced to the simplest form possible. However, the result expressed in mean gas velocity in feet per second does not necessarily mean these are the velocities that are to be dealt with, especially when considering that if instantaneous velocities are to be considered, then the gas velocity would be proportional to piston velocity. Velocity varies as the square root of the pressure drop, and a study of pressures in the cylinder and port indicates that there are so many other factors influencing this they are not necessarily proportionate to the piston displacement, porting, manifold, timing, and so on, all affecting this. Furthermore, when considering exhaust pressure where the differences are of a much higher order, velocities are again misleading. However, since the factors of speed, bore, and stroke are the only ones that are known when making the original layout, rearrangement of these was considered to establish a valve port constant. By using a port constant, there is no misleading reference to velocities which can only be determined from extremely accurate indicator cards where the pressures are measured close to the so-called orifice.

To establish any constant presupposes knowledge of existing data, and that the data are of sufficient accuracy to make them applicable to new designs. All of the data considered were taken from existing and proved engines and should be truly representative. It was decided to establish a constant as representing the slope of the port area and a displacement-speed factor; so the port area is plotted as ordinate and the factor of speed-displacement as abscissa. ($\frac{N \times D}{2000}$ where N equals crank revolutions per minute and D equals cylinder displacement.) This in reality can be said to be just another way of expressing velocities excepting, as mentioned before, it is expressed as a constant, which it is believed can be more readily handled.

The port area is based on the cylinder-seat small diameter (see sketch, Fig. 1) rather than the valve or throat, since this dimension is more truly representative of the actual effective opening. No allowance is made for stem diameter as it is apparent from a study of existing engines where the stem diameters vary considerably, that it has little or no effect on the flow; and it is believed a casual examination of the sections through the port with the valve sketched in explains why this is not effective.

A plot of a number of representative engines (Fig. 1) shows that a line having a slope of 0.025 passes through the field of inlet ports without too much scatter; and a slope of 0.018 takes in the majority of the exhaust ports. A lower line with a slope of 0.0165 might be termed a safe low-limit slope, being based on actual engine data where smaller than average ports have been tested with no appreciable effect on performance[1]. From these slopes

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[1] See SAE Transactions, Vol. 51, September, 1943, pp. 334-343: "Intake Systems for Aircraft Engines" by Carl T. Doman.

PORTING

by VINCENT C. YOUNG

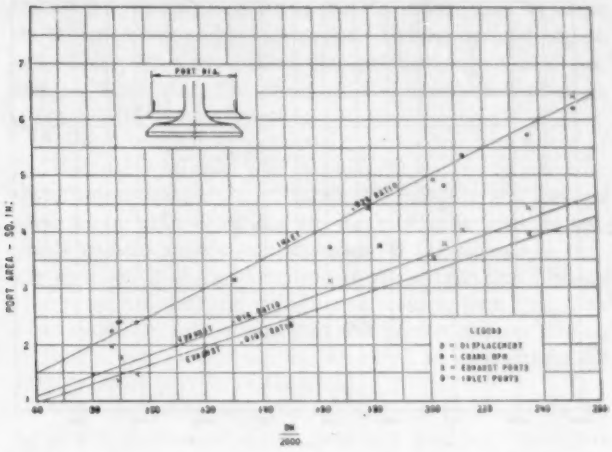
Chief Engineer, Wilcox-Rich Division, Eaton Mfg. Co.

it can be stated that if the original port dimensions are based on the displacement, speed, and area (using a factor of 0.025 for the inlet and 0.018 for the exhaust) two basic dimensions required are established.

Since volumetric efficiency is a measure of the engine's ability to breathe, it, if compared with the port areas, should be a good criterion as to the proper port size. The volumetric of several representative engines is plotted in Fig. 2 against the inlet constant, and in Fig. 3 against the exhaust constant. Volumetric data are all at a definite manifold pressure and with normal rates of speed, which accounts for the tendency of the ports to lie on the high side of the port constant. The shape of the curve, however, suggests there is little to gain by increasing port area beyond the 0.025-in. ratio, although it does suggest that this is a minimum figure to use.

Fig. 3 indicates that increasing the exhaust area over and above that which would result in a constant of 0.018 in. has little, if any, influence.

The question has been raised many times about the influence of the size of the exhaust port on operating temperatures of the valve. This is an exceedingly difficult one to answer intelligently due to so many variables influencing resultant temperatures; and a survey of field results suggests that these variables obscure the true picture as to effect of size, but it can be stated that if, by the adoption of a port size towards the minimum, the factors influencing cooling, leakage, and so forth, are improved, then the net result is a cooler operating valve.



■ Fig. 1

It might be thought that the exhaust could be a definite percentage of the inlet, but in the event, due to some constructional difficulty, the inlet constant is of necessity less than 0.025, then using a percentage factor would

ONE of the many problems confronting the engine designer is that of selecting proper port dimensions.

As a general rule, it appears desirable to keep the ports as small as possible. This procedure permits flexibility in obtaining proper sections to withstand thermal and hydraulic loading, gives maximum flow of coolant (whether liquid or air), makes reasonable temperatures possible, and gives the best flow characteristics of the passages in the head, which control the flow of inlet and exhaust gases.

To simplify design, Mr. Young uses a port constant, which represents the slope of the line obtained by plotting the port area against a factor of speed displacement – just another but more handy way of expressing velocities.

The cooling of the exhaust valve is discussed by Mr. Young as affected by heat transfer from the head section to:

1. The relatively cool new charge.
2. The heat transferred to the coolant through the seat.
3. The heat transferred to the coolant by the stem.
4. The heat transferred to the port gases during the other than exhaust portion of the cycle.

Maximum flow coefficients will be obtained for intake ports, if the approach passages are of uniform section with no re-entrant curves and the actual passage shape between the seat and the valve is considered as a venturi.

Cooling requirements for the inlet port are not as stringent as for the exhaust, so that guide and boss can be designed with emphasis on flow characteristics rather than on cooling effect.

■ ■ ■

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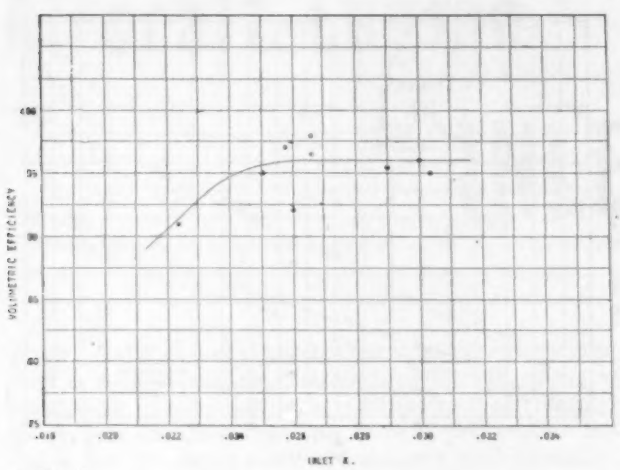
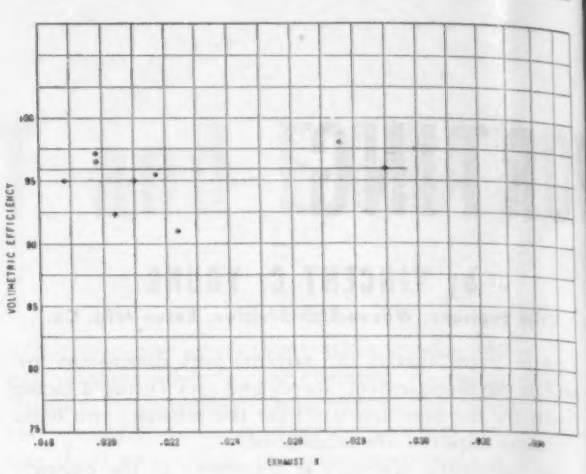


■ Fig. 2



■ Fig. 3

reduce the exhaust to the area where improper scavenging might result.

It was considered that possibly a modification of the factors should be made with respect to varying bore/stroke ratios, but from a study of engines within the range of conventional design where the ratio varies from 0.792 to 1.12, it does not seem to have any bearing and other than to mention it no data were developed with respect to the effect of bore/stroke ratio on desirable ports of engines.

It is interesting to note that while no attention was given to segregating engines into groups – such as high and low speed, naturally aspirated or supercharged, and with all the other possible variables – the results of the plot indicated the slope to be reasonably constant.

As mentioned, it is highly desirable to keep valve port areas to minimum dimensions, and as outputs increase this becomes of increased importance, especially with respect to the exhaust since the gas loading at the instant of opening generally increases with greater outputs and this seriously affects the valve gear, as the loading increases as the square of the exhaust valve head diameter.

■ Effect of Port Design on Valve Cooling

The cooling of the exhaust valve is generally considered to be affected by the heat transfer from the head section to:

1. The relatively cool new charge.
2. The heat transferred to the coolant via the seat.
3. The heat transferred to the coolant by the stem.
4. The heat transferred to the port gases during the other than exhaust portion of the cycle.

1. Heat Transfer through Relatively Cool New Charge – The ratio of fuel to air in the charge has a marked effect on the operating temperature of the valve, in that during the admission stroke with a fuel/air ratio on the rich side, considerable cooling is effected by fuel evaporation, and the temperature at the opening of the valve or the end of the expansion stroke is lower than with a fuel/air ratio of nearly correct proportions.

Observations of valve temperatures with varying fuel/air ratios indicate that even with relatively low outputs the variation from a fuel/air of 0.1 to 0.07 increases the valve temperature from that at which little difficulty would be experienced to a temperature that would result in relatively short life. Another way of expressing this would be in terms of output, certain tests showing that with a fuel/air ratio of 0.1, twice the horsepower could be delivered with the same valve temperature as when operating with a fuel/air ratio of 0.07.

Another factor affecting cooling which might be mentioned is the influence of overlap of the inlet opening and the exhaust closing, since if a portion of the fresh charge is wasted it has considerable cooling effect.

It is desirable to operate with the most economical fuel/air ratio that will not result in a rapidly deteriorating effect on engine parts, particularly with long-range equipment; and since it is definitely established that operating temperatures are increased with decrease in fuel/air ratio, the cooling effect of the charge will tend to reduce rather than increase, indicating that we must look forward to a loss rather than a gain in cooling from the charge effect.

2. Heat Transfer through the Seat – The heat transferred through the seat is of a highly variable nature, and too much emphasis cannot be given to the importance of careful design study of this portion of the cylinder head.

The question arises many times as to what percentage of heat absorbed by the valve is transferred to the coolant via the seat, and it is extremely doubtful if even an intelligent guess could be made as to this percentage. The seat itself, in many instances, is made of a material having poor heat conductivity and is installed in such a manner that while relatively new and well fitted, soon loses this fit and builds up an insulating skin which further reduces its heat conductivity.

The relative sealing of the high-pressure gases is the major variable that affects the ratio of heat transfer through the seat to the coolant, and this sealing can vary from nearly perfect to the type that will permit sufficient leakage to cause an extremely rapid build-up of temperature.

The factors affecting sealing are the relative change in cylinder-seat location with respect to the guide and the flexibility of the valve permitting or resisting movement that corresponds to the change in cylinder-seat location.

Practically all cylinder-head material is aluminum, and it is necessary to utilize some other material for the insert. The choice of material and design has considerable influence on the ability of the seat to maintain its true relationship with the guide. The insert should be considered as having a dual function in that its first and primary duty is

to maintain a seat face that will afford proper sealing and, second, to act as a stiffening member of the head structure.

Due to uneven temperature distribution and a steep temperature gradient through the insert, it is subjected to severe uneven stresses, which are of the creep type, and to avoid subjecting the insert to over-stressing it should be of a material that has a high creep strength. Its shape should be such that it tends to minimize the effect of the thermal gradient, which is maximum at the seat face, reducing toward the counterbore wall and away from the seat face proper. The initial stress should be of a low value so that the total of initial, thermal, and distortion loads will not exceed its creep strength.

The material surrounding the insert should be as uniform section as possible and cooled uniformly. If the coolant is liquid, there should be a wash sufficient to prevent incipient bubbles from building up to sizable ones that actually create dry areas which result in localized high temperatures. When air is the coolant, the fins should be so located as to give uniform temperature distribution to maintain circularity under severe operating conditions and with the minimum-length heat path from the guide to the fins.

Stresses set up by hydraulic and thermal loadings as well as those from distortion, resulting from bolting, barrel installation, and so forth, are of a complex nature making it difficult to predict their effect on the seating surface, and the ring design should attempt either to resist these loads or minimize their effect. The ideal seat would be one that is located so it always maintains alignment with the guide.

The combustion-chamber face is subjected to the maximum temperature and effect of hydraulic loading, and this suggests that the ring or insert be of sufficient cross-section to support itself inherently and installed so as to minimize the loading from the combustion-chamber-wall movement. To accomplish this the attachment, as it might be termed, should be located in a position that is the least affected by distortion which is the portion of the port away from the chamber wall.

The design that fulfills the above requirements as closely as is practical, takes on the shape of a relatively heavy flange attached to a thin wall sleeve; the heavy portion being installed without initial stress, and the thin sleeve with the minimum stress that will hold it in position under conditions of metal temperatures that result in the minimum interferences.

The insert material that possesses the best physical properties is generally one having a relatively low thermal heat conductivity, and since it is located in the path of the hot exhaust gases and a certain percentage of the heat from the valve is being transferred through it by conduction through the joint to the head wall and thence to the coolant, it operates at a temperature well in excess of the surrounding material. To compensate for this difference in temperature, the coefficient of expansion should be of an appreciably lower value than that of the head material.

Measurements and observations of inserts definitely indicate a tendency to collapse inwardly but not uniformly as a result of uneven stress distribution; such measurements of the change in shape definitely indicating the existence of stresses higher than the material is capable of withstanding.

Unfortunately, when trouble, as evidenced by high temperatures, results, the insert itself very seldom shows any signs of distress and consequently its contributing effect is overlooked.

The valve is a definite part of the porting and should be included at this point to bring out the influence of its shape on the operation and performance. The variance in engine output requirements makes this problem again complex, and to study this, the effect of the variables should be given careful consideration.

At high output, as previously mentioned, the effect of the thermal, hydraulic, and mechanical loading as affecting distortion is maximum. To offset this partially, the fuel/air ratio is, in most instances, on the rich side with its tendency to help reduce temperatures. If the change in seating as a result of the distortion is of the type that a slight change in valve shape will permit sealing, then the valve head should be of a design that will permit the gas loading to force it into contact and create a seal, at least during the high-pressure portion of the cycle.

At the lower outputs, the loadings are reduced, but not in direct proportion since generally fuel/air ratios are reduced, resulting in less cooling effect, and if the distortion is still of sufficient magnitude to cause leakage, the decreased pressures will not force the valve into intimate contact with the seat and the leakage becomes destructive; the remedy for this condition being a reconsideration of the insert design and installation. If at the lower outputs the distortion is not sufficient to result in destructive leakage, then the so-called flexible valve shape will operate satisfactorily.

The valve shape that possesses considerable inherent stiffness or resistance to deflections under high gas loadings, must operate in conjunction with a seat that approximates under all operating conditions the desired shape. Since these factors can only be determined by actual tests, this phase in reality becomes a problem of the test rather than the design group, but it should be considered in the design study and included in the test program.

3. *Heat Transfer via the Stem* – The part of the design affecting the heat transfer through the guide has been covered quite thoroughly in the past; however, the writer believes it is of such extreme importance and has been overlooked or minimized in many instances, that further mention should be included in a paper of this type.

The design of the portion of the port that has a marked influence on the operating temperatures is one that necessarily must be a compromise. To obtain the best flow characteristics the guide boss should be so located as to present the least interference to the flow of gases; but if a design of this type is followed through, the heat path from the valve head to the coolant via the stem becomes so great that the head and neck portion of the valve operate at destructive temperatures.

Observations and study of valves operating at high outputs indicate that the heat-flow path is becoming the limiting factor and very careful consideration should be given to the design to reduce this tendency.

In order to maintain a reasonable temperature gradient and take full advantage of the cooling effect through the guide boss, it should be located as close to the seat as possible. Naturally, if it is so located as to present considerable interference with the gas flow, then performance is affected and thus the location becomes a compromise.

It is difficult to specify exactly how the best compromise design can be accomplished, but if certain characteristics are given consideration then the problem should not be too difficult. Generally the cross-sectional areas of the varying locations, starting from the seat and going towards the port exit, are laid out and if the areas appear small they

are increased without too much thought as to the effect on flow, and ofttimes an increase in area can result in poorer flow characteristics.

The following are suggestions offered to the design engineer for his consideration when laying out the exhaust porting: If areas can only be increased by the use of re-entrant curves, then this area should not be increased, as the change in shape generally results in a turbulent flow which actually affects the flow characteristics to such an extent that the benefit of the enlarged area is more than offset and the net result is a loss rather than a gain.

Since considerable pressure is available, particularly at the beginning of the exhaust stroke, the areas through the cross-sections, including the guide boss, can be reduced in relation to the clear port area provided these areas reduce gradually or somewhat similar to a venturi tube where flow quantity is maintained with increased velocity.

When considering the section where the boss blends into the port wall on the side opposite the exit passage, extreme care should be exercised so as to avoid removing metal at the blend, which would result in the boss protruding into the gas stream and in so doing present a greater volume of uncooled or poorly cooled metal with the only apparent gain being that of increased areas in the sectional layout.

The coolant, if liquid, should be carried as close to the seat end of the boss as possible and around circumferentially to the fullest extent. When air is the coolant, the section should be of generous proportions to reduce the tendency of the metal to follow the temperature cyclic changes, with cooling fins so located as to abstract the heat input at the boss at the maximum rate.

4. Heat Transfer to Gases – Of this effect little is known but it is believed that a safe assumption to make is that if there is any heat transfer it is from and not to the gas; hence the necessary cooling effect must be via the coolant. The heat from these gases is generally not of a destructive nature, as examination of many valves operated at temperatures with head surfaces affected, show very little, if any, difficulty on the underside.

■ Intake Ports

Little need be said regarding the flow characteristics of the intake ports since considerable work has been carried out in the past that suggests certain definite recommendations to obtain maximum flow coefficients.[2] A general statement might be made that if the approach passages are of uniform section with no re-entrant curves and the actual passage shape between the seat and the valve is considered as a venturi, then generally the highest overall coefficient will result. However theoretically perfect are the shapes to promote flow and increased volumetric efficiency, if these shapes are not maintained, difficulty will arise in actual operation that will definitely influence performance.

Similar to the exhaust, the major section that is affected is that pertaining to the seating and consequent sealing of the valve face. This portion of the port is influenced not only by the factors that influence the exhaust, but by the characteristics of the exhaust port sections, in that if their sealing is poor the resultant increase in nonuniformity of temperatures will result in further destruction of the inlet seating, thus creating leaks that affect the volumetric as well as the durability of the parts involved and while it is not always necessary to carry out the design to the extreme required for the exhaust, nevertheless the same general thoughts should apply.

The cooling requirements are by no means as stringent as the exhaust, so that the guide and the boss can be designed with emphasis on flow characteristics rather than on cooling effect; but this factor should not be neglected.

In addition to the suggestions offered as an aid in the original design, it seems fitting to include a few remarks as to interpretation of the results, as regardless of the efforts spent in the design stage, certain modifications are generally required – these being necessitated by possible manufacturing difficulties or some unpredicted stresses resulting in local failures.[1]

Presuming that the performance indicates the desired results have been obtained as far as breathing capacity and scavenging are concerned, the next, and of prime importance, is the durability of the parts. The direct indicators of durability are generally the valves and knowledge of exact operating condition, and the appearance of the valve generally provides sufficient evidence to determine whether or not the design is one that will result in satisfactory life. If, however, this portion of the development work is not carried out in definite steps the picture is such a composite one that it becomes difficult to analyze properly.

Cylinder-seat leakage reveals itself by the appearance of the valve face, but this can be obscured by the action of the valve gear. Generally, at the speeds of maximum output the valve has a tendency to turn and the rate of turning is highly variable. With the valve contacting its seat properly, the face temperature is below that which will be destructive, but if unseating occurs, it not only loses the possibility of heat transfer but is subjected to higher than normal heat inputs since a portion of the gases at high pressure and temperature pass directly over it. If the rate or type of turning moves a given section of the face over and past the leakage area before it reaches a destructive temperature, then the temperature rise that does occur is immediately reduced when full contact is again made.

To obtain a better picture of what actually occurs it is preferable to run under the given operating conditions for a reasonable period with the valve locked against rotational movement. From a study of the valve face appearance and physical data – such as relative relation of the valve face with the stem, change in shape of the contact face that might be the result of high impact seating as well as misalignment of the insert face caused by distortion, type and location of corrosion – a fairly clear deduction can be made as to exactly what is going on during the run. This type of test should be repeated under the varying operating conditions and a careful analysis of the results should definitely indicate the necessity for further design study.

Operating temperatures can be determined by the installation of thermocouples but this is generally a difficult piece of work and, furthermore, only gives the temperature at the 'couple location. A considerable number of tests have been made in the past utilizing a heat-treatable type of steel with initial high hardness; and after the particular test has been made, measuring the drop in hardness as an indication of the temperature distribution.

By combining studies of temperature distribution and the aforementioned analysis of appearance and physical data, it should be possible to determine the necessity for further design and development work.

[2] See SAE Transaction, Vol. 50, June, 1942, pp. 212-220, 252: "Airflow through Intake Valves," by G. B. Wood, Jr., D. U. Hunter, E. S. Taylor, and C. F. Taylor.

PAYLOAD versus OPERATING SPEEDS IN AIR TRANSPORT OPERATIONS

IMPORTANCE of operational control of speeds through the use of constant power in air transport flights is emphasized by Mr. Borger in this paper.

by J. G. BORGER
Pan American Airways System

SPEED is the prime commodity of air transportation. Many other advantages are offered by air transportation, but none is so important as speed. Why do we travel by air, and why do we think of shipping cargo by air? If it weren't for speed, how could air cargo hope to compete with surface transport in this country? True, there are some places in the world where the smaller capital investment required for starting an airline gives air transportation another advantage over surface transportation, but by and large the greatest selling point of air cargo is speed.

Let us be aware that while air cargo holds a great speed advantage over surface today, the railroads and trucking companies are doing everything possible to diminish the difference. When considered on the basis of total time from shipper to receiver, which is, in the final analysis, the true measure of speed, surface transport is already competitive, especially for short ranges.

Therefore, all possible should be done to retain the speed advantages of air cargo, so that the selling advantages can be maintained and improved.

There are many ways in which the operating speeds of air transportation can be increased while each airplane is in the design stage. Speed should be obtained through aerodynamic cleanliness, not by the expenditure of brute power. But this subject has been ably discussed in previous papers, and this paper is to discuss the effect of some design factors on the operating speed of aircraft, and the effect of operational factors which set up and control the speeds at which an airplane should be flown once it has been delivered for service on a given route. This does not mean that the importance of designing more speed into an airplane should be underestimated, because it is a tremendously important factor.

THE AUTHOR: J. G. BORGER (J '37), who has been with Pan American Airways since 1935, a year after graduating from M.I.T. with a B.S. degree in aeronautical engineering, assumed his present position of staff engineer to the vice-president and chief engineer's office in February, 1943. His first job at Pan American was assistant airport engineer during the construction of the bases at Midway and Wake Islands, and as an apprentice and junior engineer in the maintenance department of the Pacific Division. In 1937 he was transferred to the Atlantic Division and several years later was named assistant to the division engineer. He later was assistant resident engineer at the Lockheed factory in conjunction with development of the Lockheed Constellation.

[This paper was presented at the SAE Air Cargo Meeting, Chicago, Nov. 8, 1943.]

The payload that any airplane carries is determined by the equation:

Payload = Gross Weight — (Weight Empty + Equipment + Crew + Gas + Oil)

Usually, weight empty, equipment, and crew weights remain relatively constant for a given operation, and since the oil load is usually fixed by a minimum, or depends on the fuel load when requirements are greater than minimum, the only variable is fuel required.

Fuel load is composed of:

1. Climb Allowance.
2. Cruising Consumption.
3. Reserve.

Climb fuel usually remains a direct function of altitude and does not vary from the optimum previously selected as a result of flight test. Reserve fuel is usually based upon some type of cruising fuel consumption. Cruising fuel consumption is, therefore, the prime variable determining the total fuel consumption, and thereby the payload, for each flight.

As we all know, range and payload are intimately tied together, and one should not quote payload without quoting the corresponding range. The basic criterion for determining the range or payload of an airplane is the ground miles covered per pound of fuel consumed. It is customary to plot miles per pound against calibrated air speed, as shown in Fig. 1.

Fig. 1 shows the miles per pound for a typical airplane, essentially similar to the Douglas DC-3, plotted for still air condition at 10,000 ft density altitude. It is very easy to determine the calibrated air speed for maximum miles per pound for the gross weights shown. (Calibrated air speed is the same as correct indicated air speed, that is, the air speed shown on the indicator, corrected or calibrated for instrument error.) For convenience, the speed for best lift-drag ratio is also shown. The reason for the difference between this and the speed for maximum miles per pound is discussed further on in the paper.

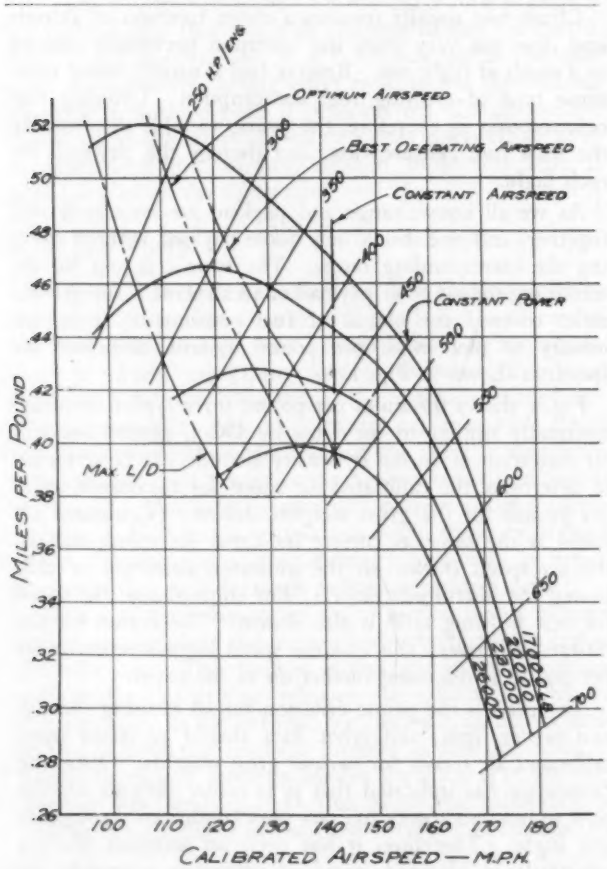
Air speed is the prime criterion for all cruising control, and power, rpm, and other data should be based upon calibrated air speeds for various gross weights. Operating experience has indicated that it is rather difficult for the pilot to hold air speeds to exact peak conditions throughout any flight. Therefore, it has been an accepted practice arbitrarily to draw a curve of optimum operating air speeds against gross weight, through points on the miles-

per-pound curve which are approximately ½ of 1% lower than the peaks, and on the high air speed side of the peaks. There is no justification for flying slower than best-miles-per-pound air speed, except for maximum-endurance type of operations. When using these operating speeds, pilots are instructed that such speeds are desired maxima. The flight tolerance is greater on the low side of the operating speed than on the high side.

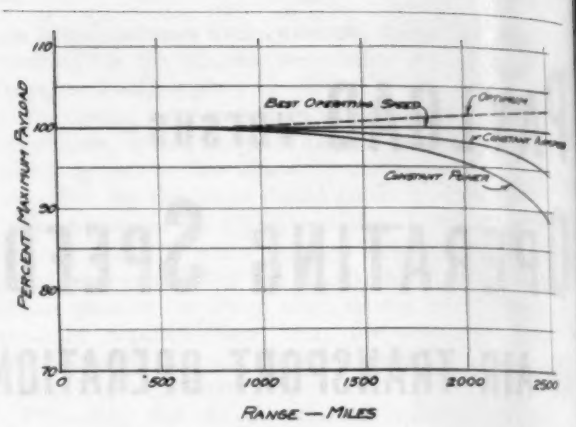
Also shown in Fig. 1 are lines of constant air speed and constant-brake-horsepower operation, all radiating from the same point on the miles-per-pound curve for initial gross weight. It appears that the use of constant-air-speed or constant-power operating techniques would entail a progressively greater loss in average miles per pound and, therefore, payload. However, this effect is relatively small, especially for the shorter ranges.

Using the data from Fig. 1, and using representative airline operating conditions and weights, Fig. 2 has been plotted to show the loss in payload through the use of different operating techniques. The loss in payload due to constant-power operation for flights up to 1250 miles is less than 1%. The advantages of the use of constant power in flight are so great that such a loss in payload would surely be justified on the basis of decreased crew fatigue and especially greater accuracy of execution of cruising control by the flight crew.

The dotted curve in Fig. 2 is considered more or less theoretical, since it is practically impossible to attain this last ½% of miles per pound in flight. Furthermore, the accuracy of flight-test data used in determining the miles-per-pound curves is not sufficient to justify counting upon



■ Fig. 1 – Variation of miles per pound with air speed



■ Fig. 2 – Loss of payload with different operating techniques

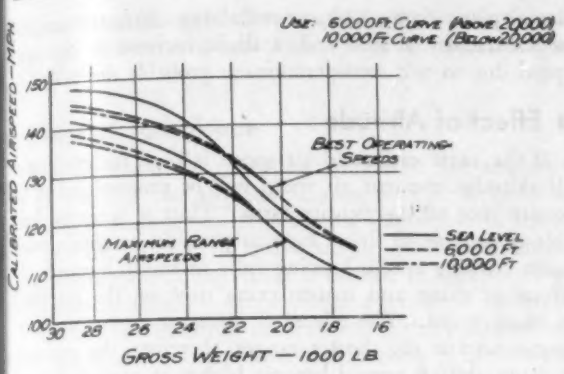
these additional miles per pound. The curve does show the desirability for maintaining very careful control of air speeds, especially as ranges are increased above 1500 miles.

The development of any type of cruising control should be based upon the premise that the end result must be simple. There are so many duties incumbent upon a flight crew that the addition of a very complicated cruising control technique may add to crew fatigue and thus decrease operating safety. Unless the cruising control technique is simple, it will not be properly applied. If cruising control is not properly applied, it is necessary that reserve gasoline be carried to compensate for any lack in accuracy of execution of the cruising control technique. From the airline operating standpoint it is far better to have a cruising control procedure that has an excellent chance of being used with care and accuracy, and with a very small tolerance of error, rather than to have a technique which may theoretically promise greater payloads, but is actually virtually impossible of execution, due to its complication. This does not imply a reflection on flight crews, it simply indicates the level of complexity of airline operation, and the high desirability of reducing such complexities.

I am unable to define specifically where the dividing line lies between the simple and the complicated procedure. It seems to be mostly a matter of judgment based on experience. One thing I do know is that most of us engineers have a tendency to make procedures a little too complicated.

Experience has shown us the desirability of constant-power operation, and other plots similar to Fig. 2 have shown us that constant power can be used on most airplanes at ranges up to at least 1250 miles. As a matter of fact, this limiting range can probably be extended to somewhere in the neighborhood of 1500 miles, by using optimum air-speed cruising control data to determine gas loads, and thereby payloads. In the early portion of flight, the plane is operated at constant power, and if the flight progresses, with little reserve being used, the pilot lets the airplane continue on in constant power and deliberately uses up some of his reserve before landing. Under such favorable conditions, at the 1250-mile point the pilot would have more reserve available than needed for the remaining 250 miles, and there is no reason why he shouldn't use up some of the extra reserve. After all, we are not yet deliberately in the business of carrying gasoline by air.

The advantage of the constant-air-speed operating tech-



■ Fig. 3 – Optimum air speeds

nique is that one air speed can be named for all operating conditions, and it can remain fixed in the pilot's mind and be placarded on the instrument panel. However, since it is necessary to reduce power to maintain the same air speed, the pilot might just as well reduce the power somewhat more to maintain optimum air speed, thus obtaining the greatest measure of improvement for his extra effort.

Fig. 3 shows a means of simplifying the cruising control technique for the varying air-speed type of operation. This method can also be used for constant-air-speed operation. From the miles-per-pound curve similar to Fig. 1, and drawn for several different altitudes, the operating calibrated air speeds are plotted against gross weight. From these plots, a single operating calibrated air-speed curve for all altitudes can be selected, with no appreciable effect upon the average miles per pound or payload. This optimum speed for all altitudes is usually selected to be nearer the optimum for the cruising altitudes at which it is anticipated that most operations will be conducted. Having such an optimum air speed it is usually possible to give the pilot or flight engineer a single plot of calibrated air speed against gross weight which he can use for all altitudes, thus greatly simplifying the flight problems.

In Fig. 3, it was not desirable to draw the resultant operating air-speed curve, since it would detract from the clarity of the other curves. The optimum in this case is considered to be the 6000-ft curve for gross weights above 20,000 lb, and the 10,000-ft curve for weights below 20,000 lb.

We have discussed above the effect of operating procedures on payloads, which indicate the desirability of using constant power for short-range and some medium-range operation, and optimum air speeds for all operations above approximately 1500 miles. As a matter of fact, if one is comparing one airplane with another it may be somewhat better to compare this in terms of hours rather than miles and say that the dividing line is in the region of a 7-hr flight. That is, those less than 7 hr should be at constant power, those greater, at optimum air speed. The use of hours provides a better comparison between different airplanes since rate of change in gross weight is more a function of time than of speed.

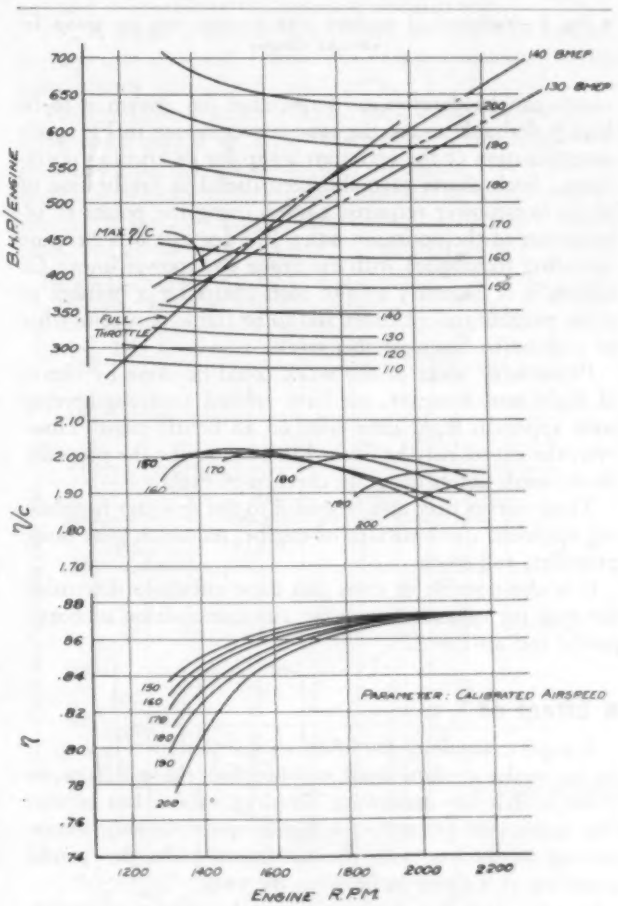
Not only should we analyze for optimum operating air speeds and/or powers, but the airplane and powerplant should be analyzed for optimum powerplant operating conditions. The optimum rpm for every horsepower and air-speed combination should be determined, so that the powerplant always operates at maximum efficiency. It is possible to lose as much payload because of operation at the wrong rpm, as when operating at the wrong air speed.

Efficiency of the engine is inversely proportional to the specific fuel consumption c. Multiplying engine efficiency times propeller efficiency η, we obtain the function η/c to express overall powerplant efficiency. This appears in Breguet's classical formula for range.

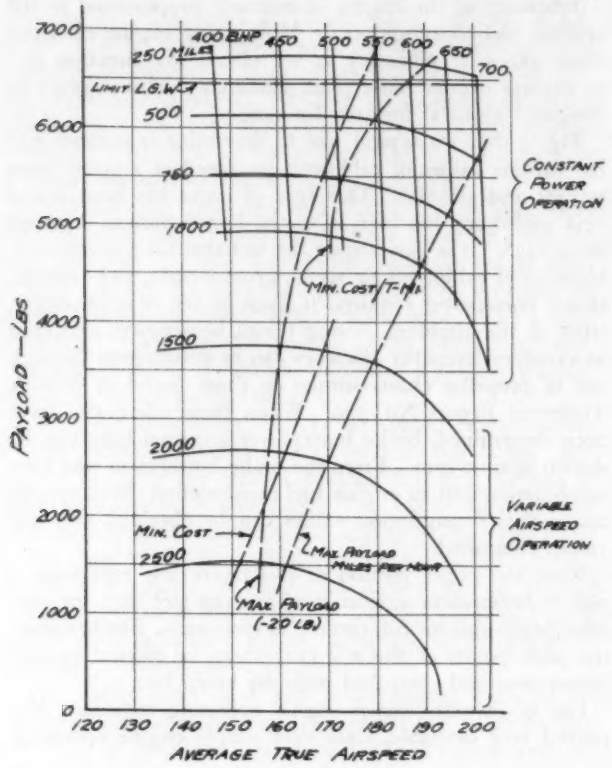
Fig. 4 shows a typical plot to determine maximum η/c for various values of calibrated air speed at a given gross weight and altitude. This type of curve has been found very useful and has been in use by Pan American Airways since 1938. It is based upon the fact that for a given combination of calibrated air speed, gross weight, and altitude, thrust horsepower required is fixed by the drag characteristics of the airplane. Using thrust horsepower and rpm as variables, propeller efficiency can be determined through use of propeller charts similar to those found in NACA Technical Report No. 749. When these efficiencies have been determined, brake horsepower required lines can be drawn against rpm. From the brake horsepower and rpm combination and an engine fuel consumption chart, specification fuel consumption values can be obtained and η/c values computed.

Since the upper portion of this figure also represents a plot of horsepower against rpm, one can plot lines for constant bmep and for full throttle as cross plots. Furthermore, the peak points of the η/c curves can be plotted against horsepower and compared with the bmep lines.

Use of constant bmep engine operating technique has proved very desirable, since very simple engine operating



■ Fig. 4 – Typical powerplant efficiencies – altitude: 10,000 ft; gross weight: 25,000 lb



■ Fig. 5 – Variation of payload with average true air speed for various ranges

charts can be developed. Experience has shown it to be highly desirable to use the type curves shown in Fig. 4 for determination of the optimum bmep for all cruising operations. Such charts have also been useful in preparation of brake horsepower required curves, using the points of intersection of the optimum bmep line (or any other engine operating technique) with the brake horsepower lines. Of course, it is necessary to plot such charts for a number of gross weights and altitudes and to be frank, it is somewhat of a chore to complete this work.

Presumably some of the work could be done by means of flight test, however, we have seldom seen engineering costs approach flight time costs on an hourly basis. However, the curves may be limited in accuracy by the propeller charts used, unless these are checked in flight.

These curves may also be useful to the designer in selecting optimum combinations of engine, reduction gear ratio, propeller, and so on.

It is also possible to cross plot these curves to determine the rpm for maximum η/c for any combination of horsepower and air speed.

■ Effect of η/c

A good example of the effect of η/c is shown in Fig. 1. As far as the airplane itself is concerned, the optimum air speed is that for maximum lift/drag ratio. But in this case, maximum η/c occurs at higher speed, causing a compromise as shown, with the maximum miles per pound occurring at a speed in between the two.

It is desirable that the propeller and engine combination be selected with care, so that there may be little or no loss in payload due to η/c considerations. While in most cases the airplane flying and controllability characteristics will be satisfactory at best l/d, a slight increase in operating speed due to η/c considerations is probably desirable.

■ Effect of Altitude

If the same calibrated air speed is used for cruising at all altitudes, the true air speed will be proportional to the square root of the density ratio. Thus it becomes desirable to be able to fly as high as possible, to obtain maximum cruising speed, keeping fully in mind, however, the effects of climb and maneuvering time on the net block to block speed. The effects of these, of course, are more pronounced at the shorter ranges, therefore, the optimum cruising altitude would become higher as operating range is increased.

If η/c remains constant, payload will remain constant for all altitudes. However, η/c usually decreases with altitude, because of greater power required, and therefore payload is usually greatest at low altitudes. The variation of η/c with altitude has a strong effect on the variation of payload with altitude.

■ Effect of Climb

Climb detracts from the average air speed during flight and is usually conducted at higher power than cruising. Hence, the total gas consumption is increased. This is not fully compensated for by descent, and so the net effect of climb and descent is to decrease the optimum operating air speed for all flights.

■ Effect of Wind

The optimum air speed when flying against headwind is greater than that for still air, while that for tailwind conditions is slower than that for still air. If operations are to be conducted on a route on which a large percentage of flights are against prevailing headwinds, it is desirable that the optimum operating speed be increased, or at least determined on the basis of miles-per-pound curves drawn for the headwind condition. On the other hand, it is not customary to decrease operating air speeds for tailwind conditions, since the increase in miles per pound, due to the tailwind, is so great, that an additional increase due to change in operating speed becomes negligible. When setting up operating techniques it is almost always desirable to allow for some headwind to be encountered under average conditions, and to adjust the air speeds on the basis of this assumed average headwind. On almost any route, the average wind component at a given altitude, is a headwind.

■ Effect of Reserve

On many short flights, the alternate may be as far away from the destination as the destination is from point of departure. In such cases, the amount of reserve carried for flight from destination to alternate airport, plus the allowance for instrument letdown, becomes considerably greater than the amount of fuel required for routine flight between airports. Since reserve fuel is usually based upon a single optimum operating condition, the amount of reserve carried should remain constant for any operating air speed used in cruising flight between airports. Therefore, since the amount of fuel varying with operating air speed is relatively small in proportion to the total fuel carried, its effect upon the optimum speed becomes less, and the optimum operating speed tends to become greater

under such conditions than it would be under conditions where the reserve is a small percentage of the total gas carried.

Effect of Turbulence

Often the optimum operating air speeds for airplanes are so low that the airplane is very difficult to control, especially in turbulence. It is almost impossible to maintain constant altitude in turbulent air conditions, unless the minimum operating speed is limited to that speed at which there is sufficient excess power available to maintain a minimum rate of climb of 200 fpm, if the indicated air speed is reduced to best rate-of-climb speed. This margin enables the pilot to maintain sufficient speed to enable him to control the airplane and maintain a more accurate cruising control during turbulent air conditions. The effect of mushing around through the air is extremely severe upon total gas consumed.

In some cases a 200-fpm rate of climb requirement has been prohibitive. This has been especially true with the older airplanes with lower wing loadings.

Effect of Landing Gross Weight

Most airplanes are operated at provisional gross weights higher than those allowed for landings. In other words, the airplane must consume sufficient gas in flight to reduce the gross weight to the maximum allowed for landing. On shorter flights, it may not be possible to consume the full difference in weight between provisional and landing gross weights. Under such conditions, flying faster (within limits) would not decrease the payload, and the optimum speed is increased. The optimum speed is then usually determined by minimum operating cost considerations.

Effect of Ground and Maneuvering Time

Both of these decrease overall operating air speeds. Since their effects are smaller on the lower air speeds, the general tendency would be to decrease optima.

In air transportation it is generally desirable to obtain the greatest total payload per year. This not only means flying at the highest possible speeds, but also obtaining greatest utilization. Much has been said about the necessity of keeping ground time to a minimum, and too much attention cannot be paid to this subject.

Effect of Other Variables

There are a number of other variables which affect operating speeds. One example is a limitation on operating rpm, due to propeller or engine vibration characteristics in a certain rpm range, airplane or engine mount vibration characteristics, minimum propeller governing rpm, minimum rpm at which the electric generators will supply power, and so forth. In such cases it is usually not desirable to let the speed decrease further, but rather to maintain constant air speed and reduce power by reducing the bmep, and maintaining constant rpm.

Fig. 5 shows a plot of payload against average true air speed for various ranges. Operations on flights up to and including 1000 miles are assumed at constant power. Flights 1500 miles and above are assumed at varying air speeds, which are directly proportional to the optimum operating air speeds. In each case, reserve has been assumed to allow for flight of 200 miles beyond destination to an alternate airport, plus one hour after arrival for instrument approach. These curves have been developed from the miles-per-pound curves of Fig. 1, and using representative airline operating weight data. However, no attempt has been made to maintain absolute accuracy in the quantitative values, but merely to show trends and shapes of the curves. Note that changes in the weight empty of the airplane would have the simple effect of moving the curves upward or downward.

Fig. 5 clearly shows that the average speed for maximum payload decreases as range is increased. It also shows that it is necessary to maintain a more accurate and careful cruising control at the longer ranges than in the short-range operations. Of course, both of these effects have been well-known for a long while.

Incidentally, the optimum air-speed curve shown here is actually drawn for a payload of 20 lb less than maximum. All curves are so flat at the top that it is very difficult to pick out the maximum, and it is felt that any airline would willingly sacrifice 20 lb payload for the increase in speed obtained. Also, please note that this airplane is not designed as a long-range airplane, and therefore, some of the long-range features and curves show up to a slight disadvantage.

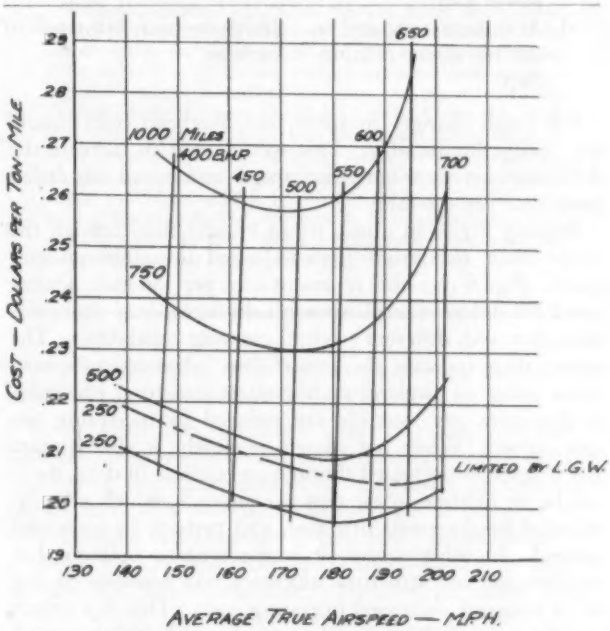
The constant-power lines are also cross plotted on this curve, so that it is possible to determine power to be used for various ranges.

The dotted line at 6450 lb shows the payload as limited by landing gross weight. In this case, speed for maximum payload is limited by maximum cruising power.

Fig. 6 shows a similar plot of cost per ton-mile against average true air speed, for the short-range operations, using the payloads shown in Fig. 5.

To determine operating costs, the average airline operating costs quoted by E. P. Warner[1] were used, and adjusted slightly, for assumed cargo operation. These adjustments were quite arbitrary and consisted of elimination of passenger service charges, and reduction of some traffic, sales, and advertising charges. No attempt should be made

[1] See *Journal of the Royal Aeronautical Society*, Vol. 47, July, 1943, pp. 185-257: "Post-War Transport Aircraft," by Edward P. Warner.



■ Fig. 6 – Variation of cost per ton-mile with average air speed

to interpret any of the curves shown herein in terms of absolute values. It is desired only to show what happens when certain reasonable assumptions are made, for it is the writer's firm belief that only a few generalizations can be made from curves of this type, and that it is absolutely necessary to analyze each given type of airplane over the specific route on which it is to be operated, and under the specific operating conditions that are anticipated, to obtain specific recommendations. The results should be carefully evaluated and simplified as much as possible so that the cruising control technique will be simple and concise to ensure excellent execution by flight personnel.

It is important to include all costs when making an analysis of this kind. Direct flying costs alone are not sufficient, for overhead costs also have their effect upon operating air speeds.

Costs may be broken down into the following types (when making similar analyses for the effect of operating speed):

Costs	Effect of an Increase in Cost on Speed for Minimum Cost
1. Proportional to hours of operation (depreciation, crew, and equipment insurance, some overhead, plane maintenance – for this type of analysis).	Increase.
2. Proportional to miles operated (some overhead, passenger service, cargo insurance, crew expense).	Theoretically no effect, but the greater the proportion of these costs to total, the less the effect of other types.
3. Dependent upon other factors:	
a. Crew: hours and speed.	Increase, but at diminishing rate.
b. Gas and Oil: amount used.	Hold even.
c. Engine maintenance: power used.	Decrease.
d. Administrative, and so on: no direct relationship.	Indeterminate, but tends to increase.

For small changes in speed, the overhead costs should not change appreciably. This would tend to increase the difference between optimum speed and speed for minimum cost per ton-mile.

Bearing Fig. 5 in mind, it can be said that through the range from maximum payload speed to minimum cost speed. Fig. 6 can also represent cost per ton-mile against speed for different airplanes with the same drag characteristics, but with different payload carrying capabilities. The curves then indicate the tremendous advantage obtained when using an airplane with greatest structural efficiency, in that costs per ton-mile are reduced by increasing the tons carried. While not shown as clearly, it also appears that if speed is decreased through an increase in drag, there will be an increase in the cost per ton-mile, which must be balanced by decreased first cost, and perhaps by increased payload. In other words, it is the writer's opinion that aerodynamic and structural efficiency will continue to pay off in terms of decreased operating costs. This is a rough conclusion from these studies, and should be substantiated by further studies, especially by design engineers, who are more qualified to judge the effects of incorporating such refinements upon manufacturing costs.

It is noteworthy that the air speeds for minimum cost per ton-mile are somewhat higher than those shown for maximum payload. This is shown in Fig. 5. The air speeds for minimum cost per ton-mile for the long-range operations are also shown on Fig. 5, but no plot of these curves of cost per ton-mile against air speed is included because the absolute values are too far out of range, due to the fact that the airplane is just not designed for this type of operation. It is obvious that it was incorrect to assume that a sacrifice of 20 lb in payload would be justified for the 2500-mile flight. In this particular case, such a decrease in payload would also be accompanied by an increase in cost per ton-mile.

Another criterion of cruising operation has been suggested, primarily for use in conjunction with present military operations. This is the maximum-payload-pound-miles-per-hour technique suggested by Dixon Speas of American Airlines.[2] While this technique has certain advantages for use in wartime operations, as Mr. Speas points out, and as shown herein, the speeds used are higher than those for minimum cost per ton-mile. Furthermore, any such analysis should take full account of the fact that ground time will not change directly with air time, the very considerable increase in engine maintenance cost due to higher power operation required, plus the decrease in engine reliability also due to higher power operation. It is very difficult to evaluate specifically the effect of per cent power used, but we know from experience that engines normally operated at low power are both more reliable and less costly to maintain, than those operated at higher powers. This might be overcome by the use of engines designed for operation at such powers, but the use of such engines would mean a weight sacrifice that probably would not be warranted.

The maximum-payload-pound-miles-per-hour technique does have greater application with large fleets of airplanes, which can operate on frequent and flexible schedules, and which are not limited by excessive servicing time at stops. It applies more for short-range than long-range operation.

To sum up, speed has an advantage in increasing the total payload carried per year; not only through its direct effect, but indirectly because a high-speed airplane can complete more schedules per year, and is less sensitive to variations in wind and weather conditions. Another factor that applies particularly to operations over unimproved airways, is the fact that the faster the airplane, the more miles covered between dawn and dusk. This will have a distinct advantage in many places outside Europe and the United States where night lighting of airways is not available.

In making similar analyses, it is again emphasized that it is very essential that each operating condition be analyzed independently. The effects of range, reserve, average headwind, operating techniques, schedules, and schedule completion are so varied that it is virtually impossible to make specific recommendations regarding operating speeds.

I wish to acknowledge the aid of L. H. Allen, Jr. and R. W. Blake of the vice-president and chief engineer's staff of Pan American Airways, in assisting me with the preparation of curves and calculation of data.

[2] See *Aeronautical Engineering Review*, Vol. 2, July, 1943, pp. 23-31, 101: "Engineering of Power Control and Altitude Selection in Airline Operation," by R. Dixon Speas.

Factors Affecting the ANTIKNOCK PERFORMANCE of POST-WAR FUELS

by T. H. RISK
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IT is possible even now to make a reasonable prediction of the antiknock value of post-war fuels in terms of laboratory octane number, the conventional yardstick for measuring this characteristic. However, owing to tremendous advances in refining technique during the war and to other important factors, laboratory-determined octane numbers alone may result in inaccurate conclusions as to the "true" antiknock quality of these post-war fuels when used in a particular vehicle. For example, it is quite within the bounds of possibility that post-war fuels made by the new processes and rating 83-85 octane number by the ASTM laboratory method might give better antiknock performance on the road than pre-war aviation fuel of 87 octane number.

It is the purpose of this paper to show that the question of antiknock value of post-war gasolines is not as clear cut as mere octane number would indicate and to point out some of the factors involved.

In the main, there are four subjects to consider:

1. How new wartime refinery equipment, now used for making aviation fuel, will be used to make motor gasoline.
2. The possible variations in chemical composition of gasolines and the effect of such variations on the road antiknock behavior of the gasolines.
3. The effect of possible variations in engine and vehicle design on the antiknock ratings of the gasolines.
4. The probable laboratory octane values of post-war gasolines to be used as the "convenient" means of expressing fuel antiknock quality for refinery control.

Obviously, a complete discussion of these factors leading to a single conclusion is impossible in the light of present knowledge and well beyond the scope of this discussion. Rather, it is our purpose to show, by means of a limited number of examples, why it is necessary to consider the relationship of these factors when considering the antiknock quality of post-war fuels.

THERE are four subjects to consider in determining the question of antiknock value of post-war gasolines:

1. How new wartime refinery equipment, now used for making aviation fuel, will be used to make motor gasoline.

2. The possible variations in chemical composition of gasolines, and the effect of such variations on the road antiknock behavior of the gasolines.

3. The effect of possible variations in engine and vehicle design on the antiknock ratings of the gasolines.

4. The probable laboratory octane values of post-war gasolines to be used as the "convenient" means of expressing fuel antiknock quality for refinery control.

■ ■ ■

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■ New Refinery Equipment

How will the new wartime refinery equipment be used after the war? Most of this new equipment was installed for the purpose of boosting the production of 100-octane aviation gasoline. By the time the war is over, capacity to produce aviation fuel will greatly exceed the forecasted peacetime demand. What will happen to excess production, or, put another way, what will happen to the facilities not needed to produce 100-octane gasolines?[1]

Conceivably the petroleum industry could continue to produce 100-octane gasoline up to the full limit of capacity and dump the excess into motor gasolines. To determine the effect of this procedure, an estimate of post-war aviation gasoline consumption is required. Estimates of this demand vary considerably from about 50,000 to 150,000 bbl per day. Due to wartime restrictions, the excess cannot

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[1] See *Oil and Gas Journal*, Vol. 42, Dec. 9 1943, pp. 64, 67: "Today's Cracking Equipment in the Post-War Era," by C. R. Wagner.

be stated in actual figures, but it has been estimated that, if the excess post-war aviation gasoline were used in this manner, the overall effect would be a 2-3 octane-number[2] increase in motor gasoline octane number. This relatively slight increase in octane number would be expensive, since aviation gasoline costs the refiner considerably more to make than motor gasoline.

It is, therefore, much more likely that the new refinery equipment not needed to make aviation gasoline after the war will be used more directly to produce gasoline suitable for automobiles, trucks, and buses. The exact manner of utilization will vary within the industry and will be based on such factors as:

1. Relative demand for various grades of fuel, that is, aviation, premium, housebrand, and third grade.

2. General refinery economics, that is, equipment available, crude supplies, and so on.

3. Post-war car requirements and competitive demands.

A survey of refinery equipment available shows that the American refiner now has at his disposal almost twice as many practical methods or processes for making gasoline blending stocks as he had before the war. It is unlikely, however, that expensive components, such as alkylate and hydrocodimer, will be used in the manufacture of motor gasoline after the war. In Tables 1 and 2 an attempt has been made to forecast the blending stocks that may be used in appreciable quantities, together with their antiknock values.

Table 1 – Motor Fuel Components

Pre-War	Average Octane Number	
	ASTM Motor	CFR Research
Straightrun Gasoline	55	55
Natural Gasoline	65	65
Thermally Cracked and Reformed Gasoline	70	80
Polymer Gasoline	80	92–95
Ethyl Fluid		
Additional Post-War Components		
Catalytically Cracked Gasoline	78–80	90–92
High Aromatic (Hydroformed	80	90–92
High Aromatic (Cycloversion	80	92–94
Superfractionated Straightrun Gasoline	70–75	70–75
Isomerized Straightrun Gasoline	70–75	70–75

Table 2 – Aviation Fuel Components

Pre-War	Average Octane Number	
	ASTM Motor	CFR Research
Straightrun and Natural Gasoline	70–75	72–77
Alkylate and Hydrocodimer	90	90–92
Isopentane and/or Neohexane	95	95
Ethyl Fluid		
Additional Post-War Components		
Catalytically Cracked – Second and Third Pass	80–82	88–92
Superfractionated Straightrun Gasoline	70–75	70–75
Isomerized Straightrun Gasoline	70–75	70–75
Cumene	Very high blending value	

It is clear from the foregoing that the new blending stocks will increase the antiknock value of the finished gasolines, but it should be emphasized again that laboratory octane number does not tell the full story. New characteristics derived from improved refining processes and the blending possibilities of these stocks will not be too well indicated by laboratory octane number alone. The very fact that there is more than one test method is evidence that the "end use" of a gasoline has a considerable effect on its antiknock quality. Actually, a gasoline can have – and usually does have – several different laboratory numbers. Selection of the most representative will depend on the intended use of the fuel – not merely on some inherent quality of the fuel.

There are three general methods of rating fuels in CFR laboratory knock test engines, and to avoid misunderstanding they are reviewed briefly in Table 3.

Table 3 – Laboratory Octane Methods[3]

Test Method	ASTM Motor	CFR Research	I-C Aviation
Operating Conditions			
Speed, rpm	900	600	1200
Coolant Temperature, F	212	212	374
Mixture Temperature, F	300	125	220
Spark Advance, BTC	Variable with compression ratio	13	38

All of these methods require the use of the CFR single-cylinder test engine but, as indicated, employ different operating conditions. In all cases, the test fuel is compared, in knock intensity, to blends of reference fuels of known octane number. The matching blend determines the octane number of the test fuel.

When discussing "octane numbers" without other qualification, the ASTM Motor method is generally implied.

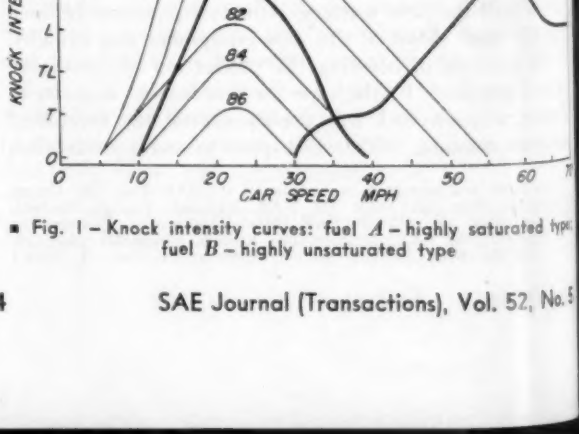
■ Blending Refinery Stocks

The statement that laboratory ASTM octane number does not necessarily give a true picture of the road antiknock behavior of all gasolines will come as no revelation to technical men in the oil and automotive industries. At the time it was adopted by the Cooperative Fuel Research Committee in 1932, the ASTM laboratory method was designed to rate fuels in terms of octane numbers which would be comparable to the average road octane number of these fuels when tested by the CFR Uniontown road-test procedure.[4] Even at that time the ASTM laboratory octane number did not in all cases give correlation with road results. Since little gasoline was then marketed that did not give reasonable correlation, the method was considered satisfactory for all practical purposes.

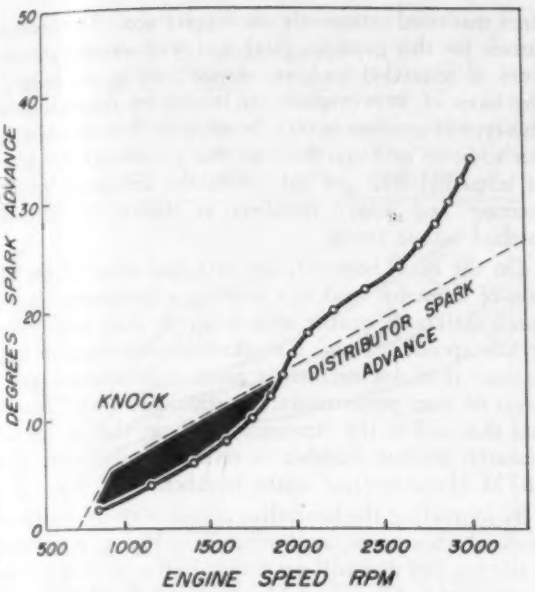
[2] See "An Informal Long-Range Forecast of Motor Gasoline Developments," by D. P. Barnard and R. F. Marschner. Presented before the SAE War Engineering – Annual Meeting, Detroit, Mich., Jan. 10, 1944.

[3] See ASTM Designation D614-43T, "Tentative Method of Test for Knock Characteristics of Aviation Fuels;" ASTM Designation D357-43T, "Tentative Method of Test for Knock Characteristics of Motor Fuels;" and SAE Transactions, Vol. 44, June, 1939, pp. 277-280: "CFR Research Method of Tests for Knock Characteristics of Motor Fuels," developed by the Cooperative Fuel Research Committee.

[4] See SAE Transactions, Vol. 28, March, 1933, pp. 105-120: "Antiknock Research Coordinates Laboratory and Road Tests," by C. B. Veal, H. W. Best, J. M. Campbell, and W. M. Holaday.



■ Fig. 1 – Knock intensity curves: fuel *A* – highly saturated type; fuel *B* – highly unsaturated type



■ Fig. 2 – Borderline knock curve – method used to obtain curve for any fuel

Largely because of improved refinery methods, more and more gasolines have been produced which show better antiknock characteristics in cars on the road than is indicated by laboratory ASTM octane values. Because of this, much research work has been directed toward the development of improved test methods.

When using the CFR Uniontown road-test method, difference in fuels could be shown as indicated in Fig. 1. Fuel *A*, the saturated-type gasoline,[5] knocks at low speeds, while fuel *B*, the unsaturated-type gasoline, knocks at high speeds. This method, however, rates fuels on the basis of knock intensity regardless of the speed range wherein the knock occurs. Ratings are assigned to the test fuels by comparing their knock intensity to that of blends of secondary reference fuels. As shown in Fig. 1, both test fuels *A* and *B* have maximum knock intensities equal to that of an 80-octane reference fuel blend and, therefore, both are given a road rating of 80 octane in spite of the obvious difference in knocking tendency with respect to speed.

In view of these shortcomings and since no idea of fuel antiknock behavior is indicated outside of the actual knocking range, it was obvious that a road-test method giving more information was required.

The borderline-knock-test method, adopted by the CFR as a standard procedure in 1940, gives more information of the type sought by automotive engineers and petroleum technologists. With this method, the road antiknock performance of a gasoline is indicated by means of a curve instead of a number. A borderline curve shows, graphically, a fuel's ability to tolerate spark advance throughout the engine speed range. As its name implies, it represents the dividing line between nonknocking and knocking operation.

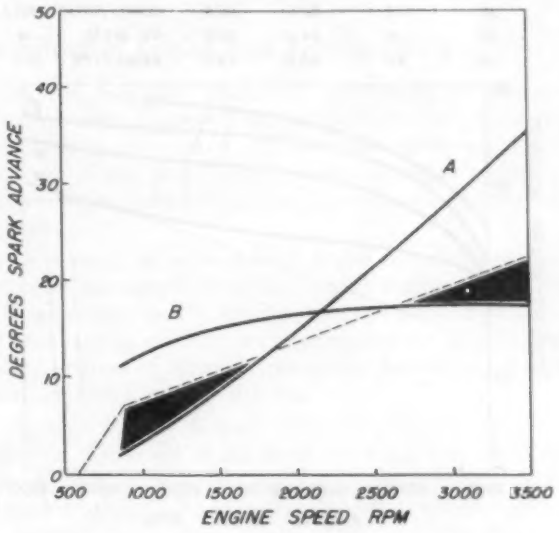
For testing by this method, the test car is equipped with a fixed or nonautomatic distributor that can be varied manually by the driver, a suitable spark-advance indicator, and an engine tachometer. The procedure for determining the borderline curve for a fuel is as follows: The spark is set manually to a relatively low advance, usually 2 to 6 deg BTDC. The car is slowed down to as low a speed as possible in high gear, generally about 6 to 8 mph and then accelerated with wide-open throttle. The speed at which knock-die-out occurs is noted. This gives one point on the border line curve – representing the two dimensions, spark advance and knock-die-out speed. The spark is then advanced 2 deg and the entire procedure repeated to give the second point on the curve. This process is continued until the complete curve has been established for the entire spark-advance and engine speed range under consideration.

Borderline test data usually are plotted as shown in Fig. 2. The light dotted line represents the distributor-spark-advance curve for the particular engine in which these tests were conducted. For a fuel not to knock in this car, it must be able to tolerate at least as much spark advance at all speeds as that produced by the distributor-advance mechanism. Wherever the solid line representing the borderline curve for the test fuel falls below the distributor-spark-advance curve, the engine is considered to be knocking. The borderline curve, for this particular fuel, indicates that it will knock at relatively low speeds in the range from 700 to 1800 rpm.

In this example, the actual knock-die-out points obtained on the road are indicated by the heavy dots to illustrate the number of points obtained in the development of a curve of this type. The individual points are not shown on the borderline curves given hereafter, but they all were obtained and recorded in a similar manner.

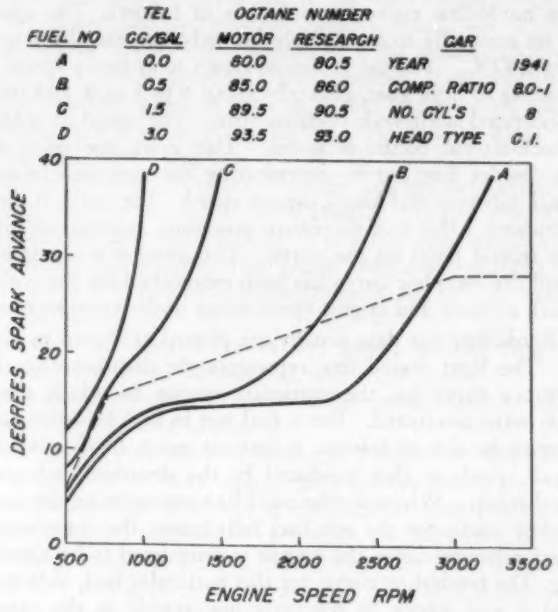
Using the borderline method, as shown in Fig. 3, we get a much better idea of the difference in road antiknock behavior of the two fuels *A* and *B* previously rated by the CFR Uniontown method. The *A*, or saturated type, knocks severely in the low-speed range, from 800 to 1800 rpm, but is much better than the engine requires at high speeds. On the other hand, the *B*, or unsaturated type, has excellent low-speed antiknock characteristics, but knocks at high speeds, above 2800 rpm.

Obviously, a better fuel than either of the above would be one with a borderline curve which was parallel and



■ Fig. 3 – Borderline knock curves: fuel *A* – saturated type of gasoline; fuel *B* – unsaturated type of gasoline

[5] See "Gasoline – Past, Present, and Future," by Graham Edgar. Presented before the Annual Meeting of the American Society for Testing Materials, Edgar Marburg Lecture, 1942.



■ Fig. 4 – Borderline knock curves: road lead susceptibility – saturated type of gasoline

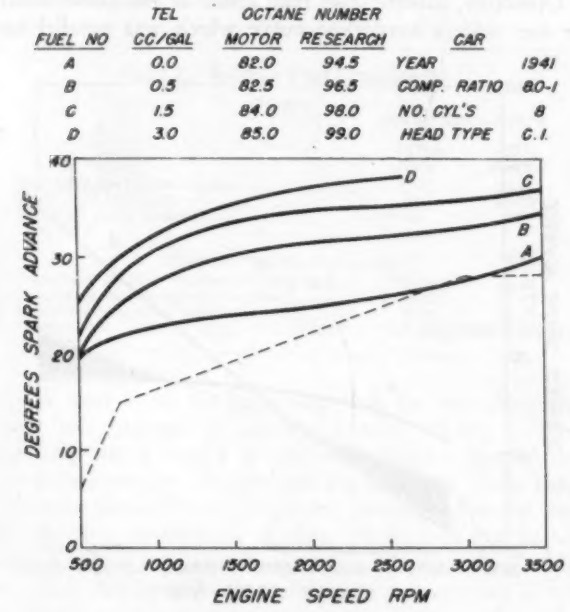
slightly above the distributor-spark-advance curve of the car throughout the whole speed range, thus ensuring knock-free operation. This is by no means a hopeless idea. By blending various refinery stocks, the shape of the borderline curve can be materially changed to obtain a better "fit" between engine and fuel. We submit, however, that conventional laboratory octane number alone would give little or no clue to the fuel characteristics needed to accomplish this end.

In some instances conventional laboratory octane values are actually misleading. Take the case of a saturated-type fuel which is similar chemically to the straightrun gasolines marketed extensively some years ago. The borderline curves for this gasoline, clear and with various concentrations of tetraethyl lead, are shown in Fig. 4. Judged on the basis of improvement in laboratory octane numbers, this type of gasoline is very "susceptible"[6] to tetraethyl lead. An addition of Ethyl fluid to give a concentration of 3 cc of tetraethyl lead per gal causes the antiknock rating to increase 13.5 octane numbers as shown by the Motor method octane rating.

On the road, however, the principal effect of the addition of tetraethyl lead is a marked improvement in high-speed antiknock quality with relatively little improvement at low speeds. Thus, a marked improvement in octane number does not necessarily mean a comparable gain in terms of road performance. It should be noted, however, that this fuel is the "insensitive"[7] type, that is, the CFR Research method number is essentially the same as the ASTM Motor method octane number.

By comparing the borderline curves with the distributor-spark-advance curve, as illustrated in Fig. 4, it is possible to select a fuel that will not knock in this particular engine at any speed. Obviously, blends *A* and *B* fall short of the ideal by a wide margin. Blend *C* is almost good enough but not quite. Thus, by process of elimination, we find that an entirely satisfactory fuel would contain about 1.6 cc of tetraethyl lead and that the octane number by both the ASTM and Research methods would be about 90.

The lack of correlation between conventional laboratory octane number and road antiknock quality is equally apparent in a fuel of quite a different type. The example is a highly unsaturated stock, the borderline curves for which, clear and with various concentrations of tetraethyl lead, are shown in Fig. 5. Rated in terms of laboratory octane number this gasoline has a low tetraethyl lead susceptibility. The addition of 3 cc of tetraethyl lead raises the laboratory antiknock value by only three octane numbers, as compared with 13.5 for the saturated type of fuel. Yet, with respect to road performance, the fuel is more suitable – in this particular engine – than the saturated type, although there is a tendency to knock in the high-speed range. The addition of tetraethyl lead improves this fuel most at high speeds where the unleaded stock is inherently weak. Here again, the "road effect" is not properly indicated by laboratory octane number. For this engine a satisfactory fuel of this unsaturated type would contain only a small amount of tetraethyl lead, approximately 0.2 cc, and would have an ASTM Motor method rating of 82.0 to 82.5 octane number. Being a sensitive type of fuel, the Research method rating would be considerably higher – about 95 octane number.



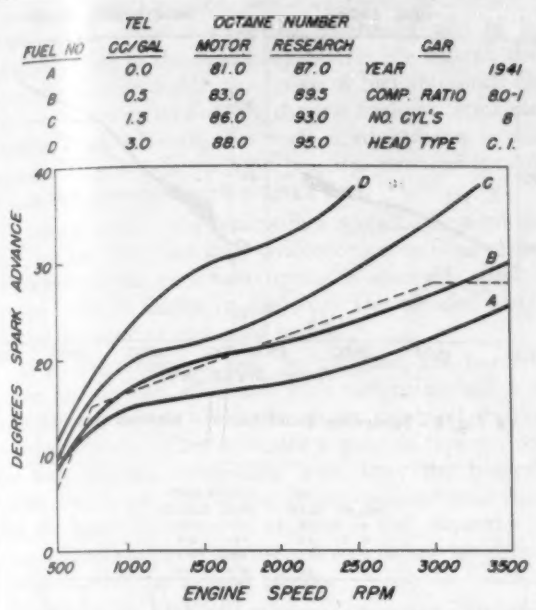
■ Fig. 5 – Borderline knock curves: road lead susceptibility – unsaturated type of gasoline

It is obvious from the foregoing that a blend of saturated and unsaturated types may result in a fuel of better overall characteristics. Borderline curves for such a fuel, a 50-50 blend, clear and with various concentrations of tetraethyl lead, are shown in Fig. 6. We now have three variables – two blending stocks and Ethyl fluid.

[6] See *Industrial and Engineering Chemistry* (Industrial Edition), Vol. 31, July, 1939, pp. 862-865: "Ethyl Fluid Blending Chart for Motor Method Octane Numbers," by L. E. Hebl, T. B. Rendel, and F. L. Garton.

[7] The definition of Motor gasoline sensitivity as used generally by both the petroleum and automotive industries applies only to laboratory values and is: CFR Research octane number minus ASTM Motor Method octane, expressed in octane numbers. Thus, a gasoline having a CFR Research rating of 78 octane number and an ASTM Motor Method octane of 72 would have an octane-number sensitivity of 6.

A "highly sensitive" gasoline is generally considered to be one having a sensitivity of at least 8 octane numbers. An "insensitive" gasoline is one having a sensitivity of less than 3 octane numbers.

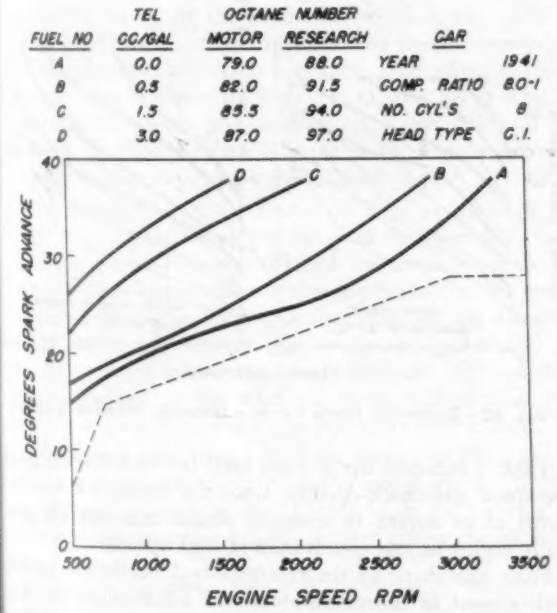


■ Fig. 6 – Borderline knock curves: road lead susceptibility – blended gasoline: 50% saturated, 50% unsaturated

The tetraethyl lead susceptibility of this combination of gasolines by laboratory knock methods is about the average of the saturated and unsaturated stocks – 7.0 octane numbers with 3 cc of tetraethyl lead. The road response is more "even" than for either stock alone, although here again there is a definite tendency, especially with high tetraethyl lead concentrations, to effect more improvement at high speeds than at low speeds. In this case, a gasoline slightly better than blend *B* is required – probably 85 Motor method and 90 Research, with a lead content of about 0.7 cc.

This is an example of what can be achieved by blending two simple combinations of base stocks. With the new refinery processes now available, innumerable combinations are possible. To give some idea of the enormous possibilities due to blending alone let us examine a special blend, clear and with several concentrations of tetraethyl lead. Borderline curves are shown in Fig. 7. This gasoline is the result of selective blending,[8] that is, the control of hydrocarbon type with respect to distillation range. In this case when tetraethyl lead is added, we obtain an almost uniform increase in both low- and high-speed antiknock quality. A gasoline of this type having an ASTM octane number of 79.0 and a Research octane number of 88.0 will be satisfactory for this car.

Table 4 gives a summary of the antiknock ratings in terms of laboratory octane number of the four types of gasolines examined thus far.



■ Fig. 7 – Borderline knock curves: road lead susceptibility – selectively blended gasoline

Table 4 – Summary of Antiknock Ratings

Type of Gasoline	Octane Number: ASTM Motor	Octane Number: CFR Research
A – Saturated	90.0	90.0
B – Unsaturated	82.5	95.0
50-50 Blend	85.0	90.0
Special Blend	79.0	88.0

By blending at the refinery, it is possible to produce fuels varying by as much as 11 ASTM Motor method octane numbers which, under the conditions illustrated, will have essentially comparable road antiknock performance. While these examples may represent extreme conditions, it is reasonable to expect that future gasolines will be more like the special blend, that is, "tailored" to do a particular job. And like the special blend, when in actual service on the road, these fuels of the future will probably rate considerably better than laboratory ratings would indicate.

The sizable variations between actual road antiknock performance and laboratory octane numbers, as shown above, are probably due to the combined effect of physical and chemical factors not usually considered when determining the rating of a gasoline by the conventional laboratory procedure.

■ Effect of Engine and Vehicle Design

In the preceding discussion, differences in antiknock quality were illustrated by the performance of different gasolines in one type of engine. Now, however, it is our purpose at this point to demonstrate the effect of different engine types upon the antiknock performance of the same gasoline.

In general, the valve-in-head engine is inclined to knock more at low speeds than the L-head engine, whereas the L-head engine usually knocks more severely at high speeds. These varying characteristics emphasize the importance of using engines of various types when determining the general antiknock quality of a fuel.

In tests with an L-head and a valve-in-head engine, an attempt was made at the outset to use a fuel that would satisfy both engine types with a minimum of excess antiknock quality. This was done by using a special gasoline

[8] See Proceedings of Annual Meeting of API, Vol. 22 [III], 1941, pp. 70-89: "Improved Motor Fuels through Selective Blending," by C. R. Wagner, W. B. Ross, L. M. Henderson, and T. H. Risk.

blend. Nevertheless, as the borderline curves in Fig. 8 illustrate, the divergent knocking tendencies of these engines became apparent; that is to say, the valve-in-head engine knocked at low speeds and the L-head engine knocked at high speeds.

Thus, in order to satisfy both engine types, the overall antiknock level of this fuel – at both low and at high speeds – would have to be raised. If the fuel were to be used in only one of the engine types, it is probable that either the high- or the low-speed performance could be improved by reblending the fuel. This, however, would result in a fuel that was more satisfactory for one engine but less satisfactory for the other.

■ Amount of Tetraethyl Lead Varied

A more composite picture of the effect of engine type on fuel antiknock quality was obtained by using test fuels with various tetraethyl lead concentrations. In these tests, a saturated-type gasoline and a specially blended gasoline were used, each containing concentrations of tetraethyl lead ranging from 0.0 to 3.0 cc.

The resulting borderline knock curves are shown in Figs. 9 and 10. In each diagram, the heavy borderline curves *A, B, C,* and *D* are of the test fuels with various tetraethyl lead concentrations, as indicated. The light lines are the borderline curves for blends of reference fuels of varying ASTM octane number, indicated by the numbers 75, 80, and so on.

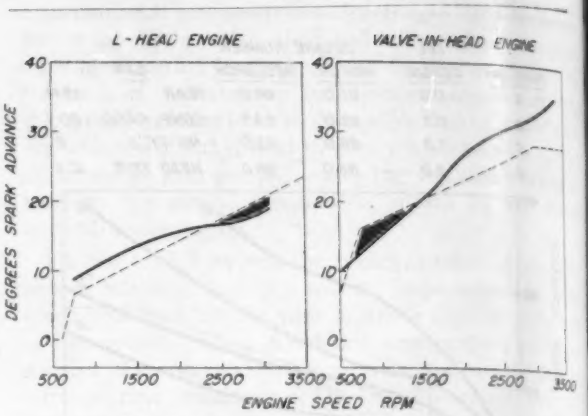
In the case of the saturated-type fuel the effect of engine type is quite noticeable. As an inspection of Fig. 9 reveals, an addition of 3 cc of tetraethyl lead to this fuel permits a spark advance of only 5 deg at 700 rpm in the valve-in-head engine, compared with an 8-deg spark advance in the L-head engine. On the other hand, the improvement at high speeds is more pronounced in the valve-in-head engine than in the L-head engine. In passing, it is interesting to note the similarity in shape between the borderline curves of the test fuel and the reference fuels. This is as expected since the secondary reference fuels[9] in common use are saturated-type gasolines.

When borderline curves of secondary reference fuels are obtained in conjunction with curves of the test fuel, it is possible to obtain the octane value of the test fuel at any speed by interpolation and also to obtain the octane requirement of the engine throughout the speed range considered. This latter can be determined by noting the speed at which the borderline curves of the various reference-fuel blends intersect the distributor-spark-advance curve. In this case, because of the similarity in type between the test and the reference fuels, there is little change in the octane value of the test fuels with speed. It should be noted, however, that the octane requirement of both test cars varies with speed, as shown in Table 5.

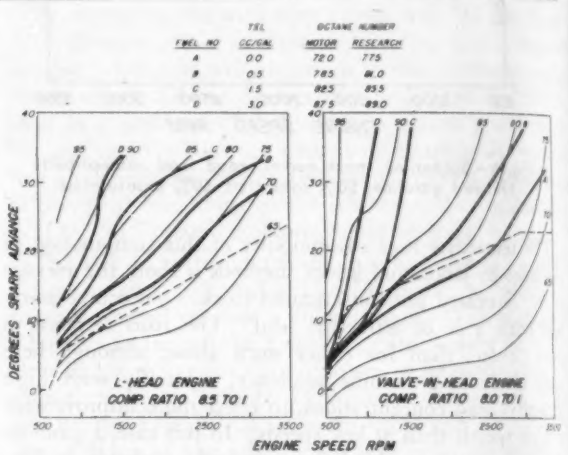
Table 5 – Variation of Octane Requirement with Engine Speed

L-Head Engine		Valve-in-Head Engine	
Speed, rpm	Octane Requirement	Speed, rpm	Octane Requirement
1000	73	1000	83
2000	72	2000	79
3000	66	3000	75

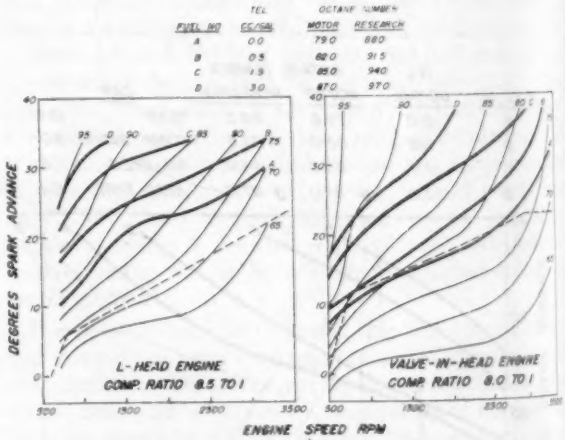
[9] See ASTM Designation D357-43T, "Tentative Method of Test for Knock Characteristics of Motor Fuels."



■ Fig. 8 – Borderline knock curves – blended gasoline



■ Fig. 9 – Borderline knock curves – saturated type of gasoline



■ Fig. 10 – Borderline knock curves – specially blended gasoline

Table 5 indicates the general need for fuels having good low-speed antiknock quality, since the antiknock requirements of an engine in terms of octane numbers are generally higher at low speeds than at high speeds.

From the shape of the reference-fuel borderline curves, with respect to the distributor-spark-advance curves, it is apparent that a change in the basic spark advance will affect the octane requirement of both engines considerably

more at low than at high speeds.[10] For this reason, a discussion of fuels based on octane numbers throughout the entire speed range is somewhat confusing and of little practical value to the vehicle operator. We believe, therefore, that a more usable conception of fuel antiknock quality is obtained by evaluating the test fuels in terms of allowable spark advance, inasmuch as adjustments in spark advance are generally the most practical means of obtaining knock-free operation with a given fuel.

Turning now to the tests with a specially blended gasoline, we see the effect that various concentrations of tetraethyl lead in this fuel have upon the allowable spark advance. This is shown in Fig. 10. The blended gasoline used is the same as that cited in Fig. 7.

As an inspection of Fig. 10 indicates, the borderline curves for this fuel, clear and with various tetraethyl lead additions, are not at all similar to the curves of the reference-fuel blends. They represent a gasoline type that satisfies both engines remarkably well, since the borderline curves tend to parallel the distributor-spark-advance curves. Yet, the basic difference in engines is still noticeable, the valve-in-head engine being more severe in low-speed rating compared with the L-head engine.

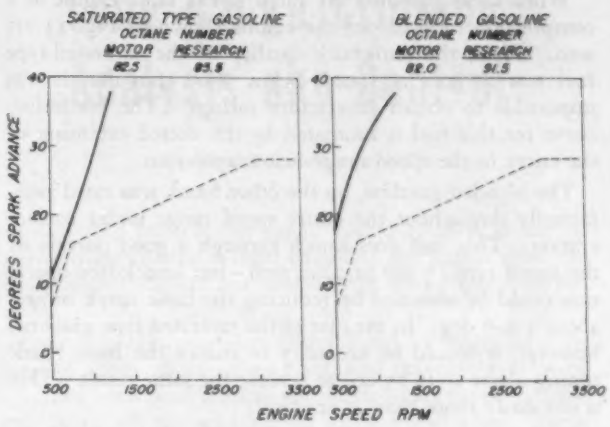
This specially blended gasoline, moreover, has an excellent road tetraethyl lead susceptibility, which permits a relatively great increase in spark advance at low speeds. In the case of fuel *D*, containing a concentration of 3 cc of tetraethyl lead, a spark advance of 10 deg at 700 rpm was made possible in the valve-in-head engine. This compares with a spark advance of only 5 deg at the same speed in the same type of engine, using a saturated-type fuel. But once again, the difference in engines must be noticed. Against a spark advance of 10 deg at 700 rpm in the valve-in-head engine, the addition of 3 cc of tetraethyl lead permits a spark advance of 20 deg in the L-head engine.

■ Effect of Compression Ratio

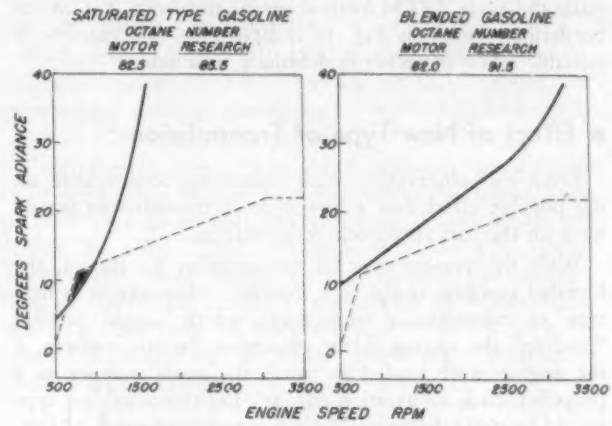
In evaluating the influence of engines upon the antiknock performance of gasoline, the effect of compression ratio must be considered. In this connection, to obtain a reasonable rating by the borderline method, the curve of the test fuel should not be displaced too far above or below the distributor-spark-advance curve. The effect of compression ratio is shown in Figs. 11, 12, and 13, in which a comparison between the saturated-type gasoline and the specially blended gasoline is made at three different compression ratios, using the valve-in-head engine.

The borderline curves shown in Fig. 11 illustrate the effect of using fuels that are too good for an engine, or of using an engine whose antiknock requirement is too far below the antiknock value of the test fuels. In this case, a valve-in-head engine with a 6.5 compression ratio was used, and the contrast between the borderline curves and the distributor-spark-advance curves is obvious. Not indicated in this diagram is the relatively great difference in road antiknock quality between the two types of test fuels.

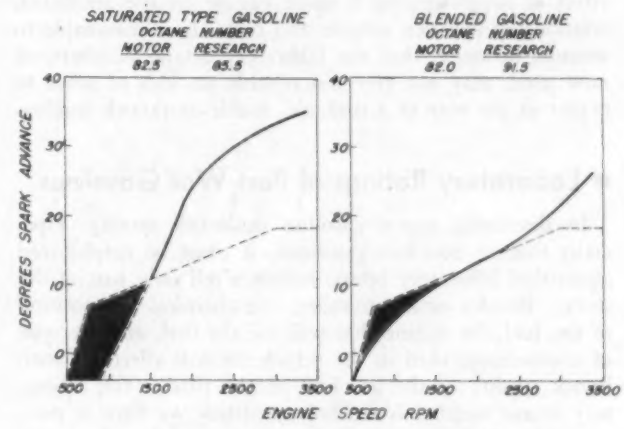
For a good comparison between these two test fuels, an engine compression ratio of 8.0 is about right. Such a comparison is shown in Fig. 12. Here, it will be seen, the saturated-type gasoline knocks slightly at low speeds, while its high-speed characteristics are much better than necessary. To obtain knock-free operation using this saturated-type gasoline, it is necessary to retard the basic timing by about 4 deg. On the other hand, the blended gasoline does not knock at all and is satisfactory for this engine with the standard spark timing.



■ Fig. 11 – Borderline knock curves – valve-in-head engine, 6.5 compression ratio



■ Fig. 12 – Borderline knock curves – valve-in-head engine, 8.0 compression ratio



■ Fig. 13 – Borderline knock curves – Valve-in-head engine, 9.0 compression ratio

[10] See SAE Transactions, Vol. 36-49, May, 1941, pp. 193-204: "1940 Road Detonation Tests. (Compiled from Report of Cooperative Fuel Research Committee), by J. M. Campbell, R. J. Greenshields, and W. M. Holaday.

When these gasolines are rated in the same engine at a compression ratio of 9.0, the results shown in Fig. 13 are seen. Here, the antiknock quality of the saturated-type fuel was so poor at speeds below 1500 rpm that it was impossible to obtain satisfactory ratings. The borderline curve for this fuel is estimated by the dotted extension of the curve in the speed range below 1500 rpm.

The blended gasoline, on the other hand, was rated satisfactorily throughout the entire speed range under consideration. This fuel does knock through a good portion of the speed range – 500 to 2200 rpm – but knock-free operation could be obtained by reducing the basic spark timing about 3 to 6 deg. In the case of the saturated-type gasoline, however, it would be necessary to reduce the basic spark timing about 10 to 15 deg to achieve the same result. This is obviously more than is practical.

In the previously described test with an engine of 9.0 compression ratio (see Fig. 13), the comparison – between the unsaturated type and the blended gasoline – involved two separate types of fuels which nevertheless had essentially the same ASTM method octane number. Yet, as the borderline curves in Fig. 13 indicate, the one gasoline is passable while the other is definitely unsuitable.

■ Effect of New Type of Transmission

From this observation, it is interesting to speculate on the possible effect that a new type of transmission would have on the fuel antiknock performance.

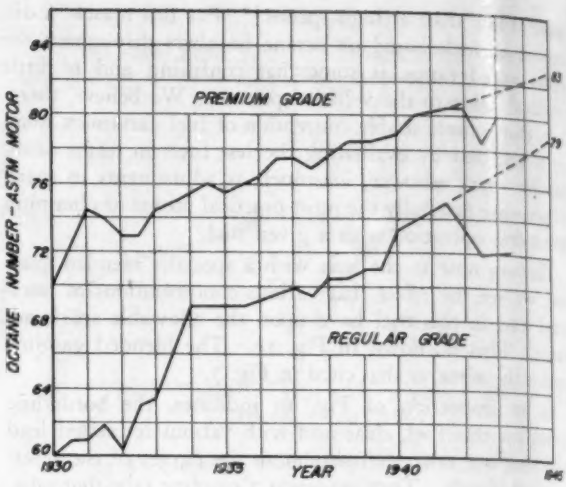
With the present type of transmission, as shown, the blended gasoline is the type desired. However, if a new type of transmission were used, which would prevent "loading" the engine below 1500 rpm – in other words, if the engine were loaded in much the same manner as a propeller loads an aviation engine – the choice of fuel type would be quite different. With this new transmission, low-speed antiknock quality would cease to be important, and the saturated-type gasoline might be the preferable fuel.

We do not have data available to illustrate the possible effect of supercharging a small engine on the antiknock relationship between vehicle and fuel. It is reasonable to assume, however, that the laboratory octane numbers, as now used, may not give too reliable an idea of what to expect in the way of actual and usable antiknock quality.

■ Laboratory Ratings of Post-War Gasolines

In discussing motor gasoline antiknock quality – especially that of post-war gasolines – it must be emphasized again that laboratory octane numbers tell only part of the story. Besides octane number, the chemical composition of the fuel, the engine that will use the fuel, and the type of transmission used in the vehicle all will affect the antiknock quality of the gasoline on the road. Yet, laboratory octane number is the only yardstick we have at present with which to estimate the post-war antiknock quality of gasoline, because there are, of course, no post-war cars in which to test the newer fuels.

Within these limitations, then, we can make some predictions about post-war antiknock quality of gasoline. In making any prediction, it is usually necessary to establish certain qualifying conditions. We are assuming, therefore, that the war will not end until the new refinery equipment now under construction will have been completed and, therefore, available for the production of post-war gasoline.



■ Fig. 14 – Projected octane number to 1945

By projecting the past trends in the antiknock quality of motor gasolines, values of 83 ASTM octane number for premium fuel and of 79 ASTM octane number for regular-grade fuel would be indicated in 1945. This is shown in Fig. 14. Such extrapolation, however, makes no allowance for the effect of the new wartime refining equipment.

As previously indicated, if all excess post-war aviation gasoline were dumped into motor gasoline, it would result in an increase of only a few octane numbers. Further, this gain would be expensive. While the present wartime capacity of aviation gasoline is a military secret, it must be remembered that sizable quantities are required to affect the average of an industry which produces gasoline at the rate of 1,600,000 bbl or 67,200,000 gal daily.

Instead, it is likely that much of the new refinery equipment will be converted to the production of motor gasoline components. Since this new equipment so far has been used only to make aviation fuel, it is necessary to estimate the effect this conversion will have on the national average of motor fuel antiknock quality. After considering the opinion of many oil company technologists, it is probable that the new wartime refining equipment, when used for making motor gasoline, will permit the production of gasolines with octane numbers for each of the several grades, as indicated in Table 6. For comparative purposes, similar data on the octane number of average pre-war

Table 6 – Post-War and Pre-War Gasolines Compared

Grade	Post-War Gasoline (National Average) % of Total Production	Octane Number Motor	Research
Aviation	6–8	100	
Premium	10–20	85–88	93–98
Regular	60–75	78–80	83–88
Third Grade	5–15	68–70	70–74

Grade	Pre-War Gasoline (National Average) % of Total Production	Octane Number Motor	Research
Aviation	2	91–100	
Premium	10	80	88
Regular	78	75	80
Third Grade	10	64	65

gasolines are also given. The table also shows the percentage of total production for each of the various grades of gasoline prior to the war and assigns upper and lower percentages that may be expected for each grade of gasoline.

Aviation fuel in the post-war period is expected to account for 6-8% of all gasoline produced, against only 1-2% before the war.[11] This is about a fourfold increase and reflects the effects of wartime expansion on future aviation gasoline demands.

■ Factors To Be Considered

Opinions vary in regard to the amount of premium, regular, and third grades of gasoline that will be sold in the post-war period. A number of factors other than refining equipment that will be available must be considered in arriving at an estimate. Post-war prosperity, or the lack of it, for instance, will affect public demands for gasoline quality. Other factors that must be considered include the probable condition of engines in older cars and the antiknock requirements of new cars. Therefore, because of post-war uncertainties, the amount of premium-grade gasoline sold may be anywhere between 10 and 20% of total production. It is not likely, however, that the octane number of premium gasoline would be similarly affected. New engines designed to use the new premium gasoline will have a stabilizing influence on antiknock quality. In this connection, it is interesting to note that the octane number of post-war premium gasoline – estimated at 85 to 88 (ASTM Motor method) – is comparable in antiknock quality to the 87-octane aviation gasoline sold prior to the war. It is, therefore, conceivable that many of the post-war airplanes, estimated at 500,000[11] by 1950, will be designed to use premium-grade motor fuel rather than a special aviation gasoline.

■ Gasoline Quality

As to gasoline quality, the figures in Table 6 indicate that regular-grade motor fuel will be about as indicated by the extrapolation of past trends of ASTM Motor method octane numbers in Fig. 14, but that premium grade will be better than normally expected. In passing, it should be noted that the CFR Research octane ratings of post-war motor fuels (premium and regular) are expected to increase to a greater extent than the ASTM Motor method values. This increase in sensitivity or the relatively greater increase in CFR Research octane value results from the use of new equipment such as catalytic cracking, which produces fuels higher in aromatic content than usual thermal cracking processes.

It is our opinion that the increase in ASTM octane numbers, indicated in Table 6, is not altogether a true indication of the actual improvement in road antiknock quality. Using their new equipment and processes, the oil refiners will be able to control product type much better than before the war. This, combined with an increased knowledge of engine requirements, will allow the petroleum industry to produce gasoline blends which will perform considerably better than indicated by laboratory octane numbers.

[11] Estimated by Civil Aeronautics Administration.

Long-Range Cruising Analysis and Control

continued from page 197

engined ship reduces the range available. The saving in fuel consumption on a dead engine is more than offset by the increase in power and specific fuel consumption on the others.

Fourth, maximum range is produced by manifold pressures of about 29 in. at sea level, and decreasing value of 1 in. per 3000 ft of altitude for single-speed superchargers, and 29 in. all the way up on turbo installations. These map's normally produce about 130 to 140 bmep.

Fifth, trim angle of the ship is frequently a good index of economical cruising attitude, especially on big boats. While this item will vary with the model airplane, 4 to 5 deg is ordinarily near optimum.

Sixth, center of gravity control produces a decided influence on airplane economy. Too far aft CG locations due to personnel or cargo placement has resulted in losses of as much as 6% in miles per gallon.

Even though tremendous strides toward the ultimate have been taken in the past few years, the future must produce even more attention to details; details which will reduce the specific fuel consumption a last 5%, details which will provide engine cooling without the excessive drag of cowl flaps, details in instruments by which flight-power control may be more accurately effected with increased reliability and decreased weight. A tabulation of these details is endless.

APPENDIX

List of Formulas

1. $\text{tas} = \text{ias} \times 1/\sqrt{\sigma}$

2. $V_{tw} = \dfrac{\text{tas}\sqrt{\sigma}}{\left(\dfrac{\text{gw}}{W_s}\right)^{\frac{1}{2}}}$

3. $P_{tw} = \dfrac{\text{thp} \times \sqrt{\sigma}}{\left(\dfrac{\text{gw}}{W_s}\right)^{\frac{3}{2}}}$

4. $V/ND = \dfrac{\text{tas} \times 88}{\text{prop rpm} \times \text{prop diameter}}$

5. $C_p = \dfrac{\text{bhp/engine}}{2 \times \sigma \times \left(\dfrac{D}{10}\right)^5 \times \left(\dfrac{N}{100}\right)^3}$

6. $\text{thp}_{\text{req}} = P_{tw} \times \left(\dfrac{\text{gw}}{W_s}\right)^{\frac{3}{2}} \times \dfrac{1}{\sqrt{\sigma}}$

7. $\eta C_p = \dfrac{\text{thp/engine}}{2 \times \sigma \times \left(\dfrac{D}{10}\right)^5 \times \left(\dfrac{N}{100}\right)^3}$

8. $\text{bhp} = \dfrac{\text{bmep} \times \text{rpm}}{k}$

9. $k = \dfrac{792{,}000}{\text{cu in. displacement}}$

Some Physical and Wea
POROUS-CHROMIUM

by TRACY C. JARRETT
American Hammered Piston Ring Division
Koppers Co.

THE use of porous-chromium-plated rings in aircraft engines has improved the performance and increased the number of hours between overhauling.

Mr. Jarrett presents data showing the effect of temperature upon the decrease in hardness of the chromium plate. The time and temperature data on the chromium plate indicate that the decrease in hardness occurs in a very short interval of time, particularly at temperatures around 700 F.

Actual field tests show that ring life is increased and barrel wear reduced when using porous-chromium-plated rings in the top groove of each piston.

THE AUTHOR: TRACY C. JARRETT (M '43), chief metallurgist, American Hammered Piston Ring Division, Koppers Co., is author and co-author of several technical articles as well as inventor of the Jarrett Metallurgical polishing and lapping machine. An alumnus of the Dakota School of Mines and Harvard Engineering School, where he received his B.S. and doctor's degrees respectively, Dr. Jarrett was assistant chief metallurgist at the American Optical Co. before joining Koppers Co.

IN using porous-chromium-plated rings, one must take into consideration both the physical and wear characteristics of the plated surface. Field tests have shown that there are tremendous advantages in using chromium-plated rings.

The object of this paper is to study the effect of temperature on the hardness of chromium plate, and to report field tests of engines using porous-chromium-plated rings in two different types of barrels. The specifications covering the plating, finishing, and physical dimensions of the plated ring have been discussed in a previous article.[1]

Effect of Heat-Treating Chromium Plate

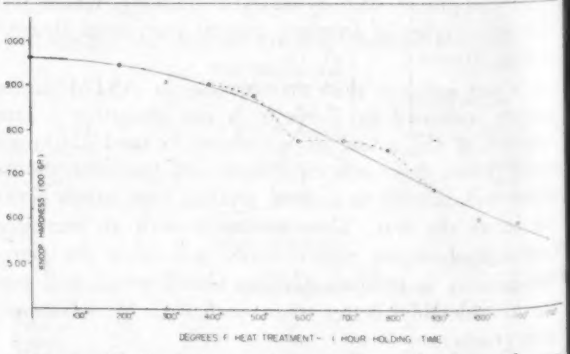
In view of the success of porous-chromium-plated rings and the importance of their application, it was decided to study effect of temperature on the hardness of the plate. It was thought that the chromium plate, due to its extreme hardness, would show good wearing qualities in the first hours of engine operation, but decrease rapidly as the time of engine operation increased. This, as will be shown later, did not occur.

Standard plain gray-iron piston rings, porous-chromium-plated to a thickness of 0.005 in., were used for test purposes. The test specimens were heat-treated at various temperatures ranging from 100 to 1200 F. The specimens were held at these temperatures for a period of 1 hr, as previous experiments indicated that the greater part of the hardness change occurred within this time. The porosity was removed from the face of the ring (cylinder contacting surface) leaving the hard chromium highly polished, for hardness measurements. The Tukon tester with a 100-g load was used to determine the Knoop hardness numbers, and the results of these tests are shown in Fig. 1.

The decrease in hardness is gradual up to a temperature of 500 F, with a sudden drop taking place between 500 and 600 F. The hardness then appears to remain at this new level up to a temperature of 700 F, followed by a second sudden drop in hardness, ending at 900 F. The drop from 900 to 1200 F is gradual, ending with a total hardness change of 400 Knoop hardness numbers.

The sudden drops in hardness were rather unexpected, and a time-hardness curve was run on plated specimens at a constant temperature of 700 F to see if further changes would take place if held at this temperature for a long period of time. Fig. 2 shows that 90% of the drop in hardness occurred within the first hour, and no further changes took place after 5 hr at this temperature. It is impossible, at this time to offer an explanation why the two changes occur, but it is hoped, with the research now going on, that in the very near future, a reason for the changes will be forthcoming.

There have been occasions when a drop in hardness was observed on rings installed in engines using superheated



■ Fig. 1 – Effect of temperature on Knoop hardness of porous-chromium plate

[This paper was presented at the SAE War Engineering-Annual Meeting, Detroit, Mich., Jan. 13, 1944.]

[1] See *Mechanical Engineering*, Vol. 65, September, 1943, pp. 633-635: "Cylinder and Ring Life with Porous-Chromium-Plated Rings," by T. C. Jarrett.

haracteristics of
LATED RINGS

team. The hardness of the chromium surface dropped 300 points in Knoop hardness numbers after a few hours of operation, indicating that the rings had operated at an extremely high temperature.

Knowing that high temperatures would soften the chromium plate, it was decided to investigate the effect of engine operation on chromium-plated rings in the field in regard to the softening and wearing characteristics. Aircraft engines, the largest users of chromium-plated rings, were chosen as the ones to investigate.

Field Test Results

In an aircraft engine, it should be pointed out that only the top ring of each piston is chromium plated and the remaining compression and oil rings are plain gray iron.

Field test data were obtained from different sections of the country and from different types of engines. Tables 1-6 show the barrel wear of different types of engines, some using plain cast-iron rings in the top groove, others with porous-chromium-plated rings in the top groove; hardness of the remaining chromium plate; and the approximate amount of wear on the chromium-plated ring.

From the data presented, it is obvious that cylinder wear is greatly reduced by the use of one porous-chromium-plated ring in the top groove. The porous-chromium-plated ring shows little wear under abnormal dust conditions, in fact the apparent life of porous-chromium-plated rings is exceptionally good.

In view of the fact that the chromium-plated ring possessed a hardness of approximately 850 to 950 Knoop hardness numbers originally, Tables 1 to 6 indicate that the chromium plate did not soften to any great degree. This would also show that apparently no unusually high temperatures had been reached in these engines.

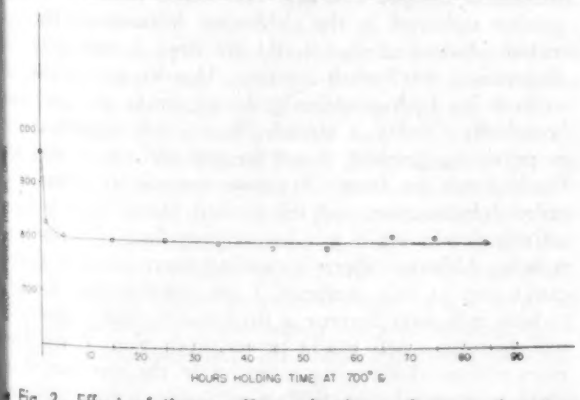


Fig. 2 – Effect of time on Knoop hardness of porous-chromium plate at 700 F

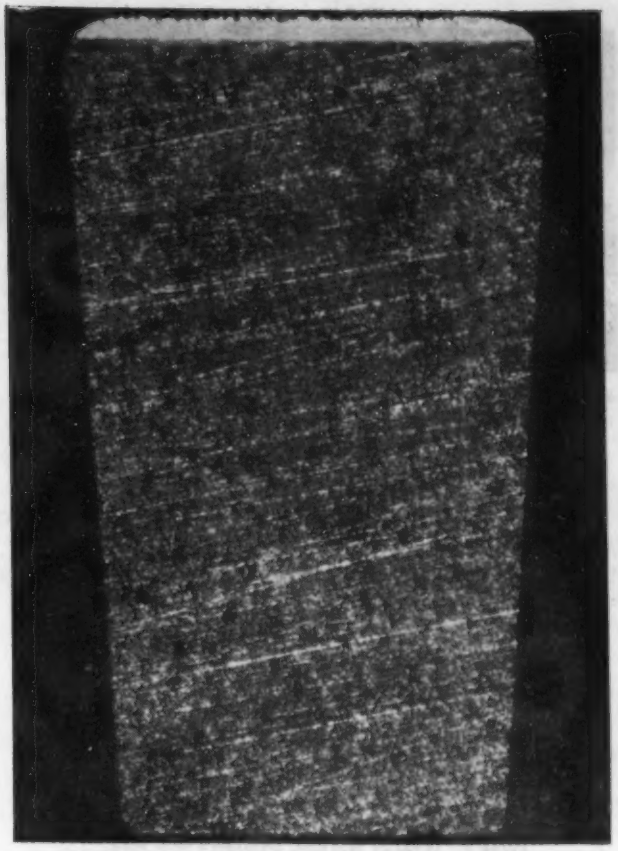


Fig. 3 – Cross-section of porous-chromium-plated cast-iron ring – nital etched – reduced from photomicrograph taken at 25X

Photographs showing cross-sections of a finished porous-chromium-plated ring before test are shown in Figs. 3 and 4, and of a plated ring which had operated under abnormal dust conditions for a period of 447 hr, Fig. 5. (See p. 224.)

Table 1 – Wear of Chromium-Molybdenum Cylinders Run with Porous-Chromium-Plated Rings[a]

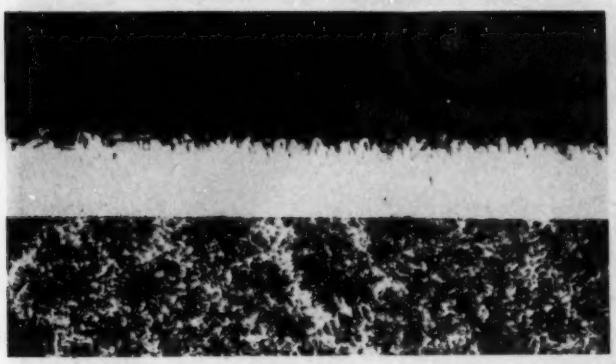
Engine No.	Approximate Horsepower	Operating Time, hr	Cylinder Wear Measured on Diameter, in.
A	2000	34	0.002
B	2000	94	0.002
C	2000	106	0.002
D	2000	192	0.002
E	2000	245	0.004
F	2000	310	0.003
G	2000	315	0.003
H	2000	460	0.002
I	2000	539	0.001
J	2000	590	0.003

[a] Operated under abnormal dust conditions.

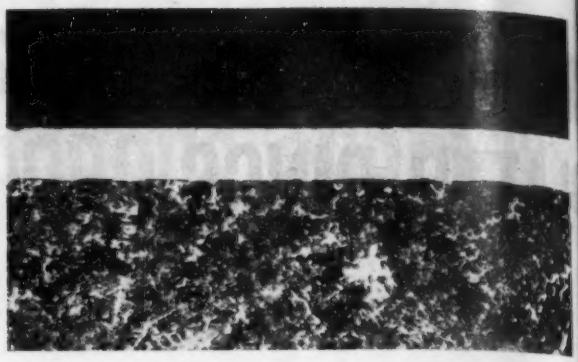
Table 2 – Wear of Chromium-Molybdenum Cylinders Run with Plain Cast-Iron Rings[a]

Engine No.	Approximate Horsepower	Operating Time, hr	Cylinder Wear Measured on Diameter, in.
A	2000	18	0.005
B	2000	70	0.008
C	2000	107	0.012
D	2000	110	0.006
E	2000	210	0.005
F	2000	224	0.010
G	2000	268	0.008
H	2000	301	0.016
I	2000	307	0.007

[a] Operated under abnormal dust conditions.



■ Fig. 4 – Cross-section of porous-chromium-plated cast-iron piston ring – nital etched – reduced from photomicrograph taken at 100X



■ Fig. 5 – Cross-section of porous-chromium-plated cast-iron ring after 447 hr of abnormal operating conditions – nital etched – reduced from photomicrograph taken at 100X

Table 3 – Wear of Porous-Chromium-Plated Rings Operating against Chromium-Molybdenum Barrels

Engine No.	Approximate Horsepower	Knoop Chromium Plate Hardness [a]	Operating Time, hr	Remaining Chromium [b] Thickness, in.	Average Ring Wear, in.
A	2000	799	36	0.0053	0.000
B	2000	823	68	0.004	0.001
C	2000	885	109	0.003	0.002
D	2000	810	168	0.0034	0.0016
E	2000	780	192	0.0026	0.0029
F	2000	859	315	0.0028	0.0022
G	2000	839	425	0.0018	0.0032
H	2000	807	453	0.0018	0.0032

[a] The hardness of chromium plate was tested on the Tukon tester using 100-g load and after engine operation. Original hardness before test 850 to 950.
[b] Average 0.005 in. chromium thickness on the radius originally.

Table 4 – Wear of Nitrided Cylinders Run with Porous-Chromium-Plated Rings

Engine No.	Approximate Horsepower	Operating Time, hr	Cylinder Wear, in.
A[a]	1200	28	0.001
B	1200	593	0.000

[a] Operated under abnormal dust conditions.

Table 5 – Wear of Nitrided Cylinders Run with Plain Cast-Iron Rings [a]

Engine No.	Approximate Horsepower	Operating Time, hr	Cylinder Wear, in.
A	1200	32	0.009
B	1200	81	0.006
C	1200	195	0.008
D	1200	285	0.008
E	1200	407	0.009
F	1200	559	0.010

[a] Operated under abnormal dust conditions.

Table 6 – Wear of Porous-Chromium-Plated Rings Operating against Nitrided Barrels

Engine No.	Approximate Horsepower	Knoop Chromium Plate Hardness [a]	Operating Time, hr	Remaining Chromium [b] Thickness, in.	Average Ring Wear, in.
A	1200	879	150	0.0039	0.0011
B	1200	867	593	0.0035	0.0018
C	1200	912	745	0.0040	0.001

[a] The hardness of chromium plate was tested on the Tukon tester using 100-g load and after engine operation. Original hardness before test 850-950.
[b] Average 0.005 in. chromium thickness on the radius originally.

Diesel Engines in Wartime Navy

continued from page 201

tear of the engine parts in service and to aid in the quick determination of any trouble afloat. The Station's services and experts are at all times accessible to the industry. Close cooperation with this institution cannot fail to be beneficial to the development of the diesel engine.

This brings me to another plea to the engineering fraternity and to the diesel industry. As a nation, we must encourage only the best to survive. Hence, most of the peculiar whims and hobbies of an individual engineer or an organization must of necessity be permitted to make room for the fittest.

The war has taught all of us the advantages of standardization. Mass production is impossible without it. As I envision the future, American industry must organize around mass-production procedure. The old job shop is no more, because the rest of the world is just waiting to out "job shop" us, hoping we will play according to those rules. We simply must not go back to minor-league tempo. The members of this Society must take their hair down, bury the hatchet in certain cases, and go to work based on a global horizon.

One other thought and I am finished. I want to mention gas turbines. For years this new type of prime motor was in the same category as the bumblebee. Both were classified by the scientists as "incapable of survival" – the bee because it could not fly, and the gas turbine because no thermodynamic cycle could be devised to make it competitive with the internal-combustion engine. Fortunately, the bumblebee escaped and flew ever since. Lately there was another upheaval in the alchemists' laboratory. The gas turbine discovered that it did not need a new cycle. In desperation, the French inventor, Hondri, gave it the job to save his hydrogenation process, and it did just that beautifully. Today, it already plays a very important role in providing gasoline, if not for you and me, at least for Uncle Sam's air force. Recently it made its debut in a railroad locomotive, and the United States Navy is now actively developing it as a future propelling plant for war vessels. Although there is nothing more concrete that I can report at this moment, I am positive that the gas turbine will soon become a firm reality, and probably it will open up a new era in power production. It still provides a most challenging incentive to the metallurgist to create a material that will stand up in hell if need be. Like television, it is just around the corner.

Fundamental Requirements of POST-WAR TRANSPORT AIRPLANES

by CHARLES FROESCH
Chief Engineer,
Eastern Air Lines, Inc.

AS the military power of the United Nations begins to bear inexorably on the Axis, winning the global war we are now fighting becomes a matter of time, the length of which depends upon our individual and collective effort. So that while our primary purpose now is to win the war, we must, nevertheless, take the necessary steps to win also the peace to follow. Just as air power is playing a most important military role, so will commercial air transportation serve to keep the most permanent peace the mind of man can conceive.

With this aim in mind, we must begin now to formulate comprehensive plans, national and international in scope, for a wide economic utilization of the airplane, beginning with the immediate post-war period.

The first post-war decade will be the most critical period for the air transport industry. Upon our correct interpretation of the economic factors created by the war, as affecting scheduled air transportation, our factual analysis of the problems we face and their judicious, technical, and practical solution will depend either success and accelerated growth or retarded development.

In the preparation of any future plan, trends must be assumed from known data and expectations. Call it crystal gazing if you like, yet, if these trends are based on sound fundamental principles and common sense, we should arrive at a picture which ought materially to come true.

For instance, any analysis of air traffic possibilities cannot be fully predicated on the traffic volume of existing surface means, but instead on what the speed of the airplane can develop as a supplementary service rather than competitive activity. Attractively priced, vacation travel by air can become an important source of revenue by reaching a market yet untapped. Feeder service provided with suitable equipment can render a real business service, and long-range operation with its great inherent time saving offers unlimited opportunities, to say nothing of carrying all first-class mail by air.

We must stretch our imagination and look far beyond what we thought was realizable before the war. In this respect I am reminded of an early meeting of the Aero Club of France which I attended in 1912 in Paris, just before I came to the United States. Breguet was delivering a lecture on this same subject and predicted that airplanes would soon fly at 150 kmph (approximately 93 mph). Just then I overheard someone say that it was too bad someone always made a fool of himself at those meetings.

Nevertheless, within 30 years we have seen a *predicted high speed* become a *commonly accepted landing speed.* Similar comparison could be made with other performance and design characteristics with equally astounding variations. I hope my predictions can be more accurate.

That the rate of advancement shall be the same in a general way is for the next generation to determine by review, but it behooves us now to take stock of what has been accomplished during the last few years, interpret these advances in the light of future designs, and project them into the future.

■ ■

THE first few years after the war will be a transition period of changing from wartime mobilization of airlines to normal commercial practices.

This period will be followed by an intermediate period of readjustment and expansion to meet increasing passenger and cargo traffic demands, which, Mr. Froesch feels, will last four to six years.

The next period will be the development era of efficient large-scale operation of improved airplanes and ground handling.

The transport airplane standardization of the past seven years with the DC-3, Mr. Froesch says, will give way to the development of five distinct types of airplanes in scheduled air transport during the second post-war period.

Mr. Froesch outlines the fundamental requirements for these transport airplanes and gives their basic specifications. He also lists the principal design criteria and operating requirements most desirable from the airline operator's point of view.

■

THE AUTHOR: CHARLES FROESCH (M '16) has been chief engineer for Eastern Air Lines, Inc. since 1934. Except for a brief period when he served with Mack Trucks, Inc., as bus engineer and head of gas-electric bus design, Mr. Froesch has been identified with the field of aeronautics. During World War I he was an inspector of aircraft, aircraft engines and aeronautical design. Later, he joined Fokker Aircraft Corp. as project engineer, and became successively assistant chief engineer and general service manager. He has long been active in SAE Metropolitan and Baltimore Sections, and in 1943 was a member of the SAE Aircraft Activity Committee.

[This paper was presented at a meeting of the Metropolitan Section of the SAE, New York City, June 17, 1943.]

A recitation of the most important aeronautical developments advanced by the war effort would include:

1. Considerable operational experience with given airplanes at gross weights substantially greater than their previously certificated weights.
2. Higher wing loadings and higher speeds.
3. Large airplanes and adaptation of laminar airflow to wing design.
4. Larger powerplants.
5. Universal adoption of tricycle landing gear.
6. Wide use of 100-octane fuels.
7. Air cargo and cargo handling experience.
8. Transport gliders.
9. Airport expansion program – in both size and number.
10. Radio developments which will revolutionize air traffic control.
11. Undreamed of airframe and engine production capacity and increased knowledge for speedy fabrication by proper tooling.

These developments have profound implications and will speed up the return of the air transport industry to normal business after the war. In this respect it is believed that we shall pass through three successive periods of post-war developments which might now be outlined as follows:

a) The *transition* phase of changing from the wartime mobilization of the airlines to normal commercial operation. During this period, which may last three to four years, the airlines will gradually discard their Douglas DC-3 equipment, now used almost exclusively, and replace it temporarily with modified versions of the Douglas C-54 Skymaster, Curtiss C-46 Commando, Lockheed C-69 Constellation, and C-60 improved Lodestar, and possibly with the Curtiss C-76 Caravan when feeder service is attempted. There may be several other types available depending upon the length of war.

In other words, it will be a period when available military cargo equipment, having better performance, shall be adapted until newer and more modern airplanes embodying the latest technical lessons of the war shall be manufactured.

While these military cargo airplanes were originally conceived for commercial operation, their designs have been modified to meet military requirements. It is fortunate, however, that these changes have not been basic in nature and can thus allow comparatively rapid reconversion for commercial use.

Inasmuch as the immediate post-war period will be of great economic importance in determining the place that we, as a nation, shall hold in airborne commerce, it is hoped that the military services will see fit to consider the commercial aspects of air transport in their plans.

b) The *intermediate* period of readjustment and expansion to meet increasing passenger and cargo traffic demands, which might last four to six years. This time element will depend primarily upon the ability of the aircraft manufacturer to produce the more modern aircrafts specified by the air transport industry.

c) The *development* era of efficient large-scale operation of improved small and large airplanes and ground handling, which will establish the air transport industry as big business.

Each phase presents distinct equipment problems. The railroad suffered these growing pains, the truck and the motorbus did likewise, and so will we.

This outline of fundamental requirements for transport airplanes covers basic specifications for airplanes as may be needed during the second or readjustment period.

It is safe to say, at the outset, that the day of transport airplane standardization, as we have known it for the past seven years with the Douglas DC-3, is over, for as we grow we must provide a public service ranging from local passenger service operation to long-range passenger service, as well as air cargo facilities to the shipper of merchandise requiring speed of movement.

Obviously, we cannot furnish this wide range of service efficiently and economically with a single type of airplane or even possibly two and expect to progress.

We can, therefore, without prophesying, contemplate the operation of five distinct types of airplanes in scheduled air transport which will probably come quite close to the following general definitions:

1. *Feeder Service* – An airplane of small capacity designed to stop every 25 or 30 miles to pick up passengers, mail, and express for transportation between small communities or transfer to mainline points. It should be able to take off and land with safety from any 1500-ft square cow pasture to reduce the time taken to and from the airport.

When terrain or sparsely populated areas do not warrant the economical use of this type of airplane, its substitution with pickup or helicopter service might be well contemplated, if such areas are found entitled to air service due to inadequate facilities. The low mail volume and passenger traffic would probably require that this service be established under government subsidy as it is not believed the helicopter will be classed as a mass transportation vehicle for some time to come.

2. *Local Schedule* – A larger airplane designed to stop every 100 to 200 miles, depending upon the territory served and connecting with the feeder service. It should be able to take off and land from medium-size airports and provide adjustable accommodations for both passengers and cargo.

3. *Limited-Stops Service* – An airplane of excellent speed performance and approximately twice as large as the local schedule airplane. As this airplane would operate in and out of modern airports, speed and maximum payload for the required range would be more important than short take-off and landing characteristics. Convertibility from dayplane to sleeper plane may be desired for some services.

4. *Long-Range Operation* – A large airplane with maximum speed performance and best possible L/D speed for economy of flight. Designed for 1000 miles between stops and upward. Having large cargo capacity and providing day and night de luxe passenger accommodations. The size of this airplane, which might be a very large flying boat for transoceanic operation, will depend primarily on the size of powerplants available, as it is not believed that the use of more than four or perhaps six engines will be found practicable.

5. *Cargo Service* – An airplane designed as simply as possible, able to stand a lot of abuse, virtually a "flying box car" both in general appearance and size.

I am disposed to believe that the basic characteristics of these airplanes should follow the specifications shown in Table 1, which might bear some explanatory comments.

It will be noted that airplanes Nos. 1, 2, and 5 are listed as bimotors, airplane No. 3 as either a bimotor or quadrimotor, and airplane No. 4 as a quadrimotor only. This is dictated by their size for airplanes Nos. 1 and 2, and lower

operating cost per ton-mile for airplane No. 5. After all, it is not a question of how many engines are used, but rather as to how much power reserve is available with one engine inoperative to retain safety of operation.

I appreciate that safety is relative, as every form of travel involves an accident risk, but, by the same token, we cannot compound safety indefinitely and economically survive.

A crew of 2, that is, pilot and copilot, is considered sufficient for airplane No. 1, with the addition of a cabin attendant for No. 2. The third member of the crew in airplane No. 5 is a cargo handler, whose duty should be to supervise the arrangement of the cargo for speediest ground handling, as well as sorting cargo in flight for the same reason and take care of all necessary paper work.

A high-wing location was selected for airplane No. 1 because, with frequent stops, reduction in ground time is most important if a satisfactory utilization factor is to be obtained. Furthermore, it simplifies loading ramps and carts. Required propeller ground clearance may also dictate this requirement. An additional feature, of early days sales value, is the unobstructed view of the terrain flown over by the passengers. Similarly, a high-wing location is deemed mandatory for the cargo airplane to enable its design with cargo loading doors height from the ground corresponding to truck platform heights which currently vary from 22 to 45 in. from the ground.

With improved automatic powerplant controls, it may not be necessary to use a flight engineer on airplane No. 4 and we might thus combine his duties with those of the copilot or navigator.

I have indicated the possibility of using diesel engines for the long-range airplane No. 4 and the cargo airplane No. 5 instead of the standard gasoline type. Fuel costs represent today approximately 12% of all direct flight costs, and therefore any possible reduction permitted by the substitution of a cheaper fuel is worth investigating. This applies particularly to these two airplanes. To the former because of its high total consumption and the fact that it may be possible to obtain a lower specific fuel consumption and thus a saving in weight of both engine and fuel for long flights, and to the latter because low cost of operation is absolutely essential if we are to compete with express rates of 10 to 12¢ per ton-mile, which include pickup and delivery costs.

Furthermore, as airplanes grow in size there must be a point when the diesel type of powerplant becomes definitely more economical, just as it is found with watercrafts. The future will decide where this demarcation line is to be found.

Wing loading has not been specified as with modern auxiliary lifting devices, the resultant C_L with flaps in approach or landing position, is of greater importance.

Cruising speeds have been listed at 50 to 55% of rated power, rather than a percentage of take-off power, which is only what might be termed "spot" power. Furthermore, this power ratio appears to give most reliable operation and lowest engine operating and overhaul costs.

Having thus outlined the basic types of airplanes needed to furnish the kind of service which can be of most benefit to the air traveler, we can now list the principal design criteria and operating requirements most desirable from the airline operator's point of view.

First of all, any transport airplane must be conceived around its passenger cabin and cargo compartment for passenger operation, or its cargo compartment if specifically designed for air cargo service. Great care must be taken in the layout of the passenger compartment to provide maximum comfort without space waste, yet every appointment should have its function to perform. There should be no unnecessary frills which cost weight and often do not produce the harmonizing effect sought by the decorator.

Interiors should be designed for ease of maintenance and servicing. This applies particularly to seats, floor covering, lighting fixtures, lavatory, and galley. Satisfactory solutions are often difficult to realize with lightness. Galleys are excellent examples. It is difficult to design a galley for maximum utility, light weight, and still retain attractiveness.

I remember tackling this problem two years ago with the qualifications that it would be necessary to serve 21 five-course hot meals in 1 hr.

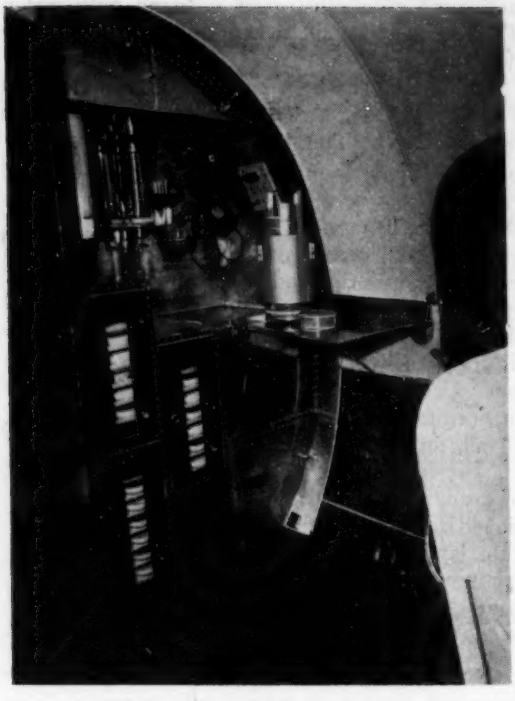
The only way to do it was virtually to prepare the trays in advance, keeping the hot courses ready to serve, thus combining speed, maximum sanitation, and lightness. The result is shown by Figs. 1 and 2, which illustrate a complete galley installation weighing but 18 lb per passenger, including food. This instance is cited to emphasize the importance of weight saving in cabin design. Appointment weights have increased from about 30 lb in 1930 to over 200 lb in 1942, and the trend must be reversed for obvious reasons.

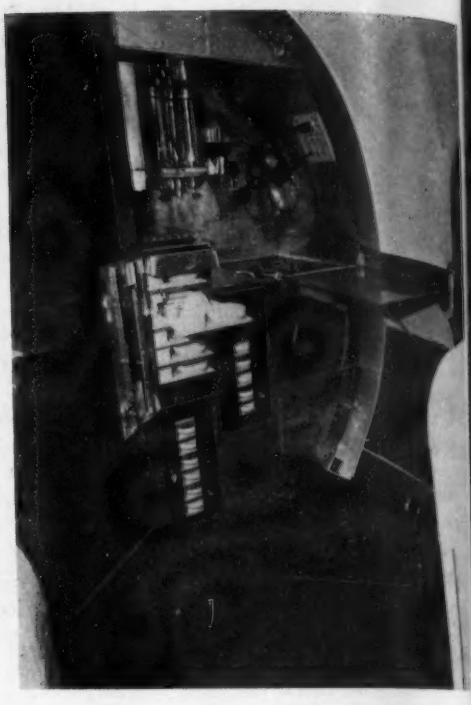
Longitudinal seat spacing should not be less than 39 in.

Table 1 – Proposed Characteristics for Future Transport Airplanes

	Airplane No. 1 Feeder	Airplane No. 2 Local	Airplane No. 3 Limited	Airplane No. 4 Long-Range	Airplane No. 5 Cargo
A. GENERAL SPECIFICATIONS					
(a) Gross weight, lb	12,000	30,000-35,000	40,000-55,000	75,000+[a]	50,000-60,000
(b) Useful load, lb	4000	10,000-11,500	13,000-18,000	25,000+	25,000-30,000
(c) Passenger capacity, day	8 to 10	25 to 30	32 to 40	60+	—
night	—	—	16 to 20	30+	—
(d) Cargo capacity, cu ft	100	400 to 450	600 to 750	1000+	2000
(e) Number of engines	2	2	2 or 4	4[b]	2
(f) Powerplant	Gasoline	Gasoline	Gasoline	Gasoline or diesel	Diesel
(g) Power loading, lb per hp	10	12	12 to 14	12 to 14	14
(h) Wing arrangement	High wing	Low wing	Low wing	Low wing	High wing
(i) Flight crew, number	2 pilots	2 pilots 1 cabin attendant	2 pilots 1 radio operator 1 or 2 cabin attendants	2 pilots 1 radio operator 1 navigator[c] 2 cabin attendants	2 pilots 1 cargo handler
B. PERFORMANCE					
(a) Cruising speed at 50 to 55% of rated hp	150 to 160	200 to 220	240 to 250	250+	200+
(b) Stalling speed (flaps in approach position)	50	80	80	90	80
(c) Take-off distance to clear 50-ft obstacle, ft	1000	2500	3000 to 3500	4000 to 5000	2500
(d) Range plus ¾ hr reserve, miles	300	750	1500	2000+	1500
(e) Single-engine ceiling	Meet CAA terrain requirements				

[a] Maximum gross weight depending on available engines. Might be flying boats in larger sizes.
[b] Possible use of powerplant installations from airplanes Nos. 2 or 3.
[c] May also act as flight engineer.





■ Figs. 1 (left) and 2 (right) – Light and compact galley – serves 21 five-course hot meals in 1 hr

(although 37 in. may be sufficient for the feeder airplane) and seat cushion width not less than 20 in. There should be no seat accommodating more than two passengers side by side. The center aisle width should have a clear passage of at least 18 in. and a floor to ceiling height of 6 ft 6 in. minimum.

Baggage compartments for domestic operation should have sufficient capacity to accommodate a minimum of 4 cu ft of *usable* space per passenger and at least 7 cu ft per passenger for long-range and foreign service.

Any additional space desired for mail and cargo should be provided on the basis of 10 lb per cu ft.

Baggage and cargo compartments should be readily accessible, designed and located to permit rapid loading and unloading. This also applies to all doors for both cabin and cargo compartments. .

Secondly, every part of the airplane should be designed for utility. No operating function must be either sacrificed or compromised, yet the designer must continually bear in mind simplicity of design, as unnecessary complications are costly to manufacture and maintain. Remember that service costs continue throughout the life of the airplane and that all accessory, complicated, and unnecessary mechanisms require not only man-hours but additional maintenance shop space and equipment with their fixed charges.

Economy of operation is governed by the general efficiency of the airplane as a whole, while best performance depends upon the proper relationship of certain characteristics such as wing and span loading, power loading, drag, powerplant efficiency, and ratio of useful load to gross weight, which should be at least 33% for passenger airplanes and correspondingly 45 to 50% for strictly cargo airplanes.

In so far as speed is concerned, the problem for the designer is how fast can we afford to go profitably without sacrificing safety, reliability, and utility.

Third, more intelligent thinking must be given to the maintenance and service of various components of transport airplanes. The grouping of controls and accessories in one location has been the limiting factor in speedy service and overhaul (see Figs. 3, 4, and 5). Serious consideration must be given to the decentralization of major components to reduce maintenance time. This must be carried further by providing each system or component with quickly removable accessory panels and inspection doors to allow rapid change in case of failure. This objective can and must be obtained by careful design without excessive weight penalty.

Fourth, it is felt that a 50- to 60-passenger capacity should, perhaps, be the limit for domestic operation as highest permissible system speed and frequency of schedule are more important from the traffic viewpoint than capacity. Furthermore, the time taken to load and unload a greater number of passengers and their baggage becomes an important ground handling problem, to say nothing of the greater number of cabin attendants required for personal service in flight.

Another factor which determines maximum passenger capacity for domestic operation is the ratio of equipment reserves needed to provide adequate and reliable service. Larger airplanes would mean a smaller number of airplanes for a given traffic volume and less spares which might not be conveniently located to eliminate operation delays.

Larger airplanes for long-range and transoceanic service could embody double or even triple decks in multiples of 50- or even 100-passengers each in order to keep the CG movement within a satisfactory range.

In studying operating costs, it becomes apparent that increase in size does not always imply lower cost except for long-range operation when full advantage can be taken of provisional gross weight and higher system speed.

As we increase the size of an airplane beyond a certain point, the utilization factor may drop due to the time taken for inspection checks, maintenance, and service unless the time between such periods is increased. It is true that specialized maintenance may partially offset but until we know more about it, the best way appears to

crease size gradually and acquire experience to teach w to use substantially larger airplanes efficiently.

While it is not within the scope of this presentation to cuss detail specifications we should, nevertheless, point a few fundamental requirements for future transport planes as dictated by past operating experience.

I have listed them as follows:

1. *Structures* – All-metal cantilever structures of smooth-in construction have been found highly satisfactory dur- the last 10 years and there is no reason why their *use ould be discontinued*.

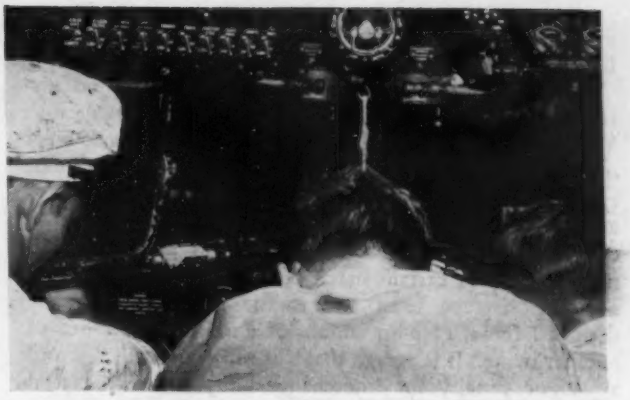
Wing loadings are still going up and will continue to do as long as aerodynamic improvements in auxiliary lift- devices for take-off and landing will permit such crease without unduly raising the maximum approach d landing speeds as determined by Civil Air Regulations d pilot reaction time.

The weight and balance engineer should consider the cation of the rearward CG of greater importance than e forward location of the CG when determining the sition of wing with reference to the loadable area of the selage because, as a rule, empty-weight increases occur- ng between the experimental airplane and the production rticle are generally caused by powerplant changes which ove the CG forward.

Tricycle landing gears also dictate that most rearward G location be the primary consideration in the original sign to eliminate any possible balance condition in oper- ion which would require the use of tail supports.

Furthermore, it is easier, as a rule, to move the fuselage engines forward to correct for satisfactory CG travel an to shift the wing rearward.

■ Fig. 3 – Radio mechanic standing: checking fuses in junction ox; plane mechanic in cockpit wishing to get out of ship; plane echanic sitting: working on hydraulic panel; plane mechanic kneeling: checking cables under floor



■ Fig. 4 – Left: electrician changing bulb in instrument spot light; middle: plane mechanic pumping up hydraulic pressure; right: electrician working in electrical instrument panel junction box

■ Fig. 5 – Front left: electrician working on landing gear signal lights; front middle: instrument mechanic checking flight instrument panel; front right: plane mechanic operating controls for man out on engine; back row: radio mechanic checking radio

With increasing wing aspect ratio, the expression of permissible CG travel in per cent of aerodynamic chord becomes meaningless. It is suggested that it be expressed in relation to length of loadable area instead, in order to give a clearer picture of loading possibilities, or better still, in inches of allowable movement.

Integral fuel tanks for the larger airplanes offer a substantial saving in weight and service cost if sufficient care is exercised in their detail design.

Refueling and reoiling from the under wing surface should deserve consideration to eliminate expensive servicing equipment and promote maintenance-personnel safety.

Wing panel and tail surface joints should be designed for ease of removal. Leading edges should be readily demountable where it is necessary for inspection and servicing of lines, controls, and accessories. An adequate number of inspection doors should be provided in all structures. This has been sadly lacking in past transport design.

Detachable wing and stabilizer tips must be made interchangeable. This also applies to all movable control surfaces.

Wing structures should be conceived to accommodate de-icing by means of engine exhaust heat. Mechanical wing de-icing as now done is aerodynamically inefficient and

expensive when it is considered that leading edge shoes last but two or, at the most, three winter seasons.

2. *Powerplant* – It is essential that powerplant installations from the firewall forward be made interchangeable. With large transport fleets comprising the five basic types of airplanes discussed, it should be possible to switch one type of powerplant from a given type of airplane to another type within given power ranges. A hypothetical case would be that of switching, let us say, a 1200-hp installation of a certain make of aircooled engine from a 4-engine airplane to a twin-engine transport originally designed for a 1350-hp engine of a similar type, but of another make. It is agreed that this procedure will demand a weight penalty for interchangeability, but the capital investment saved in spares and overhauling equipment should more than offset it, particularly if such powerplant changes can be made, as they must, in a few man-hours.

We certainly cannot afford to spend as much time in the future as we do now in removing an engine for overhaul and installing a newly overhauled engine in its place. This ought to be done in minutes rather than hours when it is considered that this operation is carried out, roughly, 20 to 30 times during the useful life of the airplane.

Every engine must, in the future, be equipped with a torquemeter and a flowmeter, with corresponding indicators in the cockpit for accurate and efficient power control.

From the overhaul viewpoint, powerplants should be self-contained from the firewall forward in order to permit duplication of flight check on the test stand after engine overhaul to eliminate the present expensive procedure of first bench testing the engine, then checking the installation on the airplane and, finally, flight checking it. This means mounting the oil tank and all accessories required for complete powerplant check after overhaul on the engine mount.

Aircooled engines must be improved to provide uniform operating cylinder-head temperature under all power and airflow conditions.

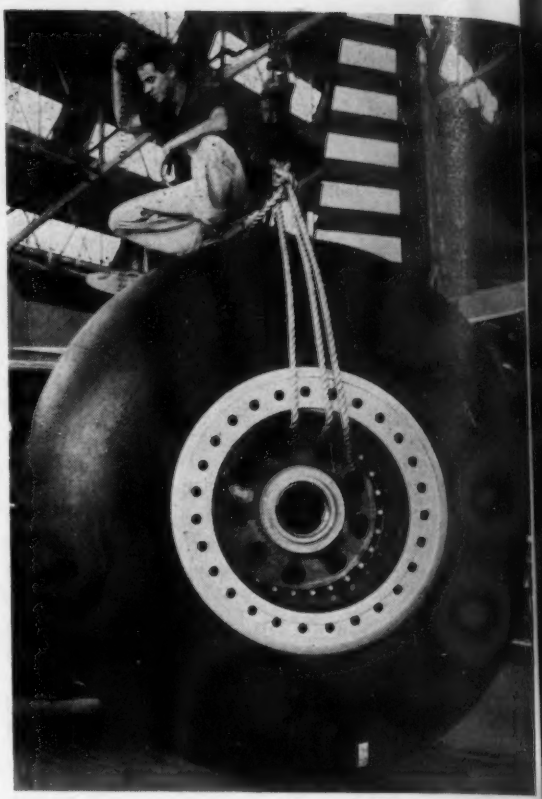
Excessively rich mixtures now used to provide some degree of cooling during take-off and climb cannot be tolerated in the future. It is an economic waste which must be eliminated particularly for feeder and local schedule operation where take-off and climb time comprise a high percentage of total engine time.

Powerplant design must be properly coordinated between the engine and airplane designers. It is useless to have a light engine if it results in a heavy installation. This applies particularly to the arrangement of accessories, provisions for cylinder and oil cooling for minimum drag.

3. *Landing Gear* – While I desire to place no limitation on the type of system used either to extend or to retract the landing gear and also wing flaps, that is, pneumatically, hydraulically, or electrically, Civil Air Regulations dealing with one-engine-inoperative take-off clearly indicate the necessity of accomplishing these operations in the shortest time permissible.

If hydraulic or pneumatic means are used, the plumbing should be as simple as possible, as door retraction and latch operation can easily be accomplished mechanically. There has been a tendency to complicate the hydraulic system unduly, with resultant excessive maintenance.

Serious thought must be given to simplicity of control, and proper indication should be provided to enable the flight crew to check the operation of the system quickly and detect trouble at a glance.



■ Fig. 6 – Wheel and tire for large airplane for long-range operation

Hydraulic pressures of the order of 3000 psi have been found satisfactory, and the weight saving incurred by the high pressure warrants its continued application.

Brakes have always been a source of high maintenance cost. In his effort to keep weight at a minimum, the designer is often inclined to be marginal. Furthermore, when a prototype passes the experimental stage, goes through series production, and is operated for a few years, its weight has increased substantially and the braking capacity has become inadequate.

As the capacity of friction brakes cannot increase proportionately to the gross weight and the square of the landing speed without involving a substantial increase in weight and installation problems, aerodynamic braking becomes a necessity with large airplanes.

Landing wheel tire life averages, at present, about 65 hr with an average of 1- to 1¼-hr flights between landings. As taxiing involves about three miles per sequence, the total tire mileage between changes is less than 2000 miles, which is a ridiculously low figure.

This short life is caused by abnormal tread wear and side-wall failure at the time contact is made by the landing gear wheels and the runway in landing when sudden acceleration of the wheel assembly up to the landing speed occurs, the necessary kinetic energy being absorbed by tread rubber abrasion.

As we increase the number of landings per hour in the case of local schedule operation and increase the size (Fig. 6) and cost of tires used on large airplanes for long-range operation, it becomes economically imperative that prerotation of landing gear wheels be embodied in any new design to reduce maintenance costs as well as operations delays.

Fig. 7 – Glare makes some of the instruments on this panel difficult to read

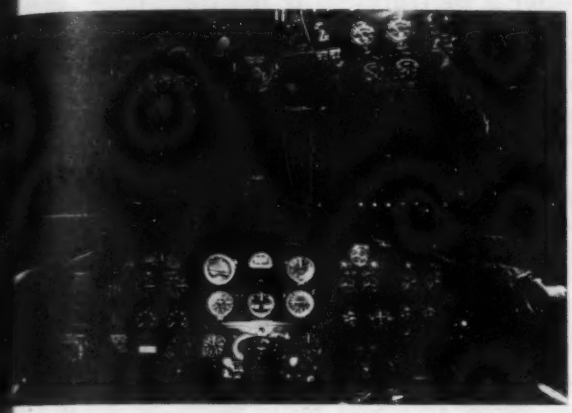


Fig. 8 – Instrument panel of DC-3 cockpit – all cockpit lights on bright, fuel and oil warning lights out

Fig. 9 – Panel with fluorescent lighting – all instruments are visible, with no glare

Prerotation must be up to 100% of landing speed as the kinetic energy involved, which is reflected in tread wear, is proportional to the square of the speed.

It is felt that every new post-war transport should be equipped with tricycle landing gear for obvious reasons. First of all, the expectation of all-weather landings and take-offs will require the best ground stability possible, especially when negotiating crosswinds. Second, the elimination of the landing flare will permit better landings to be made, particularly in poor visibility. Third, a level attitude is essential for sleeper operation and cargo service.

4. *Cockpit* – The pilot's cockpit offers a prolific source for improvements. The first item is unquestionably that of lighting. (See Figs. 7 and 8.) There is no excuse today for radium painted instrument dials and control knobs. Furthermore, the contrast between the indirectly lighted dashboard of the modern automobile and the crude arrangement used in current transport airplanes is so striking as to make one wonder why such inadequacy has not received earlier consideration.

Recent experiments and application of fluorescent lighting (see Fig. 9) have shown that it is possible to obtain diffused illumination, of controlled intensity, to reduce glare on the windshield panes, as well as pilot eye fatigue. This is a step in the right direction.

Rearrangement of controls according to selective functions must receive further thought. This will call for the location of emergency controls, such as propeller feathering buttons and master switches, in positions permitting almost instinctive operation on the part of the flight crew, but still secure their inadvertent operation.

The windshield should be birdproof. Perhaps one of the best solutions for this problem is to design the windshield as a part of the fuselage contour without impairing visibility. If the design of the fuselage is such that the windshield must protrude through it, then it should not have any surface normal to the line of flight in order to avoid direct head-on collision with birds, as a 12-lb bird at 250 mph means the absorption of over 25,000 ft-lb of kinetic energy and involves a heavy structure and windowpanes to stop it successfully.

All windowpanes should be mounted flush, not only to reduce drag, but to keep ice accretion to a minimum.

Windshield wipers have been found very satisfactory for landing in heavy rain and should be considered as standard equipment if they can be applied. Satisfactory means of preventing external icing and inside fogging is a mandatory specification.

Windshields must be watertight. This has been a specification of long standing and one which has seldom been met.

The nose of the fuselage or a large portion of it immediately ahead of the instrument board and controls should be quickly removable for ease of service.

5. *Flight Controls* – With the exception of limited service and long-range operation airplanes, no automatic pilot need be provided as cargo and local schedule airplanes will be used to a large extent for pilot training, and its elimination from these airplanes will serve to increase their payload and reduce their maintenance costs accordingly.

Automatic pilot installations should be operated from hydraulic pressure derived by a separate pump to ensure system cleanliness, eliminate pressure-reducing valves, and keep pressure cycles of the main hydraulic system at a minimum.

For the sake of simplification and reliability, it is believed that any control force boost needed should be obtained by aerodynamic means, such as the inclusion of servo tabs, rather than complicate the controls with servo motors depending on a source of power subject to possible failure.

Operation of propeller pitch reversing for aerodynamic braking when provided must be interlocked with the throttle and landing gear extension controls to prevent their inadvertent operation at high power.

While on the subject of propeller controls and operation, present Civil Air Regulations indicate the necessity of quick feathering in case of engine failure at take-off, particularly with twin-engine airplanes.

6. *Electrical System* – A minimum potential of 24 v should be specified even for the smaller airplanes. With increased functional reliability of modern generators and engines, storage batteries become more or less voltage regulators and can be of small capacity as, for air transport operation, an external source of supply is always available for plug in when the airplane is on the ground.

As airplanes grow larger in size, there is a point where it becomes more economical from both weight and operating cost standpoints to use an auxiliary powerplant aboard.

7. *Fire Protection* – Adequate fire protection must be

provided not only for the powerplant installation but also for the passenger cabin and cargo compartment.

I might mention at this time that careful study must be given to the proper ventilation of fuel compartments and their insulation from surrounding areas to reduce the collection of gasoline fumes, which in many instances have been a very serious fire hazard.

The satisfactory application of any pressure fire extinguishing system rests upon the immediate and accurate detection of any fire aboard and subjects it to a foolproof detector installation, rapid in operation and of light weight.

These are seven fundamental requirements which must be embodied in the first crop of strictly transport airplanes to be manufactured after the war and to be used until such time as the air transport industry can take stock of its immediate post-war progress and reflect upon its future long-range needs, particularly as to the size of the individual types of airplanes I discussed earlier.

In view of amazing advances in radio, it is difficult at this time to outline any specifications, except to say that modifications of the much discussed radar should give us an excellent post-war instrument landing system and perhaps provide what might be called a block system of the air.

Post-war transport radio equipment will undoubtedly include equipment for instrument landing, automatic compass, communications equipment, ultra-high-frequency radio range receivers, and one of the chief problems of the aircraft designer will be to locate and arrange this equipment properly to ease servicing and permit accessibility in flight when a radio operator is used.

Submerging of antennae should receive serious consideration to reduce drag and eliminate icing hazards.

There are some controversial requirements which we might briefly mention for the sake of future discussion.

Among these we can include cabin pressurization. No one will question its necessity for long-range operation where the importance of low drag at high altitude is reflected by high cruising speed and miles per gallon. However, the degree of pressurization must be accurately balanced between overall flight costs, which includes service and maintenance of equipment and effect of empty weight increase as indicated by revenue per ton-mile per hour.

It is not a question of flying in the stratosphere to overcome weather, but rather to select the most economical flight altitude in the lower region of the troposphere, between 20,000 and 25,000 ft and pressurize the cabin to maintain an altitude within which the use of oxygen will not be required nor cause excessive heart action for the occupants. Limited practical application indicates that 8000 to 10,000 ft is about such altitude, and flying at 25,000 ft will mean, roughly, 5 psi of pressurization. This should be our first step in this direction. When the efficiency and engine supercharging and propeller developments warrant higher flight altitudes and the use of heavier structures to maintain a higher pressure differential, we can then raise our standards accordingly.

The pressurizing of local schedule and feeder airplanes and those designed strictly for cargo service appear undesirable, as low cost of operation is the prime consideration. This calls for simplicity in design with resultant low purchase price and maintenance costs.

Another item which will require considerable thought is the heating system. Airline operators have certainly had enough trouble with steam heat and are ready to re-accept hot air as a source of cabin heat. Whether this should be accomplished by transfer of engine exhaust heat by heat exchangers or by providing an entirely separate and self-contained unit is subject to analysis. Here again, however, airplane size may determine the demarcation line. Perhaps some day someone will devise a self-contained unit for winter heating as well as summer cooling. Ground conditioning as now furnished is expensive.

Another point worthy of thought by the airplane designer is the elimination of fuel dump valves and their control on airplanes designed for local schedule operation and putting their weight in the landing gear, thus permitting a corresponding landing weight increase and perhaps eliminate the difference now existing between provisional and landing weights.

A troublesome problem today is that of weighing. Transport aircraft of the future should have their landing gear shock struts so designed as to indicate by means of remote pressure gages and strut piston deflection the weight of such airplanes in any loaded condition. Furthermore, it could be possible to determine quickly both weight and CG location for any load by suitable calibration of both main and tail wheel struts. All that is needed is a refinement of the basic parts already in the structure to provide the operator a foolproof loading method for even the most inexperienced personnel.

Last, but not least, it is extremely desirable that some degree of standardization be applied to parts subject to interchangeability and service replacement. Among such parts might be included navigation and landing light shields, door locks, cabin chairs, and berths with their equipment, lavatory accessories, cabin windows, towing attachments, powerplant mounting bolts and location, cockpits in so far as location of the various controls.

This is a very fertile field for investigation without transgressing on the initiative of the designer. With a large number of airplanes in operation, and we now think of thousands rather than hundreds, some degree of standardization is necessary. Besides, it is expected that equipment interchange between airlines will undoubtedly become common practice, especially with air cargo operation.

Some excellent work has already been done, as witnessed by the SAE Aeronautic Standards. Further research along this line must continue.

The Air Transport Association Committee on Future Aircraft Requirements has recently completed a study on operating costs and derived a standardized cost method to be used as a yardstick for comparing new types of transport airplanes. This method is based on current experience and the constants used will be subject to modification according to future experience. It was felt that it should prove helpful to the manufacturer to enable him to design new equipment to meet specific operating conditions without denying his opportunity to produce a superior article.

The air transport industry is facing a tremendous and revolutionary expansion before it matures as a stable business. It is hoped that it will be unhampered by restrictive regulations and always recognize safety and economy of operation as stepping stones to future progress using the increased air speeds as permitted by continued technical advances. Similarly, it is essential that the aircraft and engine manufacturers fully understand the problems faced by the air transport operator and join him to arrive at the most suitable solutions.

Special COLD-STARTING FUELS for Diesel Engines

by G. H. CLOUD and L. M. FERENCZI
Standard Oil Development Co.

A STUDY of fuels for the cold starting of diesel engines has brought the authors to the following conclusions:

1. Normal petroleum hydrocarbon fuels will permit the starting of the best automotive engines at temperatures not below about 20 F without supplementary starting aids.

2. Materials blended into normal diesel fuels in optimum concentration to improve cetane number and cold starting may give a reduction in minimum starting temperatures of probably not over 20-25 deg.

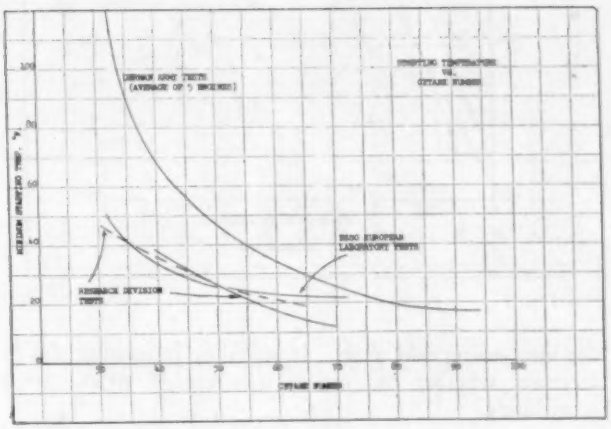
3. Ethyl ether introduced into the air stream of diesel engines (not including crankcase scavenge) will allow starting down to at least —40 F without additional starting aids, but provided the crankcase lubricant and cranking facilities give sufficient cranking speed.

■

THE AUTHORS: G. H. CLOUD (M '38) devoted his time, until 1941, chiefly to diesel fuel studies at Standard Oil Co.'s Esso Laboratories, where he has been employed since receiving his Doctor's degree from the University of Buffalo in 1935. He has recently expanded his work at Esso to include research on automotive gasolines. Mr. Cloud is the author of three other papers which he presented at SAE meetings. L. M. FERENCZI (J '42) joined the fuels research staff of Esso Laboratories in 1942, and shortly after represented his company as a civilian observer at the U. S. Army Ordnance's test grounds at Camp Shilo. Mr. Ferenczi, a graduate mechanical engineer from Lehigh University, was connected with the Pure Oil Co. for a year before assuming his present position.

THE results of studies made during recent years have shown ignition quality of normal diesel fuels to be the best index of starting performance. For a given diesel engine and lubricant, the higher the cetane number of the fuel the lower the minimum starting temperature, or for a given temperature the less cranking time required for starting. Fuel viscosity also affects starting but to a lesser degree than cetane number; the better starting is obtained with the lower viscosity fuels.

[This paper was presented at the SAE War Engineering Production Meeting (Annual Meeting), Detroit, Mich., Jan. 15, 1943.]

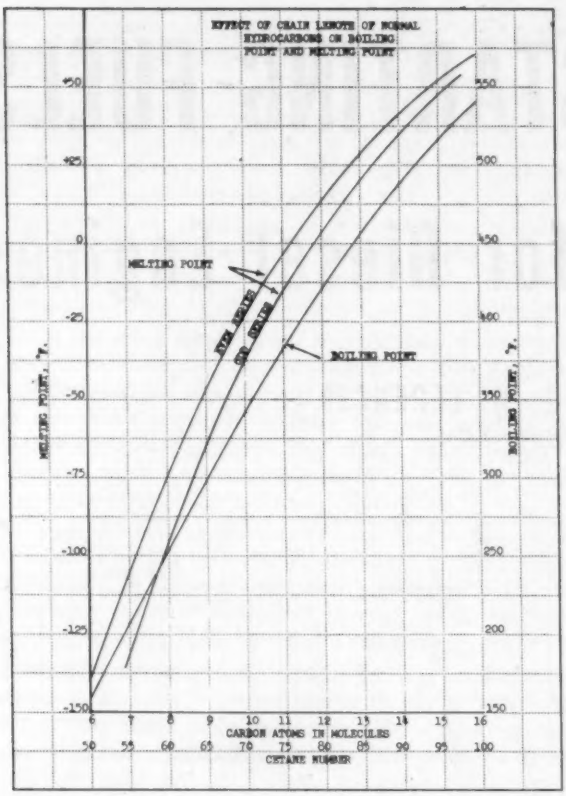


■ Fig. 1

The effect of cetane number is illustrated in Fig. 1 showing the relation between minimum temperature for a starting time of 50 sec, based on results obtained by the Esso Laboratories here and abroad, and on data published in Germany.[1] It is evident that the effectiveness of cetane number decreases as values of 55 to 60 are reached. They also show the general inability to obtain starting of diesel engines at subzero temperature with conventional hydrocarbon products even though their cetane numbers might be unusually high.

The limitations on the use of high-cetane-number hydrocarbons for obtaining starting at low temperatures is illustrated further by a consideration of the relationship between melting point and cetane number of the normal hydrocarbons, shown in Fig. 2. Since the normal hydrocarbons represent the ultimate in cetane number for a given molecular size, these data show the maximum cetane numbers possible for hydrocarbons in the range of C_7 to C_{16}. The change in cetane number with length of hydrocarbon chain shown is based on results obtained on C_7, C_{12}, and C_{16} hydrocarbons. From these data it is evident that the pure materials above 80 cetane number cannot be used for subzero starting, because their melting points are above 0 F. Pure cetane has a melting point of 60 F. It is interesting to note here that the melting points of the even series of normal hydrocarbons are higher than those for

[1] See *Oel und Kohle*, Vol. 36, March 1, 1940, pp. 78-89: "Untersuchungen über die Beziehungen der im Heereswaffenamt-Prüfmotor gemessenen Cetanzahl und dem Anlasswert von Dieselkraftstoffen in der Praxis," by P. Weber.


EFFECT OF CHAIN LENGTH OF NORMAL HYDROCARBONS ON BOILING POINT AND MELTING POINT
MELTING POINT
BOILING POINT
MELTING POINT, °F.
BOILING POINT, °F.
CARBON ATOMS IN MOLECULES
CETANE NUMBER

■ Fig. 2

the intervening odd series. It would be interesting also to ascertain whether the cetane numbers of series varied accordingly.

■ Effect of Ignition Promoters

The fact that melting-point characteristics of the hydrocarbons limit the effectiveness of cetane number as a means of obtaining fuels of good cold-starting properties leads logically to a consideration of ignition promoters (cetane-number improvers) as aids to cold starting. Some results obtained in cold-starting tests on a series of promoters are shown in Table 1. More complete data are shown in Fig. 3.

Table 1 – Effect of Promoters on Cold Starting

Promoter	Temperature for Start in 50 Sec with 5.8% Blend in Reference Fuel, F	Cetane Number Improvement in Blends 1%	5%
Reference Fuel (without promoter)	21	...	...
Secondary Amyl Nitrate	17	10	25
Isobutyl Nitrate	...	8	20
1, 1 Dichloro-1-Nitroethane	16	10	...
Diamyl Trisulfide	13	7	18

A digression here will permit an explanation of the test technique used in determining the starting characteristics of the fuels and blends used for illustration. As shown in Fig. 3, a series of starts was obtained for each fuel over an appreciable temperature range. By plotting starting time against engine block temperature a curve was obtained for each fuel, which permitted averaging out deviations due to individual tests. By selecting from the curves starting temperatures corresponding to some constant cranking time, a strict comparison between fuels can be made.

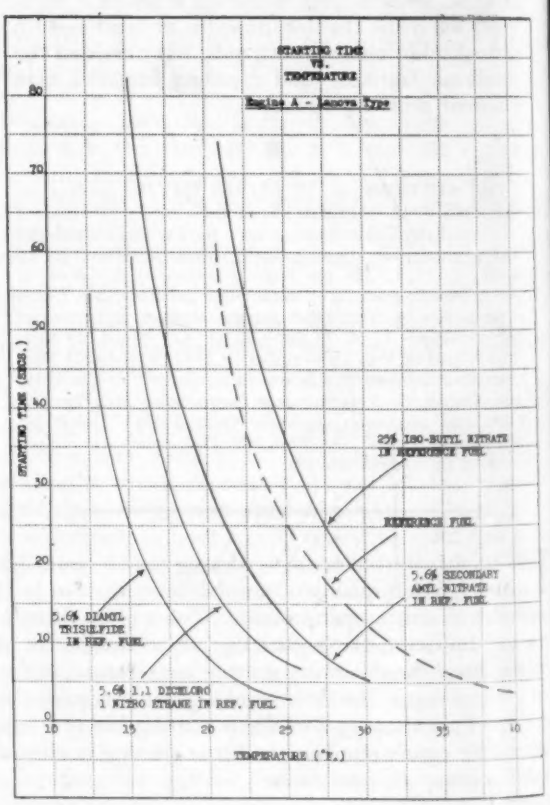
On the basis of a comparison of cetane improvement and starting performance, these data are somewhat confusing. The nitrates and chloro-nitroethane should have given better starting than the diamyl trisulfide. The results on the 25% blend of isobutyl nitrate shown in Fig. 3 are surprising and are inexplainable on the basis of cetane number, since the estimated cetane number of the blend was 100+.

Other miscellaneous tests are also of interest. For example, in tests at 40 F, a blend of 3% secondary amyl nitrate started an engine as quickly as did a straight fuel of similar cetane number. However, at 20 F, a straight fuel of 52 cetane number started as quickly as did the 3% amyl nitrate blend of 59 cetane number. At 13 F, a blend of the nitrate of 75 cetane number showed no better starting performance than did a straight fuel of 60 cetane number. From these results it seems evident that the cold-starting effectiveness of cetane improvement produced by addition of the nitrate decreases as the temperature is lowered. In order to determine whether the diamyl trisulfide would decrease in effectiveness at still lower temperatures, it was tested in a 25:75 hexyl ether-reference fuel blend, which gave starting some 9 F below that obtainable with the straight reference fuel. The results shown in Fig. 4 indicate no tendency for the effectiveness of the sulfide to fade as temperature is reduced.

■ High-Cetane Hydrocarbon Derivatives

Cetane-number determinations on a series of hydrocarbon derivatives showed some of the ethers to be high in cetane number, and this prompted cold-starting tests on these materials. Table 2 summarizes some data obtained on the straight-chain ethers.

Because 50 sec is about as long as an engine can be cranked with batteries and electric starter under normal


STARTING TIME VS TEMPERATURE
Engine A - Lenoir Type
STARTING TIME (SECS.)
25% ISO-BUTYL NITRATE IN REFERENCE FUEL
REFERENCE FUEL
5.6% SECONDARY AMYL NITRATE IN REF. FUEL
5.6% DIAMYL TRISULFIDE IN REF. FUEL
5.6% 1,1 DICHLORO 1 NITRO ETHANE IN REF. FUEL
TEMPERATURE (°F.)

■ Fig. 3

Table 2 – **Cold-Starting Tests of Straight-Chain Ethers**

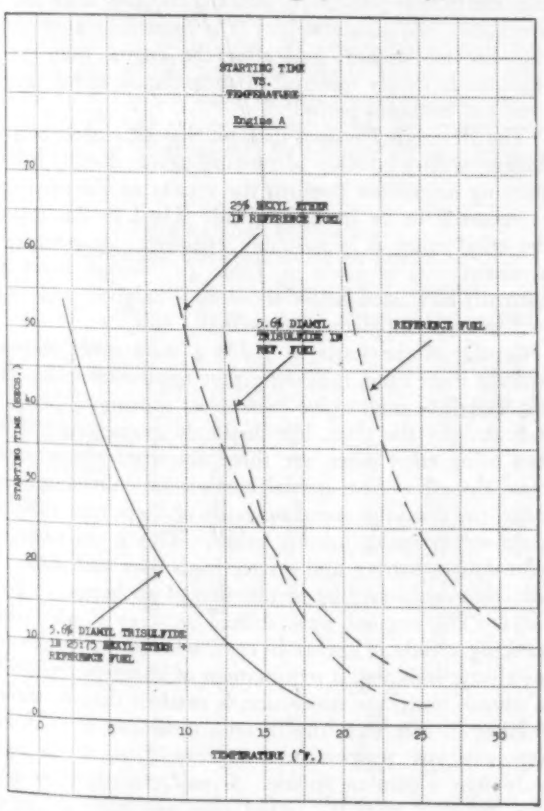
Fuel Tested	Cetane Number	Temperature Required for Start in Cranking Time of 50 Sec, F	30 Sec, F	20 Sec, F
Reference Fuel No. 1	53	33	36	39
Reference Fuel No. 2	74	21	24	27
Dimethoxytetraglycol Ether	140*	1	7	12
Diethoxydiglycol Ether	120*	0	6	10
n-Hexyl Ether	90*	0	6	10
n-Butyl Ether	85*	0	6	9
Ethyl Ether	53*	11	17	21

* Blending value in 50% concentration in Columbian gas oil of 40 cetane number.

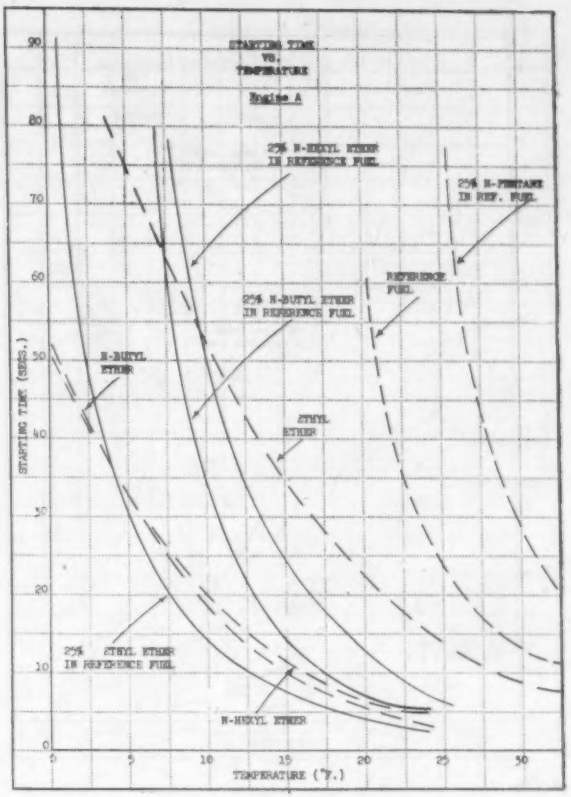
service conditions, the results in the 50-sec column probably furnish the most practical and valid comparison between materials tested. There are no appreciable differences in performance between the n-hexyl, n-butyl, dimethoxytetraglycol, and diethoxydiglycol ethers. All are appreciably superior to ethyl ether and markedly superior to the hydrocarbon fuels. By comparison of cetane numbers, the polyglycol ethers would be expected to perform better than the hexyl and butyl ethers. Reference fuel No. 2 would be expected to be superior and reference fuel No. 1 about equivalent to ethyl ether. Although cetane number has been shown above to be a good measure of starting performance of conventional hydrocarbon fuels, the lack of correlation between cetane number and starting performance of the ethers indicates that there are factors other than cetane number influencing their starting performances.

■ Ether-Hydrocarbon Blends

Comparative results obtained on hexyl, butyl, and ethyl ethers, and 25% blends of these in the 74 cetane-number reference fuel, are shown in Fig. 5. The 25% blend of


STARTING TIME VS. TEMPERATURE
Engine A
25% HEXYL ETHER IN REFERENCE FUEL
5.0% DIAMYL TRISULFIDE IN REF. FUEL
REFERENCE FUEL
5.0% DIAMYL TRISULFIDE IN 25/75 HEXYL ETHER + REFERENCE FUEL
STARTING TIME (SECS.)
TEMPERATURE (°F.)

■ Fig. 4



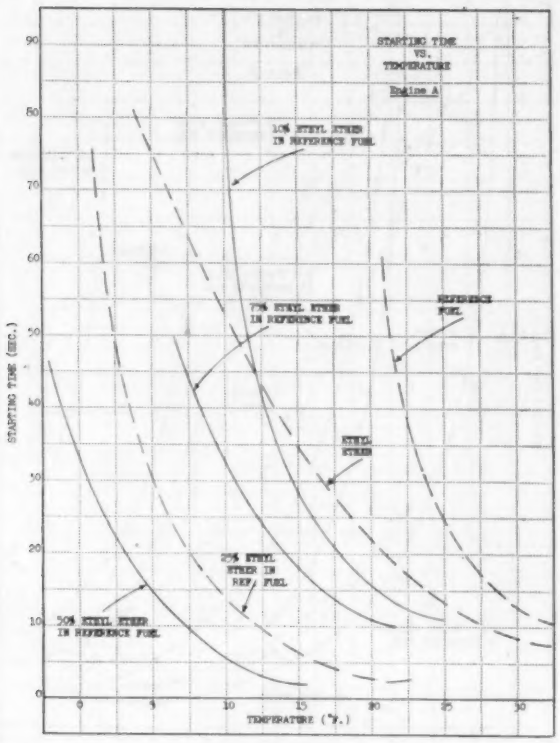
■ Fig. 5

hexyl ether performs better than would have been predicted from the performance of its constituents. The 25% blend of n-butyl ether shows an even greater increase of actual over predicted performance than that shown by the hexyl ether blend. The performance of the 25% ethyl ether blend is noteworthy in that it is far superior to either of its constituents and also superior to all the straight higher ethers tested at temperatures above about 3 F.

The increase in effectiveness of the ethers in blends when going from hexyl to ethyl might possibly indicate that volatility plays a major role in cold starting. In order to compare the effect of volatility of straight hydrocarbon fuels with that of the ether, a blend of 25% n-pentane was run in the engine. (Normal pentane boils at 97 F and has a cetane number of 45 as compared to a boiling point of 95 F and a cetane number of 53 for ethyl ether.) The results given in Fig. 5 show that instead of improving the performance of the reference fuel, the 25% n-pentane increased its starting temperature more than 5 F. These data support those given above showing cetane number to be inadequate as a measure of starting performance of the ethers. Hence, the superior performance of ethyl ether is not entirely due to its volatility, but apparently to its relatively low ignition temperature and wide range of inflammability.

■ Results on Ethyl Ether Blends

The unexpectedly good performance of the 25% ethyl ether blend prompted further tests to determine the optimum concentration of ether in the reference fuel from the standpoint of cold starting. The relation between starting temperature and starting time in Engine A for 10, 25, 50, and 75% ether blends is shown in Fig. 6. The 50% blend gives the best performance of the group.


STARTING TIME VS. TEMPERATURE
Engine A
10% ETHYL ETHER IN REFERENCE FUEL
7½% ETHYL ETHER IN REFERENCE FUEL
REFERENCE FUEL
5% ETHYL ETHER IN REF. FUEL
2½% ETHYL ETHER IN REFERENCE FUEL
STARTING TIME (SEC.)
TEMPERATURE (°F.)

■ Fig. 6

The effect of concentration of ether in the reference fuel is more obvious in Fig. 7, showing the relation between starting temperature and ether concentration for various starting times. These data indicate that the optimum concentration of ether in the reference fuel is in the range of 40 to 50%. Again assuming the starting temperature at 50-sec starting time to be the best measure of performance in service, a 42% optimum blend of ethyl ether in the reference fuel would permit starting at a temperature 24 F lower than obtainable on the 74 cetane-number reference fuel and 14 F lower than obtainable on the straight ethyl ether. Reference fuel No. 1, of 53 cetane number, which is representative of commercial high-speed diesel fuels, permitted starting in Engine A at 33 F. Since the 42:58 ethyl ether-reference fuel blend started at −3 F, the reduction in starting temperature possible with the blend over that normally obtainable was 36 F. The 50% ethyl ether-reference fuel blend gave a better starting performance than any of the ethers when run straight.

In tests on reference fuel, No. 2, using the electric intake air heaters recommended by the manufacturer, Engine A was started in 50 sec at −2 F, as compared to the −3 F obtained on the optimum (42%) ether blend without the use of the intake air heaters. At this point in the work it appeared that the use of both the heater and the ether blend should permit starting at temperatures much below those heretofore obtainable on the separate agents. However, tests on the combination showed the use of heaters to have a deleterious effect on performance of ether blends.

The effectiveness of the ethyl ether in blends does not appear to be limited to any particular fuel or engine type. In tests in Engine B, a 25% blend of the ether in reference fuel No. 1 permitted starting in 30 sec at 0 F, as against 25 F obtained with the straight reference fuel. This reduction in starting temperature effected by the 25% ethyl ether is quite comparable to that shown by the 25% ether in the 74 cetane-number reference fuel No. 2 in Engine A.

In addition to its effect on starting temperature, the ethyl ether appreciably lowered the cloud and pour points of fuels to which it had been added. Table 3 shows results obtained on blends of the ether in two reference fuels.

Table 3 – Cloud and Pour Points of Blends of Ether in Reference Fuels

Ether in Blend, %	Reference Fuel No. 1		Reference Fuel No. 2	
	Cloud, F	Pour, F	Cloud, F	Pour, F
0	−10	−10	0	0
10	−10	−15	−6	−6
25	−18	−20	−14	−15
50	−22	<−35	−28	−28

■ Adding Ethyl Ether to Intake Air

Concurrently with the test work on ethers at the Esso Laboratories – Research Division, similar studies were being made in England, and they have also found that ethyl ether is a very effective agent for improving cold starting. The development there, under the urgency of the pressing war conditions of 1940 and 1941, had progressed (by early 1942) further toward practical application than it had in this country. Their studies indicated that introduction directly into the air stream – together with the usual injection of the regular fuel into the combustion chamber – was very effective. It is our present understanding that most commercial diesel engines now built in England for road vehicles are equipped with a very simple applicator for admitting ether with the entering air during the starting period. The ether is contained in a sealed "sparklit" tube holding perhaps 30 cc. Just prior to starting, the tube is punctured, and the contents drain into the applicator. Approximately 5 cc is immediately introduced into the air stream; and after the engine fires, the remainder is slowly metered in to maintain speed, possibly over a 1- or 2-min period.

The Research Division data on that part of its program dealing with admission of starting agents directly into the entering air stream confirm the results on the ethyl ether as reported to us from England. Used in this manner, the ethyl ether is remarkably effective. A summary of pertinent data is given in Table 4. Neither engine can normally be started below about 24 F on good grade diesel fuel without supplementary starting aids.

Certain of the starts in Table 4 were made in an extremely short time, indicating that refinement in the starting technique might give even easier starting. Not clearly indicated by the data, but distinctly characteristic of the tests using ethyl ether, are three noteworthy observations. *First,* the ether gave partial combustion immediately and raised the cranking speed upwards of 50-70 rpm above the speed when not admitting ether. This is an additional reduction in battery and starter loads over and above any reduction occasioned by elimination of air heaters or glow-plugs. The engines were started at −15 F with initial cranking speeds as low as 50 rpm, whereas past experience has always indicated a requirement of about 100 rpm. On an overall basis, the conclusion is reached that, by the use of ether in this way, the diesel engine can be started at extremely low temperatures as readily as, if not more easily than, a gasoline engine. *Second,* the ether flow must be maintained after the initial start and until the engine has warmed up sufficiently to support combustion on the

main fuel being injected. *Third,* the amount and flow rate of the ether during starting is in no way critical. Most of these tests were conducted simply by bleeding the ether into the air stream from a burette.

It is believed that the importance and significance of these observations should be pressed home to the builders of diesel engines and to the commercial and military users of diesel vehicles. The practical problem of how best to handle the ethers on the vehicle – whether in sealed capsules as used by the British, or in containers feeding a hand-operated priming pump, or by some automatic carburetor device – remains to be solved, although all three methods have been successfully demonstrated. Similarly, there is the question of whether ether should be distributed in bulk or capsules, whether the applicator should be a standardized fixture sold as an accessory or a permanent unit designed and installed by the maker of the vehicle.

One manufacturer of crankcase scavenged engines has warned against the use of ether in the intake air of such engines because of the possible danger from explosion. Two other manufacturers subscribed to the belief that it might be dangerous to admit ether into the crankcases of such engines. Lacking cold-starting experience with this type of engine, we are not at present in a position to provide technical data either pro or con, on the possible risk of using ethyl ether as a cold-starting aid.

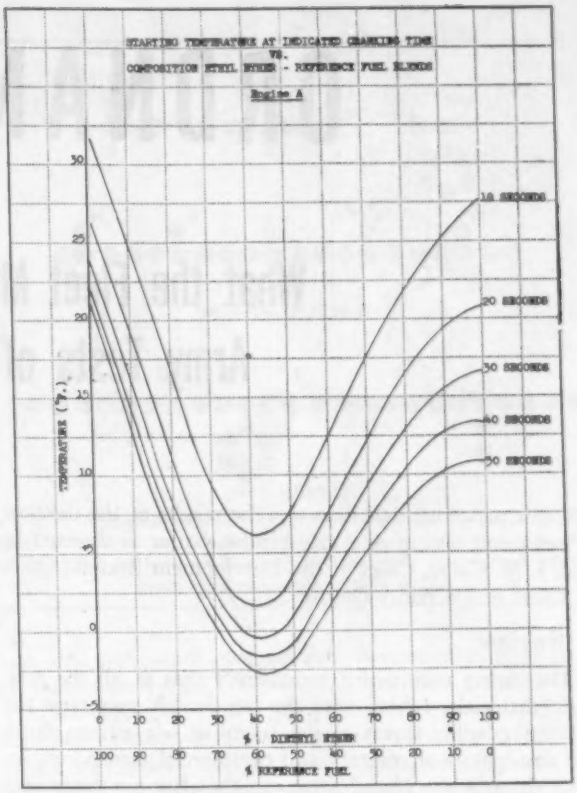
It has been our understanding that crankcase explosions with diesel engines have almost invariably occurred after the engine had been running under load or shortly after shutting down. In starting at unusually low temperature, it is unlikely that sufficient oil vapor would be present to support combustion in the crankcase. Consequently, since ether would be used only during starting and under very cold conditions, we would not anticipate any trouble.

It may be possible on such engines to provide for the introduction of the ether into the individual cylinders at the intake port so that the possibility of the ether mixing with the air in the crankcase or being scrubbed out by the lubricating oil and released later when the engine was warm, would be minimized. As a matter of fact, engines

Table 4 – Effect of Injecting Ethyl Ether into the Entering Air Stream

	Test Temperature, F	Initial Charge, cc	Rate of Feed cc per min	Initial Cranking Speed, rpm	Cranking Time to Start, sec	Total Used*, cc
Six-cyl, 4-cycle engine test, SAE-10 oil in crankcase. Variation in cranking speeds obtained by cutting out cells in battery and/or using warm batteries or by dilution of crankcase oil with kerosene	−15	0	72	90	25.5	38
	−15	5	40	90	27.0	31
	−15	5	40	63	37.0	33
	−15	10	40	34	No start – cranking speed too low	
	−15	10	40	100	23.5	28
	−15	10	40	60	2.4	27
	−15	10	40	50	33.0	37
	−15	20	40	90	1.9	31
	−15	20	36	75	23.8	35
	−15	20	35	55	26.7	38
	−15	20	30	100	21.9	34
	−15	30	38	86	19.8	44
	−30	5	40	90	31.7	38
	−40	30	252	225	37.0	..
	−40	15	168	91	74.5	..
	−40	20	162	175	22.3	..
	−40	20	144	170	30.4	96
	−40	20	144	134	34.9	..
	−40	20	144	128	25.3	120
Three-cyl, 2-cycle engine tests with other conditions as for 6-cyl, engine above	−15	0	40	252	22.5	28
	−15	0	72	225	4.6	42
	−15	0	36	298	11.2	13
	−15	5	60	186	17.0	39
	−15	20	80	200	1.4	26
	−30	0	114	230	16.0	30
	−30	0	84	220	43.0	65

* These totals include any additional ether needed to keep engine running after starting and until continuous operation on the main tank fuel was possible.



■ Fig. 7

of this type are rarely installed in locations where starting at temperatures much below freezing is a regular occurrence, for example, on shipboard. For such applications, a 50-50 blend of ether in fuel will provide a probable lowering of minimum starting temperature of 20-25 deg, and should provide a suitable method of using ethers for such engines.

■ Conclusions

The following conclusions applicable to fuel seem justified as regards cold starting of diesel engines.

1. Normal petroleum hydrocarbon fuels will permit starting the best of automotive engines at temperatures not below about 20 F without supplementary starting aids.

2. Materials blended into normal diesel fuels in optimum concentration to improve cetane number and cold starting may give a reduction in minimum starting temperature of probably not over 20-25 deg.

3. Ethyl ether introduced into the air stream of diesel engines (not including crankcase scavenge) will allow starting down to at least −40 F without additional starting aids, but provided the crankcase lubricant and cranking facilities give sufficient cranking speed.

It has been roughly estimated that a total of 95,000,000 diesel horsepower of all kinds may be in existence in the U.S.A. by the end of 1943. If the average horsepower of these engines is 75, there would then be available about 1,250,000 engines. Of these, perhaps ¼ would operate in cold climates where for 6 months a cold-starting problem might exist. Hence, if each engine used 100 cc of ethyl ether per day (two cold starts), the annual consumption of ethyl ether would be of the order of 1,500,000 gal per year. This quantity can be made available within a reasonable time, probably as fast as any demand arises.

ORDNANCE KEEPS 'E

What the Fleet Men May Learn from Army Tests of Synthetic Tires

IN this paper we will discuss some results of the development and testing of Army synthetic tires as directed by Col. J. M. Colby, Chief of the Development Branch, Office of Chief of Ordnance-Detroit.

■ Review

The Army maintained consistently that of all the strategic materials, rubber was the one which presented the greatest possible threat to the safety of our nation, from the standpoints of military and commercial transportation. That threat materialized three months after our entry into the war, when decisive enemy action cut off 90% of the world's pre-war rubber sources.

The events, following this loss of 7,000,000 plantation acres of Far Eastern rubber, are too well known to require more than a passing summary. We had on hand a fair-sized crude rubber stockpile, supplemented by better than a million tons of rubber on wheels, and a moderate but belated synthetic production program. The Baruch Committee made an estimate of the situation, as to the necessity of conserving natural rubber, the possibility of getting rubber from Latin America, and the merits of the types and processes for synthetic rubber; and laid down a broad program designed to see the nation through the threatened rubber crisis. The Office of Rubber Director and the chemical, petroleum, alcohol, and rubber industries, rapidly and accurately carried out the program and accomplished the seemingly impossible in producing nearly enough synthetic on time to meet essential needs.

■ Army-Industry Liaison

In fighting a war, there is little time to analyze the rush of events or to appraise their consequences beyond the war's end. The united objective is rightly the success of our arms. Yet under the exigent demands of a great war, there are compressed scientific and industrial developments that would have taken a decade of peace to achieve. This is true of the development of the synthetic tire.

The emergencies of a great conflict often require new or greatly enlarged agencies and personnel to deal with the emergencies. Frequently, peacetime organizations are not adequate in administration or personnel to handle war crises. Men with capabilities of making sound emergency decisions are generally too busy and too active to accept leadership and remain fixed in slower moving peacetime organizations. The rubber crisis called for capable men and fortunately Ordnance got them.

The overnight requirement of a synthetic tire called for quick action and was rightly classed as one of the major emergencies. Tire men from past practice knew of no way to convert natural rubber tires to synthetic tires but by the painstaking process of build and test. The change could be made only in progressive steps, from all rubber to varying amounts of rubber and synthetic and finally where possible to all synthetic. Past experience indicated that the larger the tire, and the more severe the service, the more difficult the problem would become. Therefore, indispensable facts must be developed, organized, and interpreted into physical products.

There was available too small a supply of synthetic for a score of companies to start separate programs. Besides, industry possessed none of the Army vehicles on which to test the military product. It was clear to both the Army and to the rubber industry that a joint program of tire research, development, and testing must be launched, which in its requirements of funds, of equipment and facilities, and of experience and know-how would go far beyond any program ever before attempted.

So the Army and industry pooled their resources and did it without stifling Army or company or individual initiative. The Army furnished the requirements of test

THE abrupt stoppage of crude rubber imports early in 1942 called for immediate crystallization of a program to produce, test, and use synthetic rubber as fast as made available for tires for military and commercial motor transportation. The Ordnance Department and the rubber industry pooled their resources of synthetic rubber, of tire know-how, and of testing facilities, to initiate a tire testing program heretofore unapproached in magnitude of effort.

Conclusions of interest to fleet men are that Ordnance and rubber industry have completed a large amount of pioneering development work on military synthetic tires, the results of which are being directly utilized in synthetic tires for commercial service; also, that synthetic tires, to do a good job, require more careful supervision

[This paper was presented at the SAE War Engineering-Annual Meeting, Detroit, Mich., Jan. 10, 1944.]

OLLING

by

Lt.-Col. B. J. Lemon and Capt. J. J. Robson

Ordnance Department

vehicles, proving grounds, operating personnel, and procured the experimental tires. Industry provided the laboratory test facilities, supplied the experts in research, development, compounding, and processing, as well as the experienced field personnel to follow the controlled proving ground tests, and the uncontrolled field tests, and built the tires, tubes, and flaps.

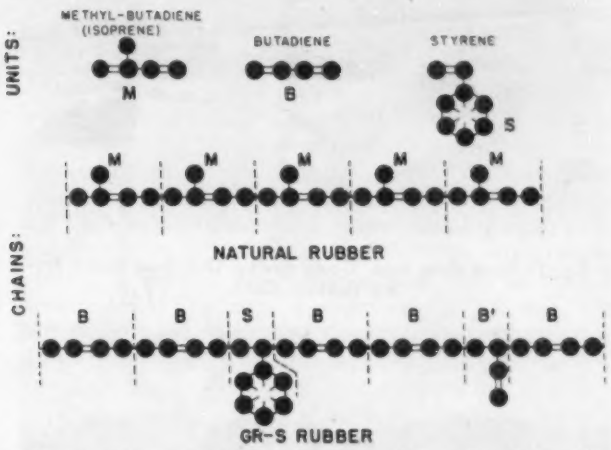
Direct working Ordnance-rubber industry liaison was set up, which functioned through committees made up of the industry's leading tire technicians and coordinated by Army and civilian personnel with long and varied rubber experience. Committee administration was furthered by the unselfish help of the Society of Automotive Engineers and the Tire and Rim Association.

Early in the Development Branch program, Major-Gen. G. M. Barnes, chief, Technical Division, Ordnance Department, addressing a joint meeting of rubber experts stated:

"There are a large number of rubber research and development problems that now need solution. They are pressing. We need the help of the rubber industry of this country. I want to be sure that industry will furnish the most capable technical representatives so that Ordnance can have advice of the highest authority on these complicated technical problems. The work that they will originate will result in the expenditure of millions of dollars of public funds.

"To get the best product for the Army, it will be necessary that companies give up data developed at high cost, so that industry as a whole may produce uniform and adequate synthetic products. This means not partial but complete pooling of information all along the line. We expect and believe that the rubber industry will meet the challenge of these difficult problems and will solve them."

The rubber companies accepted this challenge. They furnished their experts and pooled their secret information – not with a feeling of coercion or even of dignified acquiescence but with the heartiest sort of cooperation and unity of purpose.



■ Fig. 1 – Diagram showing units that make up the chain molecules of natural and GR-S (synthetic) rubbers and comparing the structural formations of the two chains made up of each unit – black dots represent carbon atoms

■ Synthetic Rubber

In spite of a tremendous amount of scientific and practical work no one, so far as known, has succeeded in producing the natural rubber hydrocarbon by synthesis. Therefore, the use of the term synthetic rubber is scientifically incorrect. However, since certain synthetic materials have many properties of natural rubber, the term synthetic rubber has come to be accepted through common usage.

In its general capacity to do work, natural rubber is superior to synthetic in most performance characteristics. On the other hand, synthetic rubbers are superior to natural rubber in resistance to most deteriorating influences, such as swelling by gasoline and mineral oils and impairment by ozone and sunlight. These special characteristics guided the pre-war limited production of synthetics into specialized fields.

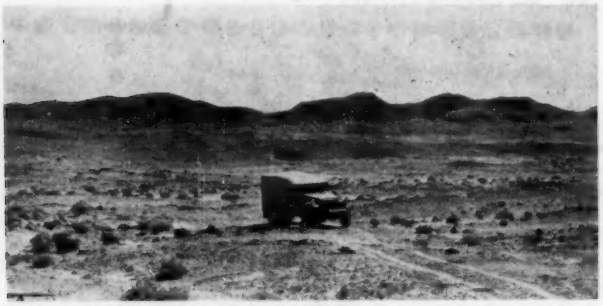
From the pneumatic tire standpoint, discussion will be confined principally to the one synthetic rubber at present in greatest production, namely, GR-S. The letters GR-S stand for "Government rubber-styrene," sometimes referred to as domestic buna S. The term buna S is the German designation for GR-S synthetic, which is made by combining butadiene and styrene, both of which must be made synthetically or derived from related materials. GR-S synthetic was chosen as our principal mass-production synthetic rubber on the basis of its general characteristics and its economical production. The raw materials commonly used to produce it are petroleum or ethyl alcohol for butadiene and coal tar or petroleum for styrene.

by fleet men respecting speeds, loads, and general maintenance attention than has been practiced in the immediate past with respect to tires made of natural rubber.

★ ★

THE AUTHORS: LT.-COL. B. J. LEMON (M '23), chief of the Rubber Section, Development Branch, U. S. Army Ordnance Department, directs the development of rubber and synthetic tires, tank tracks and bogies, recapping and repairing and the attendant operation of testing these products for Ordnance. A captain in the Motor Transport Corps in World War I, Col. Lemon gained his industrial experience with the U. S. Rubber Co., which he joined in 1915. He is a graduate from Cornell University. CAPT. J. J. ROBSON, chief of the Wheeled-Vehicle Unit, Rubber Section, Development Branch, U. S. Army Ordnance Department, is responsible for the development of rubber and synthetic pneumatic tires for Ordnance. A graduate from M.I.T., he had been in the development, testing and field service divisions of Firestone Tire & Rubber Co. before being called into active service.

■ Fig. 2 – Sand dune area, Camp Seeley, Ordnance Desert Proving Ground, Calif.



■ Fig. 3 – Cross-country test course, Ordnance Desert Proving Ground, Calif.

The differences in the fundamental properties of natural and GR-S appear to be due to variations in molecular structures; the arrangements of the atoms in the molecules controlling such valuable working properties as elasticity and strength.

If we represent in Fig. 1 the chemical structures of synthetic and natural rubbers as link chains, GR-S appears to have a less-uniform molecular chain structure than does natural rubber, with the styrene branching off irregularly along the synthetic chain. In natural rubber, the links are alike; in GR-S, branching by styrene occurs. The long-chain molecules of GR-S, with branching side groups, being less flexible than rubber, are prone to break at lower elongations.

Some of the deficiencies of a GR-S tire compared with one of natural rubber are:

Develops higher flexing heat.

Loses stretchability when hot, causing greater cracking and chipping.

Has low tear resistance at higher temperatures.

Sticks to itself poorly, before vulcanization.

Requires more time and power to mill.

Much has been done to improve the undesirable characteristics of GR-S by the use of reinforcing substances. For example, in tires GR-S does not have good physical properties unless mixed with carbon black. Progress made by Ordnance and industry in overcoming shortcomings of GR-S in tires will be discussed under test results farther along in this paper.

■ Purpose and Authority for Army Tests

The Army tire development and testing program was initiated to save natural rubber and to convert to synthetic rubber as fast as synthetic became available, maintaining adequate military requirements in both the natural and synthetic rubber products.

On January 6, 1942, Under Secretary of War Robert P. Patterson issued the following directive to Major-Gen. L. H. Campbell, Jr., the Chief of Ordnance:

"The outbreak of hostilities in the Pacific intensified the problem of synthetic rubber production. Accordingly, your branch of the service is authorized to purchase and test such semisynthetic tires and tank tracks as it deems advisable."

■ Scope of Testing

A program was drawn up authorizing the four major rubber companies to produce several hundred thousand synthetic tires, to be built with from 80 to 100% synthetic

■ Fig. 5 – Mud course, Camp Seeley, Ordnance Desert Proving Ground, Calif.

■ Fig. 4 – Rock course, Camp Seeley, Ordnance Desert Proving Ground, Calif.

in the tread and sidewall, but with natural rubber carcass, for unrestricted service in Army field organizations in the United States.

A comprehensive controlled laboratory test program was prepared covering rubber industry and Bureau of Standards tests, Army controlled highway and cross-country tests, and uncontrolled Army field service tests, in order to get conclusive factual data and experience as to synthetic tire performance and durability.

▪ Fig. 6 – Cross-country area used by Ordnance Tire Test Fleet, Normoyle, Tex.

▪ Fig. 7 – Cross-country trail, Ordnance Tire Test Fleet, Normoyle, Tex.

In March, 1942, the Army Desert Test Command was established at Camp Seeley, El Centro, Southern California, for the purpose of evaluating the performance of tires under high atmosphere temperature conditions, as well as to study tire flotation and traction in sand. Figs. 2, 3, 4, and 5 show types of terrain around Camp Seeley. Lt.-Col. J. E. Engler, who established the Desert Proving Ground and has continued in command, has shown unusual ability, resourcefulness, and efficiency in carrying out the testing operations. The operating personnel is generally military except for resident technical advisers furnished by the various rubber companies. Tests at this California Desert Proving Ground magnify tire failures caused by heat separation.

The Ordnance Normoyle Tire Test Fleet, based at San Antonio, Tex., was established to satisfy controlled tire and tube test requirements for trucks designed for military service, covering hard surface and gravel roads and cross-country operation. Figs. 6, 7, 8, and 8A illustrate the nature of the cross-country course in Texas. The 165-mile test route is made up of 70% hard-surface highway, 15% gravel, and 15% cross-country trails. The Army trucks operate 24 hours a day, 7 days a week, with an average weekly vehicle mileage of approximately 270,000. The test facilities are open to all tire and tube companies and all such companies who submit products for test may send company observers to follow their products and to receive reports. The Normoyle test is managed efficiently under Development Branch, Ordnance Department Contract by S. R. Filer of The Firestone Tire and Rubber Co., with an operating personnel of about 750. Tests at this Texas Proving Ground provide operation under conditions of high average year around temperatures.

Canada, with a parallel problem of using synthetics, is running a sizable fleet of its own special military trucks and tire sizes as an integral part of the Normoyle Tire Test Fleet.

During the past summer, another Ordnance controlled tire test was operated out of Aberdeen Proving Ground to supplement the test results of the California Desert and the Texas courses under the more moderate temperature conditions of Maryland.

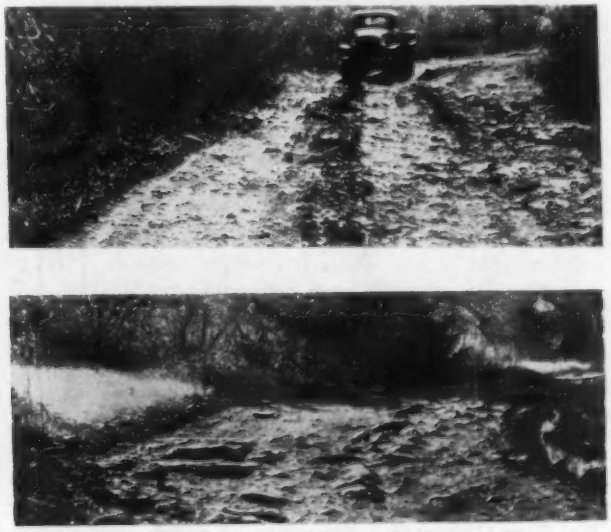
Last winter, Ordnance ran controlled winter tests at Shilo, Manitoba, Can., to determine the effect of extreme low temperatures on natural and synthetic rubber products, where temperatures ranged as low as —45 F. Similar tests are running elsewhere this winter.

Special controlled tire tests are intermittently run by the various Army Service Boards in different parts of the United States.

Finally, uncontrolled field service tire tests, by the troops of using Arms and Services provide a continuing check on rubber and synthetic tire and tube performance.

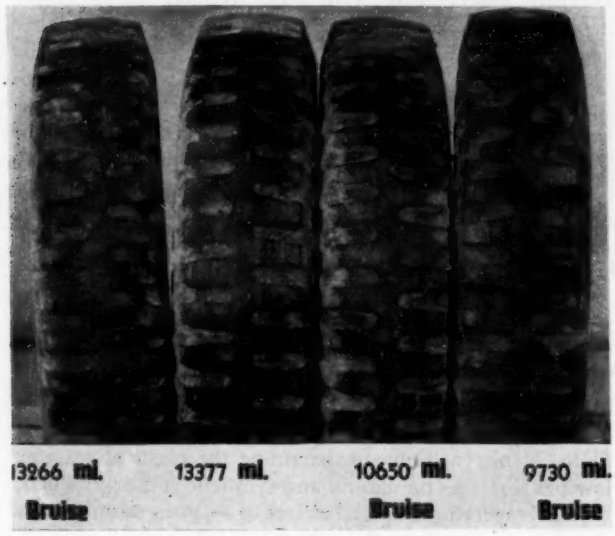
These tire testing operations are believed to be the most

▪ Fig. 8 – Embedded rock, cross-country course, Ordnance Tire Test Fleet, Normoyle, Tex.



▪ Fig. 8A – Cross-country course, Normoyle Test Fleet, Tex.

■ Fig. 9 – 6.00-16 100% synthetic tires tested at Camp Seeley at 50 mph on pavement – course: 10% gravel for 10,000 miles, 40% gravel for remainder – original 'S-3' tires



■ Fig. 10 – 6.00-16 100% synthetic tires tested at Normoyle – course: 70% highway, 15% gravel, and 15% cross country

extensive of their kind ever attempted anywhere – either by a military or an industrial group.

■ Army versus Commercial Tires

From the start of the testing program, a necessary distinction was drawn between the service required of Army tires, which must be capable of cross-country maneuver, and the highway service of tires used in commercial transport.

Pneumatic truck tires, in commercial transportation service, generally roll over hard-surfaced, well-maintained highways to unbelievable mileages, with obsolescence depending largely on tread life. The same size tires on military vehicles in combat area service or even on poorly maintained roads in communication zones, adjacent to combat, have, at best, minimum service expectancy, and go out of service generally from carcass failures.

At the beginning of the Ordnance tire testing program, many experienced commercial tire testing engineers, unfamiliar with Army tire requirements hesitated in approving as practicable the rugged, cross-country type of maneuver area, selected by the Army for tire tests. However, concurrent experiences in large-scale domestic Army training maneuvers, as well as subsequent return of tires from war theatres proved that the inclusion in the Normoyle Tire Test Course of a percentage of rough cross-country routes was absolutely essential to the development of an adequate military tire, as well as an adequate military vehicle.

Army tires have what we call a mud-and-snow tread pattern. This is an aggressive tread, designed to give good traction in soft going, with reasonable wear on pavement. This Army tread differs from that of road transportation tires, which are known as highway type, and are primarily designed for maximum wear with traction secondary.

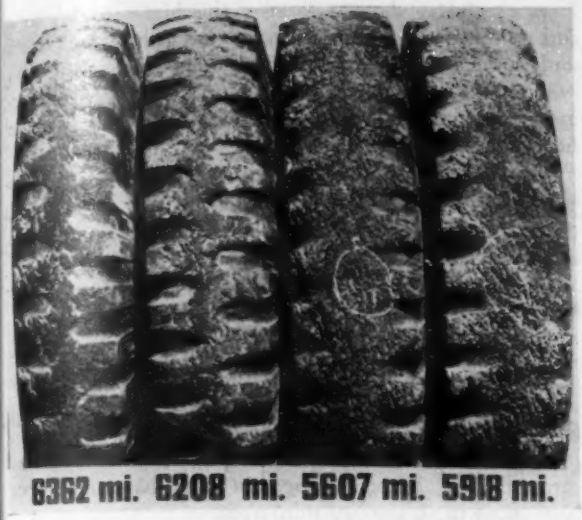
Another important difference between Army and commercial tire requirements is reflected in the design of Army trucks where dual rear tires on multirear axle vehicles are purposely oversized or underloaded, first to provide greater flotation, second to furnish a safety factor from tire bruising during cross-country operation and third to take care of a certain amount of uncontrollable overloading in combat areas. On the other hand, the front tires of Army all-wheel drive vehicles using dual rear tires, carry capacity loads – again differing somewhat from commercial practice. Also, if operating in convoy, in blackout or cross-country, Army trucks move at slower speeds than highway commercial transportation. These variations in Army and commercial vehicle design and tire requirements are recognized and considered in the conclusions drawn as to what the fleet men may learn from Army tests of synthetic tires.

In the nick of time, when the Army synthetic tire pressed for solution, an officer was ready with the right qualifications to tackle the job. He is Capt. J. J. Robson, who has carried out in a superior manner the development and testing program of the Army synthetic pneumatic tire. The remainder of this paper will cover the detailed results of the work that has been done.

Ordnance tires are roughly divided into three groups:

Group	Sizes	Vehicles
Small tires	6.00 through 7.00 (8-ply)	Passenger and light trucks
Medium tires	7.00 through 10.00 (10-ply)	¾-ton through 6-ton trucks
Large tires	11.00 through 14.00; also large earth-mover	Large trucks, 10-ton, and so on, Tank transporters, and so on

■ Fig. 11 – 9.00-20 synthetic tread tires tested at 35 mph under 4200-lb load – course: 10% gravel



■ Fig. 12 – 9.00-20 synthetic trend tires tested at Normoyle under 2350-lb load – course: 70% highway, 15% gravel, and 15% cross country

Synthetic rubber has presented increasingly difficult problems in each of these groups, and separate consideration is given to each group in order to avoid confusion.

■ Small Tires

The small tire group in the Army is largely devoted to the 6.00-16 tires for the ¼-ton 4x4 truck, known as the jeep. The first tests we ran were synthetic treads on natural rubber carcasses, which showed such promising results that we proceeded to conduct tests on 100% synthetic tread and carcass construction.

Shown in Fig. 9 are the results of a test of the first of these tires at Camp Seeley, operating at 50 mph sustained speed, at atmospheric temperatures up to 115 F. These were run 10,000 miles on our standard course, then shifted to a 40% gravel course. It will be noted that the tires show a few failures, after 13,000 to 15,000 miles, which is a very good record for such severe conditions.

Also shown in Fig. 10 are the same 100% synthetic tires tested at Normoyle, operating on the highway at 55 mph, and on severe cross-country at reduced speeds. Here again, they look very good.

These tests, together with numerous rechecks, proved conclusively that 100% synthetic tires are successful in smaller tire sizes. We later established a qualifying test at Camp Seeley, which required that the tires run 5000 miles without heat failure. Twenty-four companies supplying this size have passed this test, with most groups running 10,000 miles without failure.

Since February of 1943, when synthetic rubber started to come into limited production, small tires have been 100% synthetic. They are doing an outstanding job, as is certified to by the following comment from D. G. Roos of the Willys-Overland Motors, Inc., on a test his company ran with 100% synthetic, 6.00-16 tires: "The wear on the synthetic tires is no greater than the wear on tires made of crude rubber. The synthetic tires seem in every way equivalent to the crude rubber tires." We feel these tests of small-size tires should remove any serious questions about synthetic from the mind of fleet owners who are operating large groups of passenger vehicles or light trucks.

It has been rumored that the synthetic passenger-car size tire cannot be driven over 35 mph, or that it will not give adequate mileage. Our factual data, derived under test conditions more severe than any civilians should be driving today, prove that this is not true.

To show how Ordnance feels about these 100% synthetic tires in the smaller sizes, we've bought over a million and so far we have not had a complaint from the using troops, who are our customers.

■ Medium Tires

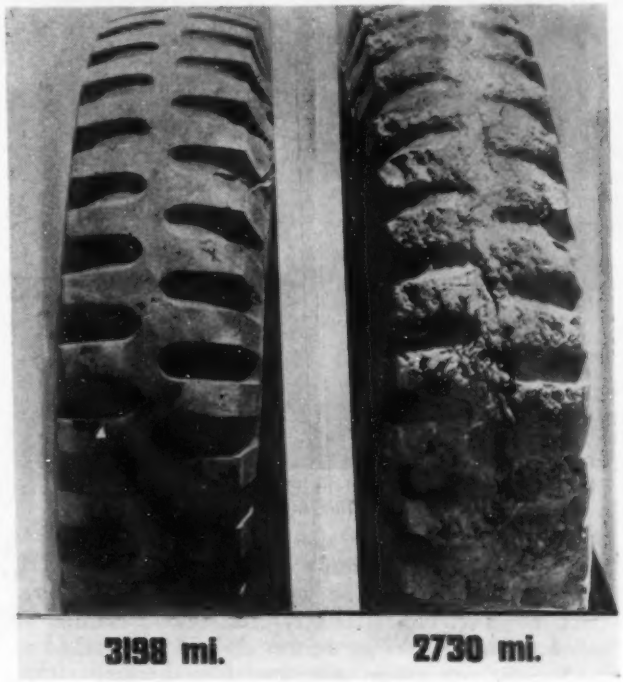
This is by far the most important group, since it consumes approximately 70% of all the rubber in the Army's pneumatic tires. It is probably also the most important to fleet operators, since the tire sizes involved – 7.00 – 10-ply through 10.00 – are those used on the great highway fleets, as well as most of the trucks used in city or short-haul work.

We ran our first tests in May, 1942, using synthetic treads, and received quite a shock. The reason is indicated in Fig. 11. These tires, tested at Camp Seeley at 35 mph, sustained speed, 20% overload per tire, with 10% gravel, resulted in a severe cracking condition.

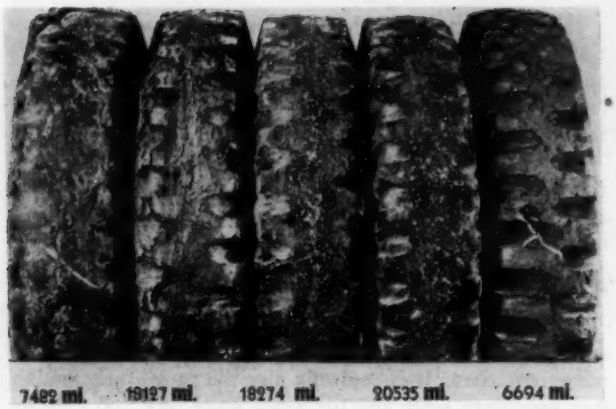
The same tires at Normoyle, on cross-country service, developed severe chipping and cracking, at very early mileage, as is demonstrated by Fig. 12. This condition is shown very clearly by Fig. 13, comparing one of the first synthetic tires with a natural rubber tire of the same make, at 3000 miles. Note that the synthetic tire is much more chipped and cracked than the one made of natural rubber.

These tires demonstrated one of the principal weaknesses of synthetic rubber – lower resistance of the tread to cutting, chipping, and cracking, both on the highway and cross country. Synthetic rubber has poor tearing resistance, which means that once a slight cut or crack has started, it progresses rapidly.

This phase of the problem, the resistance to chipping



■ Fig. 13 – 7.50-20 tires tested at Normoyle under 2700-lb load – course: 70% highway, 15% gravel, and 15% cross country



■ Fig. 14 – 7.50-20 90% synthetic tires tested at Normoyle – course: 70% highway, 15% gravel, and 15% cross country

Fig. 15 – Section of Alaska Highway

■ Fig. 16 – Section of Alaska Highway

and cracking, has been greatly improved by intensive work by industry chemists. You will agree that the improvement shown by Fig. 14 is very great since these tires at 10,000 to 18,000 miles are neither chipped nor cracked as badly as the previous group at 3000 to 6000 miles.

We are now running controlled tests over the Alaska Highway to determine the results on all-gravel operation. Figs. 15 and 16 show sections of this highway. We have just examined a number of tires running there, and apparently even under these severe conditions the synthetic tires will perform successfully. This will be of interest to operators who operate a lot on gravel roads, particularly in the northcentral and northwest states.

And just to show that others using synthetic have experienced similar trouble, Fig. 17, a captured German tire, displays the similar conditions of chipping and cutting to a very advanced degree.

■ Synthetic Rubber in Tire Carcass

The first step in this work, as already described, was the development of synthetic tread tires, with natural rubber carcasses. These used about 40% synthetic and about 60% natural crude rubber. Because of the critical situation in the crude rubber supply, it was apparent that synthetic carcasses must also be developed in order that crude rubber be reduced to an absolute minimum.

The first of these synthetic carcass tires we produced had overall synthetic contents of from 90 to 100%. Again we experienced a shock.

The first group of 7.50-20 size tires run at Camp Seeley is shown in Fig. 18. Very low mileage was experienced and practically all tires blew out from heat failures and separation of treads from carcasses at a few thousand miles. At our Normoyle test course we ran into this same heat and separation problem, as indicated in Fig. 19. These results manifested the higher running temperature of synthetic rubber, which causes excessive heat build-up and blow-out. Laboratory test data compiled by one of the tire companies show this situation graphically in Fig. 20.

At Normoyle, we simultaneously ran into the problem of reduced ability of synthetic carcass tires to withstand the bruise or shock breaks which are encountered in military

■ Fig. 17 – German synthetic tire

cross-country operations. We found that in some cases twice as many failures occurred in the synthetic carcass tires due to bruises when tires struck rocks or other obstructions. Such failures are the principal cause of premature tire removal in Army service and therefore immediately constituted a major problem. Such a typical bruise failure is shown in Fig. 21.

This established two new problems to be overcome in the use of synthetic rubber:

a. Heat-type failures, which are experienced in long-haul supply-type operations.

b. Bruise or shock breaks in the carcass due to striking rocks or other obstructions in cross-country tactical-type operations.

An extensive series of development programs have been directed at heat failures and very effective results have been obtained. It was found that placing the slight amount of allowable natural rubber in the breaker region and last plies, which is the part developing the highest temperature, helps to resist heat. The industry chemists have made important contributions by working out new compounds which run at lower temperatures and resist heat failures.

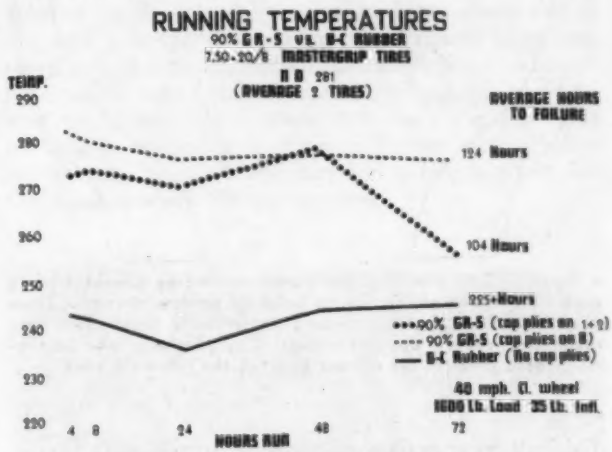
Special measures have been taken to resist bruise blowouts. The first of these is the use of extra reinforcing plies extending across the crown of the tire, known as "cap plies." The location and construction of these cap plies are indicated in Fig. 22. The cap plies, which any company can adopt, are essentially two extra breakers across the crown to improve resistance to shock- and bruise-type failures. The construction is used in the great-volume 7.50-20 and 9.00-16 tires, with cotton cord, as well as in the 7.00 10-ply and 7.50-16 sizes. In the 8.25, 9.00, and 10.00 size truck tires using 10 plies or more, we are now specifying rayon cord where synthetic rubber is used in the carcasses. This results in a thinner carcass, which reduces running temperatures and resists heat failures, and at the same time provides a stronger cord which resists bruise or shock breaks to a higher degree.

These constructional features and better compounds have given us sufficient improvement that synthetic tread and carcass tires will provide mileage closely comparable to natural rubber tires in normal military service.

A typical group of late 90% synthetic tires compared with natural rubber tires of the same make is pictured in Fig. 23. The synthetic tire shows slightly more chipping, but only one tire has failed from bruise whereas two of the

■ Fig. 18 – 7.50-20 90% synthetic tires tested at Camp Seeley for approximately 2000 miles under 2700-lb load – course: 10% gravel

■ Fig. 19 – 9.00-20 100% synthetic tires tested at Normoyle under 3650-lb load – course: 70% highway, 15% gravel, and 15% cross country

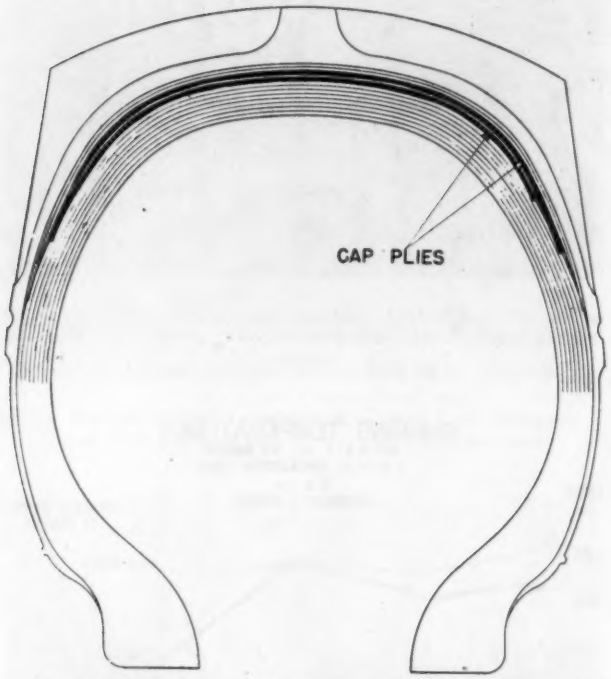


■ Fig. 20 – These typical laboratory results selected at random show higher running temperatures of synthetic tires as compared with natural rubber tires

■ Fig. 21 – Bruise break of a 7.50-20 synthetic tire tested at Normoyle

natural rubber tires are out of service. None have heat failures. This is a severe test and those who know our Normoyle course will agree that synthetic tires which will give this kind of service are going to stand up.

All our medium-size tires are now being produced with synthetic treads and carcasses. Ninety per cent overall synthetic is used in the large-volume 7.00, 10-ply, through 9.00 sizes and 70% overall synthetic in the 10.00 size.



■ Fig. 22 – Cap plies are additional reinforcing plies extending across the crown of the tire to build up carcass strength. These plies have been found necessary in synthetic tires where very rough cross-country operation exists. Cap plies may also be situated lower in the carcass between the full-width plies

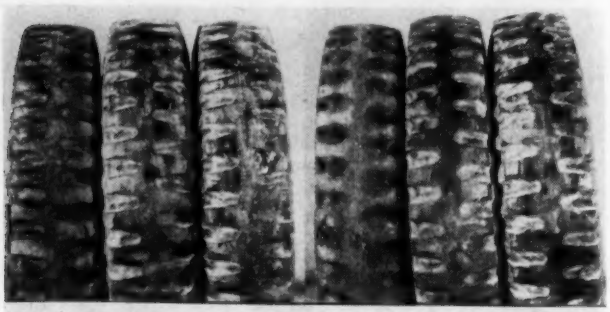
This will mean millions of synthetic Army tires in 1944.

Tread wear has been measured in all these tests but has not been a major factor because of the more pressing problem of improving resistance to premature failure. Our figures show that synthetic treads have as good wear as the war-quality natural and reclaim rubber treads, and are in many phases superior for wear.

Before leaving this group of tires let me explain that 90% synthetic is the highest we plan to use at least in the near future. A recent test is shown in Fig. 24 on some 100% synthetic tires, all of which failed at 1037 miles. These results indicate that the 10% or more of crude rubber is necessary to give adequate service in truck tires.

■ Large Tires

The largest tires, sizes 11.00 through 14.00, represent the most severe problem, since they are thicker and heavier



■ Fig. 23 – Comparison of natural rubber tires (left) with 90% synthetic rubber tires (right) – 7.50-20 tires tested at Normoyle

and are considerably overloaded on some Army and Transportation Corps vehicles. Test programs are in progress which show favorable results, but which are not sufficiently conclusive to indicate final constructions the Army will use. This includes tests of large earthmover tires, up to 21.00-24 size, used by the Army Corps of Engineers.

This large tire size group is not of general interest to fleet owners, since few road vehicles use such sizes, and their volume in Army requirements is small compared to the previous groups. Accordingly no detailed discussion will be made of this group.

■ Tubes

At first the work on synthetic tubes was discouraging. Early samples split along the tire bead line, separated at the splice, pulled away at the valve patch and chafed when used in drop-center rims or with beadlocks.

A separate committee of tube technicians was set up and special test trucks were provided so that tube failures could be segregated from casing failures. Tubes of GR-S synthetic were worked out which when applied to rims with flaps resulted in an average mileage of 17,000 and have been run to 30,000 miles without failure.

Tubes made of GR-I, butyl synthetic rubber were found to present no such major problems as GR-S and averages of better than 20,000 miles were obtained with butyl tubes.

At present, the Army is accepting military tubes of GR-S synthetic in sizes 6.00 through 9.00 for all wheel diameters when applied on rims using flaps.

Although the synthetic rubber tubes are not as good as the natural rubber tubes, it is evident that with reasonable care on application and keeping inflation up to standard, also keeping within the specified speed limit, they will give satisfactory service.

■ Repair and Recapping

A problem concurrent with the production of an adequate military synthetic tire and tube is that of recapping and repairing. As a direct result of the Ordnance-industry development program, all camelback used to recap the bulk of the Army and civilian tires is 100% synthetic and doing a good job as indicated in Fig. 25, which pictures results after 17,000 miles of operation.

Repairing with synthetic is a tougher problem than recapping, and has reached a stage where repair materials now use a blend which includes 25% synthetic rubber. Fig. 26 shows rubber repairs of two synthetic tires that have run 14,000 miles.

Repairing of synthetic tubes has presented a tricky problem. Ordinary hot or cold patches which are standard in the field have not been fully effective on synthetics to date. To repair a synthetic tube properly, it is recommended that a professional repair on a hot plate be made. The heat should be confined to within the temperature range of 285 to 310 F. When a repaired synthetic tube is subjected to higher heat the repair quickly begins to crack at the joint and continues to crack until there is a leak at the edge of the repair. On the low-temperature side, adhesion is not satisfactory below 285 F.

Synthetic tubes are more sluggish and less elastic than those made from natural rubber; therefore, in repairing synthetic tubes with natural rubber, or with a blend of rubber and synthetic, the vulcanized repair gum should have as near as possible the same modulus of elasticity as the tube itself. For example, ordinary rubber tube repair

■ Fig. 24 – 7.50-20 100% synthetic tires tested at Camp Seeley for 1037 miles

gum is not as good as tread repair gum, because the latter is stiffer and has more nearly the same elasticity characteristics as a synthetic tube. Rapid improvement has been made in compounding and repairing of synthetic tubes so that the service received from repaired tubes is fast becoming satisfactory.

■ Conclusions

From these data and discussion, you will see that a great amount of development has been done on synthetic rubber. The question is: What can fleet operators learn from this work?

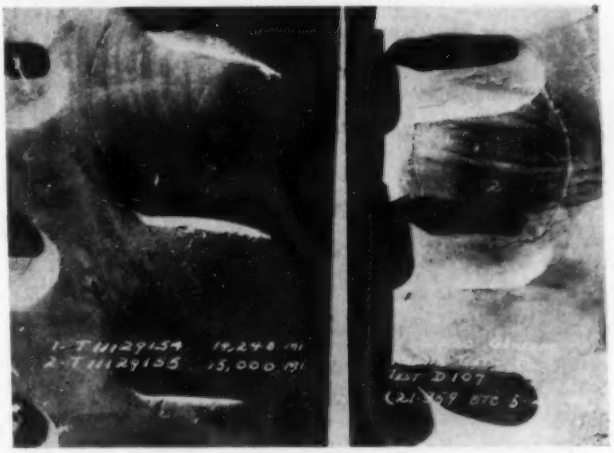
The first thing is that certain fundamental problems which originally appeared extremely difficult have now been solved through intensive Ordnance-industry development. If some of your first tires do not live up to expectations, do not jump to the conclusion that synthetic tires are fundamentally bad. If Ordnance had quit because its first synthetic tires were unsatisfactory neither you nor the Army would have sufficient tires today because our precious crude rubber would have been further expended and synthetic tires would not be in production.

The second thing is that synthetic tires to do a good job must be properly used. For most types of trucking they will give adequate service. There is one class of service where they may give trouble – the long, sustained high-speed intercity runs during hot summer months. Under these conditions synthetic tires will not stand abuse. The effect of overload on synthetic tires is of major importance. As an example, tires on a recent test averaged 4954 miles at 20% overload. The same tires at 10% underload were all still running at 10,000 miles. These were run on long hot runs similar to the most severe trucking operations. This tendency becomes more critical at the higher overload. You cannot abuse them by severe overloading, and this will be forcefully driven home to some operators by epidemics of heat blow-outs. The remedy is to get sufficient tire capacity on the trucks to carry the load, and to give them the best of maintenance.

The third thing we feel you can learn, perhaps the most important, is that the Army has every confidence in synthetic tires. This confidence is based on factual data from severe tests, plus the experience of hundreds of thousands of tires in service extended well over a year. Understand clearly that we do not claim synthetic tires are equal to natural rubber tires, because they definitely are not as good. The gap between the two is steadily decreasing, however, and the tire industry is making a tremendous effort to eliminate this difference completely. We hope, however, the fact that the Army is so thoroughly sold on synthetic tires will give fleet operators and truck engineers confidence and help to dispel some of the doubts that have been expressed.

The Ordnance Department wishes to take this opportunity to express appreciation again to the entire rubber industry for the splendid cooperation it has manifested in this vital activity. Competitive considerations have been completely abandoned, and all companies, large and small have pooled their technical resources to put the Army on synthetic rubber. We believe that when the history of this war is written, no industry will show more effective results than the rubber industry has accomplished in this Ordnance synthetic tire development.

■ Fig. 25 – Tires recapped with buna S synthetic camelback after 16,974 miles on Normoyle Test Course



■ Fig. 26 – Rubber repairs of synthetic tires after more than 14,000 miles on the Normoyle Ordnance Tire Test Course

DIESEL-ENGINE DESIGN in War Experience

by EDWARD C. MAGDEBURGER

Head Engineer, Bureau of Ships, Navy Department

WHATEVER else may be said of war, it is a highly potent incentive forcing warring nations to mobilize every available resource. It has made us feel that we are a nation and that "united we stand and divided we fall."

To Americans, the U. S. Navy is, without a doubt, the principal embodiment of their proud spirit of independence and the main reason for their quiet confidence in the ultimate victory. It represents the highest of our inventive genius and the cream of our manhood.

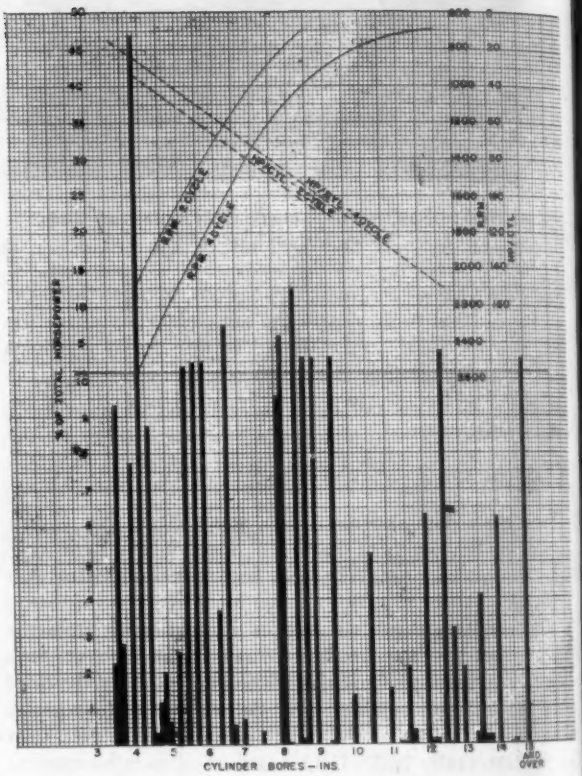
Not being an aggressive nation, we were subjected to a "Pearl Harbor," but aroused thereby we will secure respect and establish decency in international relations. To do this, we are now ready to carry war to the enemy. Enormous concentrations of men and material have been and are being transported in safety to the theaters of war and a multitude of types of landing craft are ready to challenge the enemy's armed forces, which overran all Western Europe and Pacific islands up to Australia.

Such an enterprise would have been utterly impossible, if it were not for the diesel engine and the existence of automotive mass-production methods.

The purpose of this paper is to take advantage of the unusual circumstances created by the war, whereby practically the whole diesel-engine industry of this country contributed not only great numbers of the standard types of engines but also developed many new and improved designs to meet the Navy's war needs. It is, therefore, intended to present the design characteristics of the diesel engines used in the Navy and to point out some of the probable design trends to help the industry chart its course under new peace conditions when the normal process of survival of the fittest will resume its functioning undisturbed.

The principal characteristics of the engines for the Navy are lightness of weight, compactness, and reliability of performance, even when handicapped by being "a long way from home." The justification for these requirements can be visualized easily in the light of everyday war news; and these are the requirements which will help American engines to survive the competition of the post-war world.

Exact figures on number of engines or their total output are not essential for our discussion. Such figures, however, were used in evolving the picture it is intended to present. Fig. 1 gives the percentage of the total horsepower creditable to every cylinder size. Since most of the sizes had less than 1% of the total, two different scales of ordinates were used. The upper curves in this figure give the rpm



■ Fig. 1 – Percentage of total horsepower creditable to every cylinder size

and horsepower output of an individual cylinder, or rather piston, for both 4-cycle and 2-cycle engines at full power rating, based on 20 of the ranking engines, the data for which are assembled in Table 1. It is obvious that these curves represent actual data for engines in greatest use in the Navy and not any one's individual opinion as to what the present-day diesel engine is or should be.

The most striking observation from the study of Fig. 1 is the fact that 47% of all diesel engines in the Navy are of the 4¼-in. bore, which develops 33 1/3 hp per piston at 2100 rpm, when operating on a 2-stroke cycle, and 25 hp at 2600 rpm when it is of the 4-cycle type. I am sure that it is not necessary for me to call attention to the overwhelming preponderance of 2-cycle engines and of high-speed engines of small bores in Table 1, or to the fact that the ranking 4-cycle engines are of the supercharged type, leaving only the engines for standard ships' boats in the naturally aspirated class and that only because the desira-

[This paper was presented at the SAE War Engineering Annual Meeting, Detroit, Mich., Jan. 13, 1944.]

Trends from ... of the U. S. NAVY

DESIGN characteristics of 20 diesel engines now used by the Navy are discussed by Mr. Magdeburger, who also points out some of the probable design trends to help industry chart its course when the peacetime conditions of normal competition return.

A series of diagrams drawn to the same scale are given to illustrate the various arrangements of cylinders used in diesel engines up to the present time. Goal of all cylinder arrangements is to help provide those characteristics most needed in Navy diesels: lightness, compactness, and reliability of performance.

The author also discusses the successful use of gears at high engine speeds, the freezing of fuel-injection-equipment design, lack of combustion research, and some of the problems that have developed because of the scarcity of many strategic materials. He suggested further that there be more adequate appreciation of the ballistic effect of exhaust gases as an aid in inducing supercharging.

■

THE AUTHOR: EDWARD C. MAGDEBURGER is a cosmopolite among engineers, having worked on gas engines in Germany, shipbuilding in England, diesel engines in Russia, and various engineering jobs in the United States. Now head engineer for the Bureau of Ships, U. S. Navy, Mr. Magdeburger has been employed with that office since 1922, where he started as an aide on diesel engines. A native of Russia, he came to this country in 1910, and spent his first two years here at Allis-Chalmers and Fairbanks, Morse & Co. From 1912-1920 he designed diesel engines for Busch-Sulzer in St. Louis.

bility of "freezing" the design of these engines as originally adopted was far greater than any advantages to be derived from modernizing them.

■ Craftsmanship Versus Mass Production

With the development of mass-production methods and the adaptation of the American automobile to the needs of the masses, the number of automobile owners in this country grew into millions. Compare this with the hand-made exclusiveness and perfection of the European automobile. In a similar manner, the diesel engine never reached its maximum usefulness until adopted by the automotive industry under pressure of war and produced on assembly lines with greatly increased speed and with consequent reduction in price.

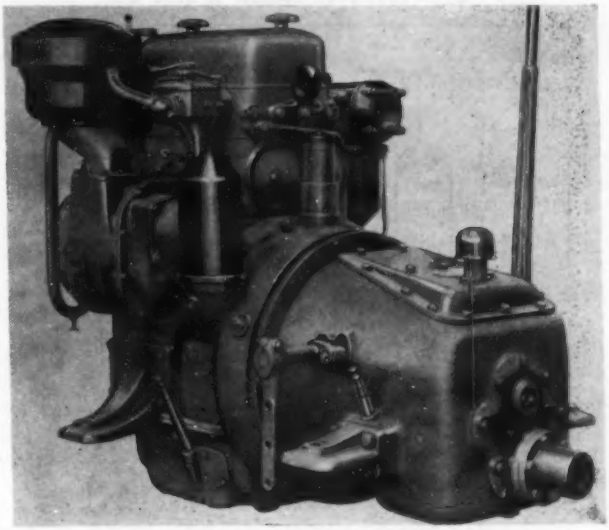
This process of adapting the diesel engine to mass production robbed it of its aura of craftsmanship, which surrounded the process of assembling and testing a diesel engine only a few years ago. But it gained immensely in adaptability. This is a revolutionary process of great importance to the entire industry and will require considerable attention on the part of the heretofore recognized leaders of the diesel industry, because now that the wall of prejudice is broken, the future development will most likely follow an entirely different course. This new adaptability of the diesel engine is best illustrated by the ranking engine of Table 1, and is manifestly the reason for its magnificent usefulness in this war. This cylinder size is used in the Navy in engines from 15 to 800 hp and in units of from 1 to 24 cylinders. Figs. 2, 3 and 4 illustrate the single-cylinder, 6-cyl, and 24-cyl engine units of this size. The second engine of Table 1 has likewise a notable range of power – from a 3-cyl engine driving a 100-kw generator unit to a 32-cyl engine of over 2000 hp output.

Thus the advantages of mass production forced the

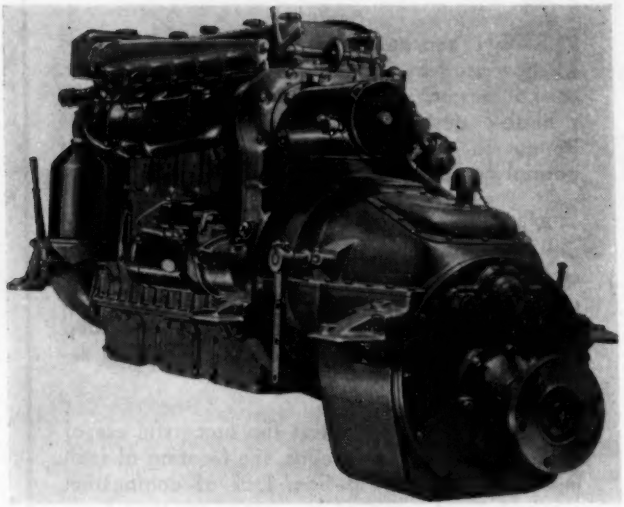
Table 1 – Twenty Diesel Engines Arranged by Percentage of Total Horsepower in U. S. Navy

No.	Bore and Stroke	Rpm	Cycle	Bhp per Piston	Total Bhp, %
1	4¼ x 5	2100	2	33.3	38.8
2	6½ x 7	1270	2	62.5	7.02
3	8½ x 10	744	2	75.0	6.33
4	8½ x 10	800	2	85.0	5.55
5	4¼ x 5	2100	2	33.3	4.59
6	8½ x 10½	750	2	100	2.68
7	9½ x 12	900	4s*	112.5	2.57
8	6 x 6½	1620	2	60.0	2.48
9	8½ x 10½	750	2	85.0	2.36
10	8¾ x 10½	750	2	106.3	2.33
11	5¾ x 6	2000	4s	41.7	2.31
12	4¼ x 4¾	2600	4s	25.0	1.71
13	9 1/16 x 13⅜	700	2	100.0*	1.64
14	9 1/16 x 13⅜	700	2	85.4	1.16
15	5½ x 7	1200	4	19.75	1.13
16	3½ x 4½	1425	4	6.25	0.87
17	4¼ x 5	1200	2	15.0	0.81
18	4¼ x 5½	1700	4	17.5	0.80
19	4 x 5½	1885	4	10.0	0.75
20	12½ x 13	700	4s	150.0	0.71
	All Others	2 and 4			13.4

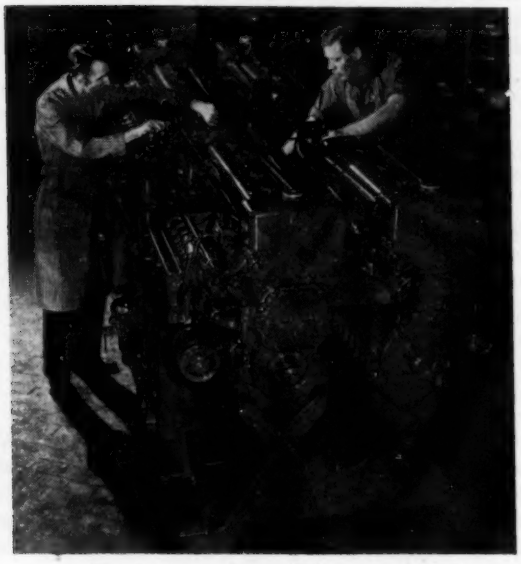
*4s = Supercharged.
*Separately driven blowar.



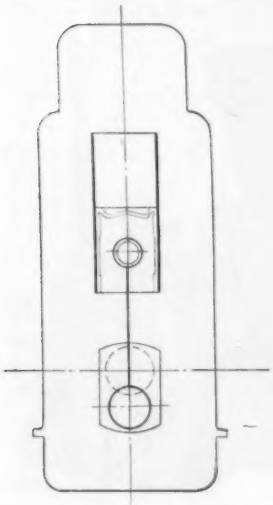
■ Fig. 2 – Single-cylinder engine



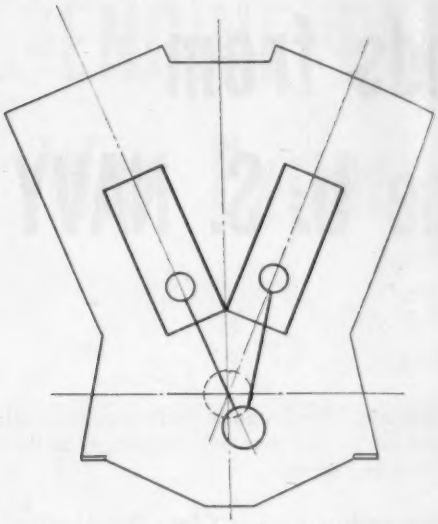
■ Fig. 3 – Six-cylinder engine



■ Fig. 4 – Twenty-four-cylinder or quad unit



■ Fig. 5 – Basic 2-cycle cylinder with uniflow scavenging



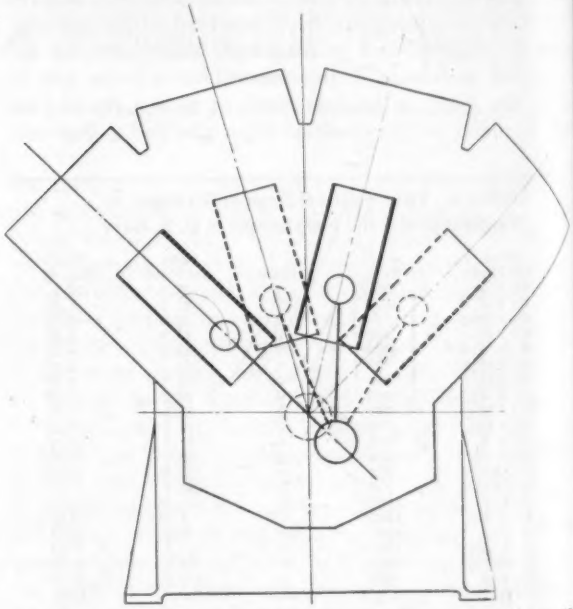
■ Fig. 6 – Standard V arrangement of two cylinders, built in 6, 8, 12, and 16 cylinders

development of units of greater number of cylinders, since only by so doing could the Navy obtain the desired ranges of engine output.

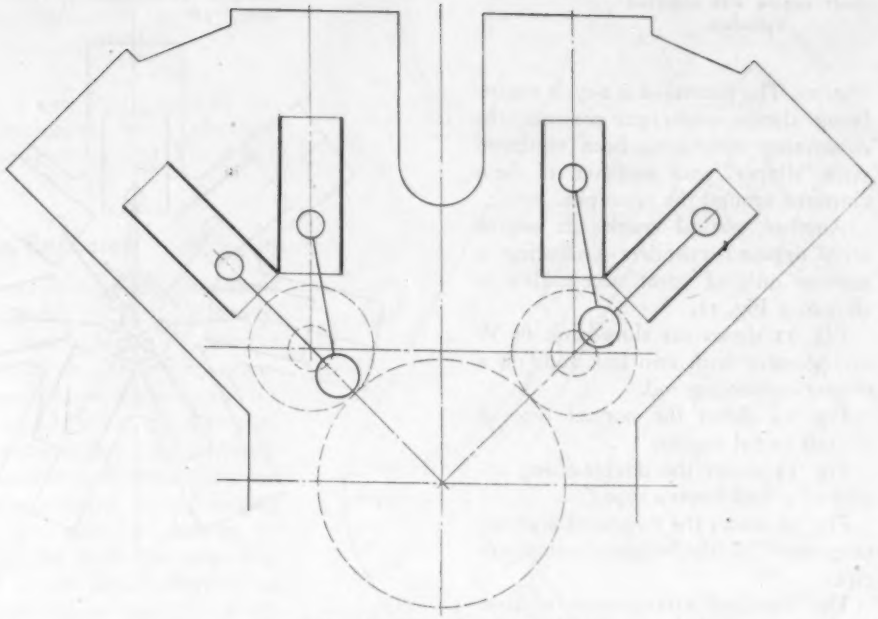
■ Cylinder Arrangement

Cylinder arrangement then became the problem, and I would like to illustrate diagrammatically all of the arrangements used to date. These have all been drawn to the same scale around a cylinder of the same bore and stroke and illustrate at a glance the space requirements of each arrangement.

The basic 2-cycle cylinder with uniflow scavenging through exhaust valves in the cylinder head is shown in Fig. 5. The standard V arrangement of two cylinders,



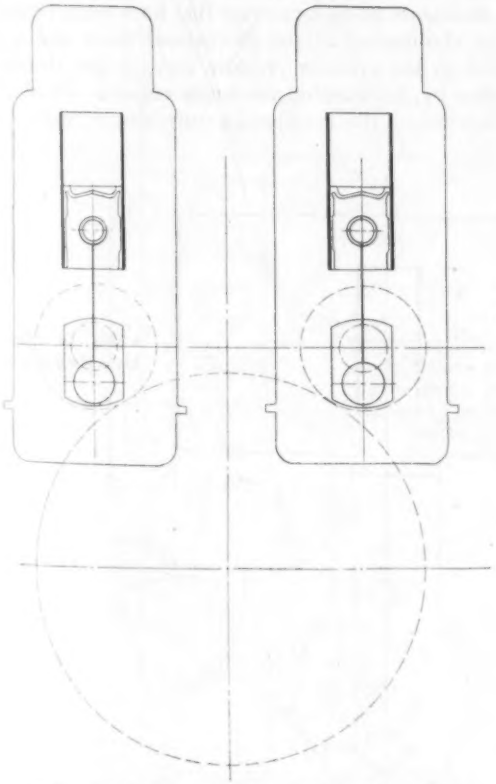
■ Fig. 7 – Porcupine type with four cylinders arranged as two V's



■ Fig. 8 – Double-V arrangement

built in 6, 8, 12, and 16 cylinders, both with side-by-side connecting rods and with fork-and-blade type of rods, is shown in Fig. 6.

The "porcupine" type with four cylinders or two V's arranged on a single crankpin and two pairs of fork-and-blade rods arranged side by side is illustrated in Fig. 7.



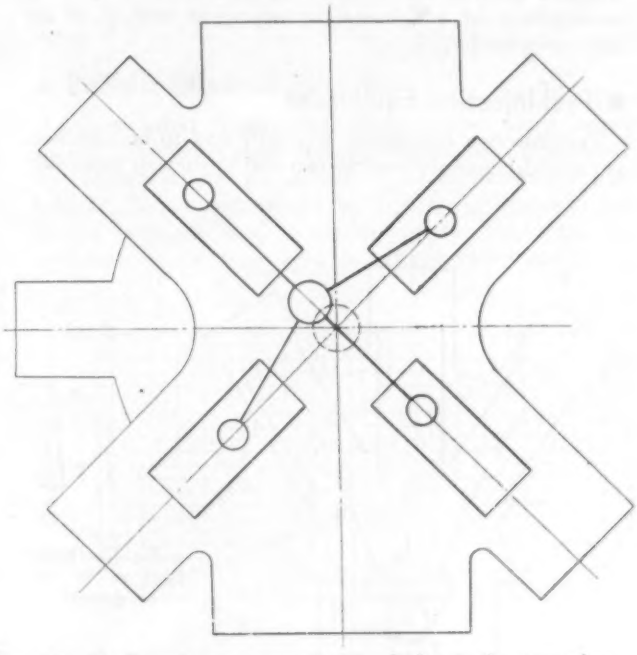
■ Fig. 9 – Quad or 4-engine unit of 24 cylinders

It is used in producing a 32-cyl unit on one crankshaft.

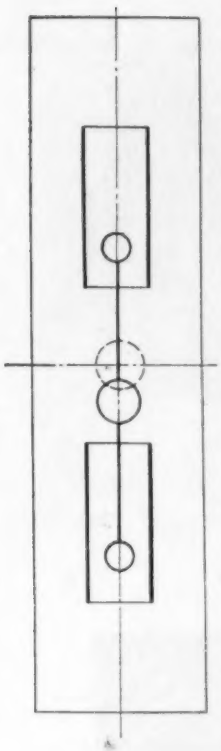
Double-V arrangement with a pinion between each pair of 4-throw crankshafts in tandem engaging a bull gear on the driving shaft is given in Fig. 8. This arrangement is used in another 32-cyl engine.

The quad or 4-engine unit of 24 cylinders has also a pinion between each pair of crankshafts in tandem, engaging a bull gear on the driving shaft, and is shown in Fig. 9.

The pancake arrangement of 4 cylinders in the same plane on a single pin of a vertical crankshaft is shown in



■ Fig. 10 – Pancake arrangement of 4 cylinders in the same plane



■ Fig. 11 – Vertical crankshaft engine with opposed cylinders

Fig. 10. The pistons of a 2-cycle engine being always under gas pressure, the connecting rods have been equipped with "slipper" ends and four of them clustered around the same pin.

Another vertical crankshaft engine with opposed cylinders producing a narrow unit of great accessibility is shown in Fig. 11.

Fig. 12 shows the three-bank or W arrangement with two link rods on a master connecting rod.

Fig. 13 shows the normal type of aircraft radial engine.

Fig. 14 shows the double-acting engine of a well-known type.

Fig. 15 shows the two-crankshaft arrangement of the opposed-piston engine.

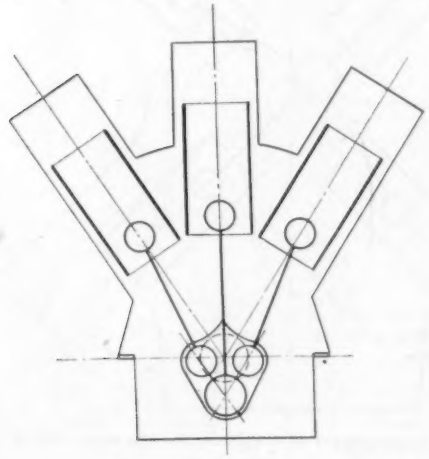
The diamond arrangement of four opposed-piston engines with eight pistons in the same plane through center of each crank is shown in Fig. 16.

■ Use of Gears

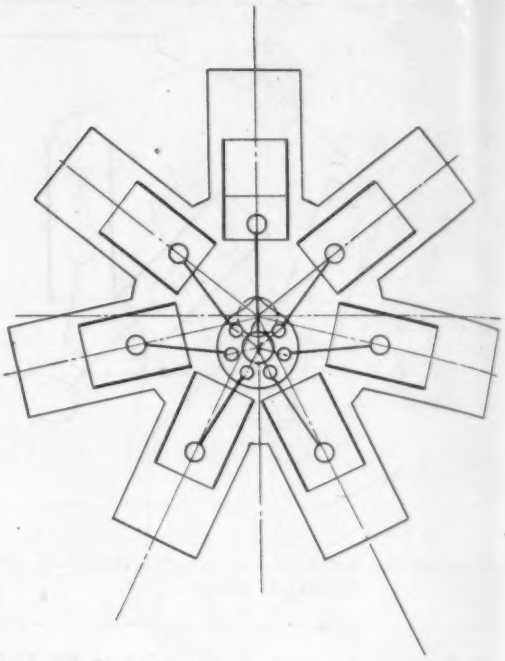
Use of gears became necessary with higher engine speeds, and to the dismay of many a pessimistic prophet was not accompanied by any tooth failures or excessive wear which was expected to result from the shock of cylinder explosions. Both spur and bevel gears have been successfully employed for the transmission of the crankshaft torque. With gears vindicated of any suspicion of unreliability, the use of light-weight high-speed engines will, no doubt, be greatly accelerated. Geared aircraft engines have been standard practice for some years and contributed to the development of a light-weight engine as well as of an efficient propeller.

■ Fuel-Injection Equipment

Fuel-injection equipment originally had to be classified as expendable or replaceable material to prevent tinkering

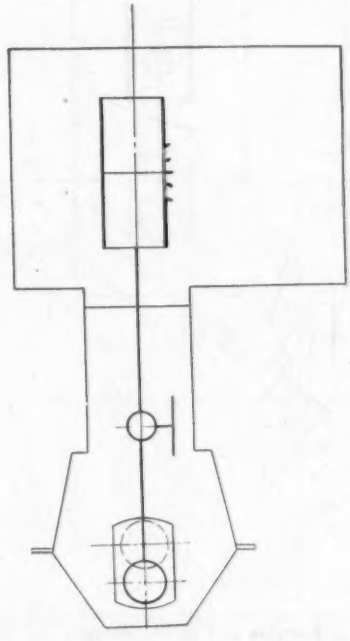


■ Fig. 12 – Three-bank or W arrangement



■ Fig. 13 – Normal type of aircraft radial engine

by incompetent hands and therefore its design was "frozen" to reduce the number of spares required. This lack of competition and incentive will and does retard its development. The cost of the present injection equipment represents too high a part of the total cost of the diesel engine and therefore new and simpler methods of solving this problem are definitely needed. Maybe the unit injector with its absence of high-pressure fuel lines could be simplified by elimination of the mechanical drive and a ram actuated by the cylinder pressure used in lieu thereof, as was done by Arschaouloff for larger engines. Or maybe a modification of the constant-pressure system with a dis-



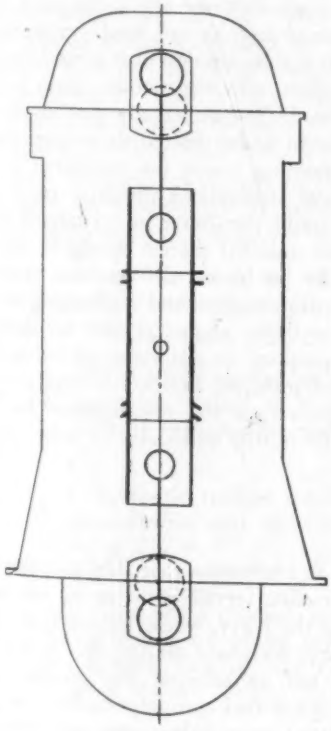
■ Fig. 14 – Double-acting engine

■ Fig. 16 – Diamond arrangement of four opposed-piston engines with eight pistons in the same plane

tributor deserves a new deal – some of the lowest fuel consumptions were obtained with this type of injection and it still has many advocates in Britain.

■ Combustion Research

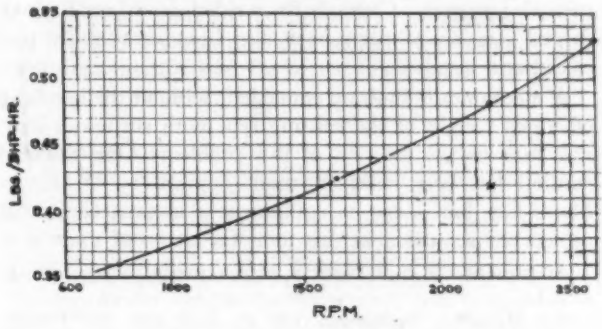
Combustion research was also eliminated to a great extent under the pressure of war, with the result that combustion-chamber characteristics which would not have survived normal competition continue in use. To provide an incentive for improvement, Fig. 17 gives a curve of best fuel consumptions at rated power and speed obtained on shop tests of engines of widely varying speeds – 700 to 2600 rpm. It may be remarked here that the high fuel consumptions at higher speeds can hardly be explained by speed alone and apparently reflect the lack of combustion research primarily.



■ Fig. 15 – Two-crankshaft arrangement of the opposed - piston engine

■ Ballistic Effect of Exhaust

Ballistic effect of exhaust as an aid or even the principal means of inducing supercharging has not been thoroughly appreciated despite the efforts of Buchi, Kadenacy, and others. Recent test results with a Buchi turbocharged engine seem to have indicated that his insistence on uncooled exhaust pipes between exhaust valves and turbo-



■ Fig. 17 – Best-fuel-consumption curve

charger to retain the temperature of exhaust gases is not the correct explanation of his excellent results since they were duplicated by a water-cooled exhaust header provided with gas-tight passages for each pair of noninterfering cylinders. Greater utilization of the kinetic energy in the slug of exhaust gases passing through the exhaust valve or port at the moment of its opening will lead to greater charge or supercharging effect of cylinders and with less power required of the mechanically driven blower or even if there is no blower and a similar intake manifold is provided. Practical considerations require a single exhaust manifold even for 16-cyl engines, but who can unscramble the interferences of these 16 cylinders with each other's charging processes? The result is a cut-and-try solution far from best obtainable, but comparable with others and therefore tolerable.

■ Strategic Materials

Scarcity of strategic materials and shortage of manpower, as another result of war, brought with them a number of different problems and have taught us many a lesson in economy on a national scale. Salvage, a heretofore despised concept, is becoming of greatest concern to all war activities. Reclamation of worn and damaged parts by welding, metal spray, and electroplating is practiced on a large scale in the Navy, and further research on different items is still in progress. It was this same impetus which finally overcame all objections to porous-chrome plating of cylinder liners, so that the problem of suitable wearing surface can be permanently divorced from that of the structural strength required. A steel liner with its better cooling and easier lubrication is thus feasible when chrome plated. Large Government and private facilities are now in existence for mass production of chrome-plated cylinder liners.

Steel crankshafts, particularly for larger engines, and of the desired degree of soundness of material and tolerance as to dimensions, are one of the most difficult parts to obtain under prevailing conditions. Existing production facilities were expanded to meet this emergency, but the fundamental scarcity of low-phosphorus pig iron to produce "acid" steel remains and with it come the inevitable "inclusions" of the "basic" steel used as a substitute. Research on the effect of these inclusions on the strength and rigidity of the shaft under stress is needed in order to perfect suitable standards for inspection. Meanwhile, the experience of the Ford Motor Co., which claims to have cast 8,000,000 crankshafts for automotive engines, has been used to produce a large cast-steel crankshaft for experimental purposes. Crankshafts welded from small drop-forged component parts have also been successfully produced and installed.

A mechanically built-up crankshaft without the handicap of added weight of the existing type with shrunk-in journals may be the solution of this problem. One patented method has been used in small engines by Waukesha Motor Co., but so far no equipment is in existence to cut the curved serrations of this joint for shafts of some size.

At present the most widely used remedy for the crankshaft bottleneck is the cast shaft from such patented compositions as Proferall and Meehanite.

■ Torsional Vibration

Torsional vibration seems to be still the perennial subtopic of all discussions on diesel engines and a legitimate excuse for flourishes of profanity in the engine room. The "hope of the world" at the moment is the Chilton "bifilar" or pendulum damper, consisting of a number of segmentally shaped weights with ingeniously arranged rollers for their suspension radially out when the engine is running and the weights are subject to the centrifugal force. There is no doubt as to the effectiveness of such a damper, the question is only how to avoid the "false Brinelling" effect resulting from the very small motions of these rollers under heavy load. This apparently is very similar to the experience of the automotive industry with ball and roller bearings during shipment of automobiles on trucks and in freight cars.[1]

The fact must not be overlooked, however, that the higher the speed of rotation of an engine and the larger the number of cylinders or crankthrows it has, the more criticals will fall within the operating range and the more urgent therefore becomes the need of reliably controlling them.

■ Vibration and Noise

Vibration and noise are also the inevitable consequences of going to higher speeds and I believe we are approaching, if we have not already reached the point, where continuous presence of the operator in the engine room may be injurious to his health on account of noise. Since it is not at all necessary for the engineer to be able to lay his hand on the engine at all times and he can read instruments from a distance through a glass window just as well, there is no valid reason, therefore, why the modern steam practice should not be followed. The boilers are now installed in an air-tight compartment under desired air pressure and separated from the adjoining room for auxiliaries and operating personnel under atmospheric pressure. Incidentally, the engine room could then serve as a charging air receiver and the engines installed therein would be automatically supercharged by the blower driven either by the engine itself or by a separate engine and discharging into the engine room. Automotive engineers will no doubt recognize in the above proposal an adaptation of the modern test cell used by the principal builders of high-speed engines. The idea, however, is too revolutionary for a seagoing engineer and he is still undecided as to his attitude toward it.

■ Conclusion

In conclusion I wish to emphasize that ship propulsion is very different from traction service, whether on rails or on highway, and even in the Navy, where full-speed ahead is not continuous as in the merchant marine, two or three days at full speed are not an unusual war occurrence. Furthermore, the efficiency of fuel consumption has a very real meaning in the Navy, quite independent and vastly more important than that suggested by the cost or even availability of fuel, and is designated as the cruising radius associated with coming home alive and victorious.

[1] See *Mechanical Engineering*, Vol. 59, June, 1937, pp. 415-422: "Lubrication and False Brinelling of Ball and Roller Bearings," by J. O. Almen.

SYNTHETIC RUBBER APPLICATIONS on Aircraft Engines

by GERTRUDE H. SPREMULLI

Ranger Aircraft Engines,
Division of Fairchild Engine & Airplane Corp.

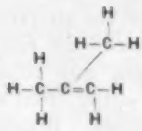
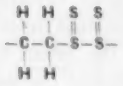
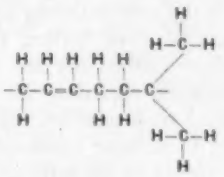
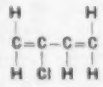
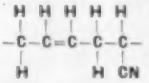
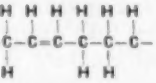
THE applications of synthetic rubbers on aircraft engines are characterized primarily by their variety and by the severity of the conditions which, in many cases, must be met. Filling the exacting demands of each application with the most suitable synthetic rubber requires both an accurate analysis of service conditions and careful testing of the synthetic rubbers.

Varieties of Synthetic Rubber – The term "synthetic rubber" will be used in this paper to designate those high molecular weight polymers whose *physical* properties can be compared to those of rubber. From the chemical point of view, of course, none of these substances is a "synthetic rubber," since they do not represent a synthesis of the hydrocarbon which constitutes the natural product. Actually, a variety of chemical compounds is represented by these polymers. Table 1 lists some of the more important of them along with the structural formulas of the monomers and their trade names.

There has been no attempt to include in this table all the polymers which might be considered to possess some rubber-like properties. Reference may be made to reviews

[This paper was presented at the SAE War Engineering-Annual Meeting, Detroit, Mich., Jan. 11, 1944.]

Table 1 – Varieties of Synthetic Rubber

Name	Monomer or Unit of Polymer	Trade Names
Polymerized vinyl chloride		Koroseal
Isobutene polymers		Vistanex
Organic polysulfides		Thiokol A
Isobutene copolymerized with either isoprene or butadiene		Butyl rubber
Polychloroprenes		Neoprene
Butadiene acrylonitrile copolymers		Buna N; Perbunan; Hycar
Butadiene copolymerized with styrene		Buna S (GR-S)

LABORATORY testing of synthetic rubber vulcanizates typical of those available for use on aircraft engines revealed that:

1. Synthetic rubber stocks vary greatly both in percentages and rates of weight loss during aging at 212 F.

2. Changes in physical properties of the stocks tested during oil aging are a function of both the temperature of testing and of the aniline point of the oil.

3. Gum extracted by aromatic fuel from synthetic rubber stocks varies with the temperature and, to a much smaller extent, with the length of time of extraction.

4. X-ray diffraction analysis provides a means of identifying some of the polymers and crystalline fillers in a stock, and also yields information regarding the particle size of the fillers present.

■

THE AUTHOR: GERTRUDE H. SPREMULLI has, in her two years as research chemist at Ranger Aircraft Engines, been concerned with setting up the synthetic rubber substitution program of the company. This has involved analysis of engine applications requiring synthetic rubber and testing of all types of synthetics. Dr. Spremulli studied at Heidelberg College in Ohio for her B.S., Western Reserve University for her M.S., and at Pennsylvania State College for her Ph.D.

by Wood[1] and Fisher[2] for more complete discussions of the varieties of synthetic rubber and for excellent bibliographies of the material available on this subject.

From the point of view of their use on the aircraft engine, only the last three types listed in the table have general importance. The first two, the polyvinyl chlorides and isobutene polymers, are not vulcanizable and the former is unsuitable for service conditions requiring temperatures exceeding 171 F. While the organic polysulfides are vulcanizable, they are thermoplastic and cannot be used in applications where a temperature greater than 158 F may exist. Butyl rubber has only recently been made available in commercial quantities so that its usefulness for aircraft-engine applications has not as yet been completely evaluated. It is known, however, that this synthetic is outstanding for its resistance to ozone and to sunlight aging. Its lack of oil resistance, however, limits its use for many engine parts.

For general use on the aircraft engine, then, there are to be considered primarily the polychloroprenes and the copolymers of butadiene with either acrylonitrile or styrene. These three classes differ from each other fundamentally in chemical constitution. Besides this, each type is itself capable of considerable variation. The conditions under which the polymerization is carried out and, in the cases of the copolymers, the ratios of polymers produce differences in the properties of the basic synthetics. In addition, modifications in the compounding ingredients and in time and temperature of cure further increase the range of properties obtainable from any one type. It is, therefore, difficult to generalize regarding the characteristics of one class as contrasted with another. A few distinctions may be made among these basic synthetic rubber types in spite of this normal spread in properties.

The polychloroprenes, or Neoprenes as they are more commonly known, are commercially available in several modifications. Neoprene types E and GN are for general purpose applications and FR has exceptional resistance to subzero temperatures. It should be noted that Neoprene ILS is a special member of the Neoprenes and is not included in this general discussion of them. Resistance to sunlight aging and stability in the presence of many chemicals are the outstanding characteristics of this group. The Neoprenes swell less than natural rubber in oils and retain their strength and elasticity to a much greater extent. Aromatic hydrocarbons, such as may be found in gasolines, will cause considerable swelling of these stocks. As far as general physical properties are concerned, the Neoprenes compare favorably with natural rubber. For use on the aircraft engine, a Neoprene stock would usually be recommended for those applications requiring moderately good oil resistance, good heat or cold resistance, and good general physical properties.

The synthetic rubbers produced by the copolymerization of butadiene with acrylonitrile (the buna N rubbers) are represented commercially by such products as Standard Perbunan, Perbunan Extra, and Hycar OR. Usually Chemigum is placed in this group although no definite information is available regarding its exact composition. The buna N rubbers are outstanding in their resistance to deterioration by oils and hydrocarbon solvents. They may be compounded for low compression set at elevated temperatures and for resistance to heat. With proper compounding they also show satisfactory resistance to low temperatures. Butaprene NF, which has recently been announced, exhibits unusually good cold resistance. Generally the butadiene-acrylonitrile copolymers will be used on the aircraft engine in those applications requiring exceptional resistance to oils and gasoline and low compression set.

The buna S synthetic rubbers are formed through the copolymerization of butadiene with styrene. When compounded with carbon black, the physical properties of this type are quite similar to those of natural rubber. Their aging properties and resistance to abrasion are good. Unfortunately, as far as many aircraft-engine applications are concerned, the similarity of the buna S synthetic rubbers to the natural product includes also lack of resistance to oils and to many solvents.

Recently the blending of one basic polymer with another to produce compounds having properties not otherwise obtainable has been announced. Martin and Laurence[3] have reported on the types of properties available by blending Thiokol FA with Neoprene GN, Hycar OR-15, and Perbunan 26. It is interesting to note that the properties of the blends do not vary as a linear function of the composition for all properties measured. Garvey and Emmett[4] have shown that the sunlight resistance of Hycar OR compounds could be improved by replacing part of the Hycar OR with Neoprene or Thiokol FA. For several of the more important applications of synthetic rubber on the Ranger aircraft engines, blends of polymers have produced compounds which, according to laboratory testing, have superior properties.

As a whole, the synthetic rubbers may be considered from two points of view:

1. As *substitutes* for rubber when an emergency has caused a shortage of the natural product; or

2. As *replacements* for natural rubber in those applications where the synthetic product performs more satisfactorily. At the present time, the buna S rubbers appear to be most useful as substitutes. The Neoprene and buna N synthetics may be considered as replacements for special applications. Natural rubber is probably superior to all synthetics in properties such as elasticity and rebound, low energy loss and low heat generation through hysteresis, and extensibility. The vulcanized synthetic rubber compounds, on the other hand, have been shown to excel in resistance to the action of air and ozone, permeability to gases, and resistance to the action of oils, organic solvents, and heat.

It follows, then, that for a great many applications of rubber on aircraft engines the synthetics must be considered as replacements rather than as substitutes. All those parts which come in contact with oils, gasoline, coolants, and hydraulic fluids give better service when made of a suitable synthetic than when manufactured from the natural product. Such parts include seals, hose, packings, diaphragms, and gaskets. Some typical parts in use on Ranger engines are shown in Fig. 1. Added to this list might be many special applications, particularly

[1] See National Bureau of Standards Circular C427: "Synthetic Rubbers – A Review of Their Compositions, Properties, and Uses," by Laurence A. Wood. U. S. Department of Commerce. 1940.

[2] See ASTM Proceedings, Vol. 41, 1941, pp. 521-551: "Natural and Synthetic Rubbers," by Harry L. Fisher.

[3] See *Industrial and Engineering Chemistry*, Vol. 35, September, 1943, pp. 986-991: "Blends of Thiokol FA with Neoprene GN, Hycar OR-15, and Perbunan 26," by S. M. Martin, Jr., and A. E. Laurence.

[4] See "The Sunlight Resistance of Hycar OR Compounds," by B. S. Garvey, Jr., and R. A. Emmett. Paper presented at the Fall Meeting of the Division of Rubber Chemistry, American Chemical Society, New York, October, 1943.

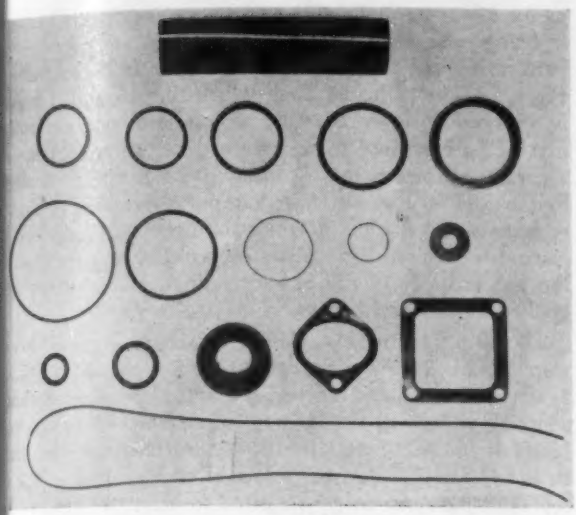


Fig. 1 – Typical synthetic rubber parts used on Ranger aircraft engines

if accessories are included. It is not intended to consider accessory parts in this discussion, but one example might be cited of an especially important use of Thiokol in the carburetor of a Ranger engine. The needle of the float-type valve was originally made of stainless steel. This pressed against a brass seat and with time a groove was worn so that the valve could not close properly. When Thiokol was substituted for the stainless steel tip, all the trouble was eliminated. Thiokol, of course, is fuel resistant and the temperature in this application does not exceed its useful range.

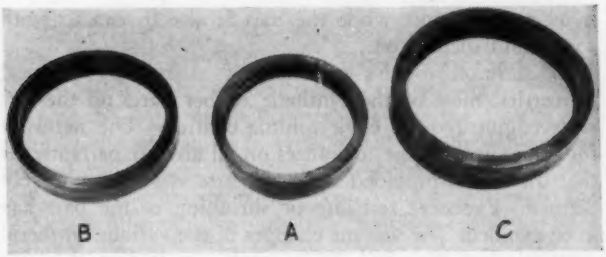
Problems in Synthetic Rubber Peculiar to Aircraft-Engine Applications – It was pointed out that a wide range of properties is available from any group of synthetics or from combinations of them. For many uses, therefore, the choice of a suitable synthetic rubber either as a replacement or substitute for the natural product is a relatively straightforward procedure. An analysis of the conditions to be encountered yields the necessary properties of the synthetic rubber stock to be used and that polymer or blend of polymers having those particular properties is selected. Unfortunately, however, for most aircraft-engine applications the situation is complicated by a number of factors, as, for example, the impossibility of accurately analyzing the service conditions. Thus, while the requirement of oil resistance for a seal is known, the types and compositions of the oils which might be used in the engine during service are unknown variables. Likewise, gasoline resistance may be found necessary for a diaphragm, but such an important factor as the aromatic content of the gasolines that might be used can only be estimated. It will be shown later in this paper that the aniline point of the oil and the aromatic content of the gasoline will determine the behavior of the various synthetic rubber stocks in these fluids.

Another variable which helps complicate the problem of selecting the proper synthetic rubber for aircraft engines is that of temperature. Not only is the actual operating temperature for a part difficult to determine accurately, but this temperature may vary through wide limits. For some applications the synthetic rubber used must remain serviceable over a range of temperature as wide as from −40 F to 300 F. In many cases this temperature requirement must be met while the part is also in contact with some fluid such as oil.

In addition to the retention of satisfactory physical properties, most of the synthetic rubber parts on the aircraft engine require close volume control. The necessity for maintaining close tolerances on all aircraft parts means that the spaces provided for such parts as seals are closely defined. Excessive swelling or shrinking of the part has to be avoided. The volume changes of the various synthetic rubbers must, therefore, be carefully checked so that the synthetic which will meet the demands of the space allotted can be selected. All the possible conditions which might be encountered in service must be anticipated in this testing and selection.

Preparation of the Aeronautical Material Specifications – Because their applications present unique problems in the field of synthetic rubbers, the manufacturers of aircraft engines found it necessary to take special action toward obtaining the proper synthetic for each of their requirements. This action was undertaken by the Aircraft-Engine Materials and Processes Committee in collaboration with the Airframe and Aircraft Accessory Materials and Processes Committees. These committees, which had been formed as a part of the SAE Aeronautics Division, undertook the task of studying the requirements and conditions to be met, organizing the available data on synthetic rubbers, and preparing suitable specifications to meet the rigid demands of the aircraft engine. It was in March, 1942, that the common need for such specifications was generally discussed. In April of that year the Aircraft-Engine Committee proposed fourteen different specifications to meet various types of requirements. In the following August, these proposed specifications were thoroughly discussed by the manufacturers and the users in the light of data collected by members of the above committees. Throughout the preparation of these specifications valuable help was provided by the members of several SAE-ASTM technical committees on rubber products. They furnished general information and data on different types of synthetic rubbers. Vendors of synthetic rubber stocks collaborated by supplying their data on the range of properties which could be expected from the stocks then being compounded. Further study of requirements to be met and testing of synthetic rubber stocks were carried out by the committees until November, 1942, when 21 Aeronautical Material Specifications covering synthetic rubber compounds were written and approved for publication the following month. These specifications were revised in August, 1943, on the basis of suggestions by Subcommittee V of SAE-ASTM Technical Committee A and on the results of further study by the members of the Materials and Processes Committees. They represent in general the best compromise possible between what would be ideal from the point of view of the aircraft-engine manufacturer and what it is possible to produce by expert compounding of the basic synthetic rubbers available at the present time.

With few exceptions, the Aeronautical Material Specifications are written around particular application requirements and do not specify the synthetic rubber to be used. Any synthetic rubber which can be compounded to give the required properties may be approved for the specification. Thus it is possible for the users to select not only the correct basic polymer but also the compounded stock of that polymer which will most nearly meet the requirements of their applications. Fig. 2 illustrates the supe-



■ Fig. 2 – Synthetic rubber packings before and after immersion for one week in 40% aromatic fuel – A: size of part before immersion; B: after immersion, part made from synthetic rubber selected by specification; C: after immersion, part made from synthetic rubber selected by name

riority of material selected by specification rather than by trade name or synthetic rubber type. An intake pipe packing, Fig. 2A, was originally made from a well-known synthetic rubber, selected only by its name. Later the application was covered by a suitable specification. Since, for good service performance, this part must be fuel resistant, the relative serviceabilities of the two were compared by immersing them in 40% aromatic blended fuel for one week. The fact that the material selected by specification, Fig. 2B, shows only a small volume change while the other is highly swollen, Fig. 2C, is evidence of the former's superiority for the application.

The procedure involved in selecting synthetic rubbers for various aircraft-engine applications may be divided into two general classifications: application analysis, and the chemical and physical testing of the synthetic rubber stocks. The application analysis is concerned with setting up the limiting conditions to be met by the part in service. It includes consideration of the necessary resistance to fluids, proper design of the part, selection of the general physical properties required, and finally selecting the specification covering these requirements. The chemical and physical testing of various synthetic rubber vulcanizates then follows to select the stocks meeting the required specification and to check on any other properties which might be important as far as the service life of any one part is concerned. This laboratory testing must include not only laboratory samples but also the finished parts, for which special adaptations of the standard tests may be required.

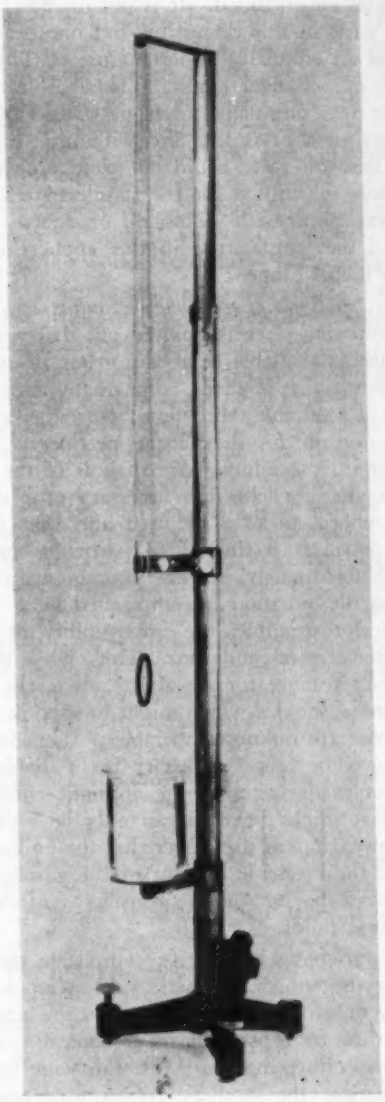
■ Physical Testing Methods

Before proceeding with the discussion of the application analysis and general laboratory testing, brief mention will be made of the more important physical test methods used. Changes in such physical properties as volume, tensile strength, and ultimate elongation form the criteria upon which the resistance of a synthetic rubber to deterioration by fluids or to various types of aging is based.

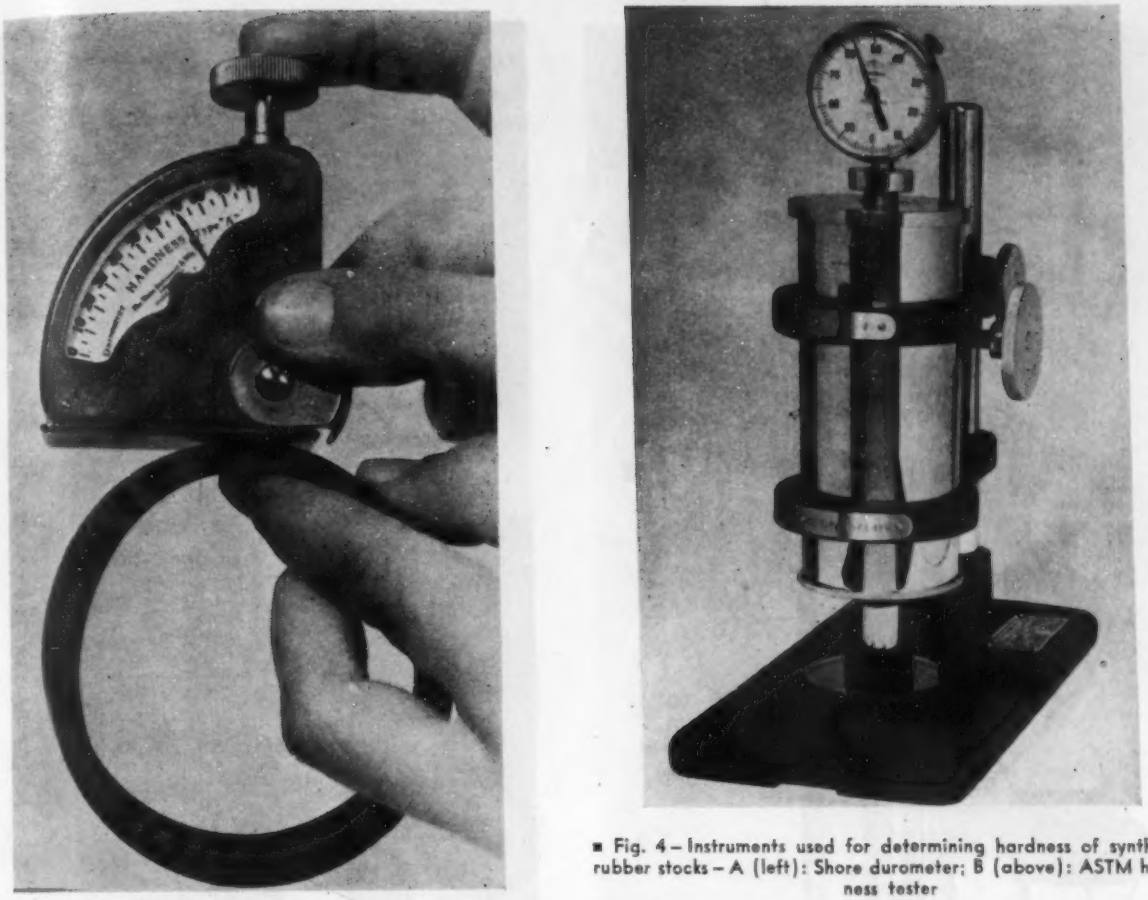
All volume measurements reported in this paper were made with the Jolly spring balance, which is illustrated in Fig. 3. The rubber sample is weighed in air and then in distilled water. The difference in weights is proportional to the volume and, when desired, the actual volume may be obtained through the use of a suitable calibration curve. Since, for the studies reported in this paper, the interest was in the percentage changes in volume during aging, no conversion into volume units was required.

The Shore durometer (scale A) was used for the determinations of hardness. While it is recognized that this method is open to much criticism, it was chosen because it lends itself most conveniently to testing the finished part. The use of the Shore durometer is illustrated in Fig. 4A. The principle upon which this method is based is resistance to indentation by an impressor pin, the hardness being measured in terms of kg per sq cm. A more accurate instrument for hardness testing has been adopted by the ASTM, Fig. 4B. The hardness value thus obtained represents indentation in thousandths of an inch. This instrument is useful for laboratory samples and for checking the accuracy of the Shore readings, but it cannot be applied to most finished parts.

The values for tensile strength which are reported here were obtained using a Scott rubber, tester, Model L6. Several types of jaws can be used with this machine, thus adapting it to both the laboratory samples and to various kinds of finished parts. Its use is illustrated in Figs. 5A, B, and C. In all cases, the tensile strength values reported represent the force in psi required to break the specimen. The cross-sectional area is always calculated on the basis of the unaged specimen. The values for ultimate elongation



■ Fig. 3 – Jolly spring balance used for determining volume changes of synthetic rubber parts



■ Fig. 4 – Instruments used for determining hardness of synthetic rubber stocks – A (left): Shore durometer; B (above): ASTM hardness tester

are obtained simultaneously with those for tensile strength. They represent the percentage the rubber can be stretched before it breaks.

■ Application Analysis

Determination of Required Fluid Resistance – In analyzing the requirements of a particular synthetic rubber part for the aircraft engine, the first consideration is whether resistance to any fluid, such as oil or gasoline, is necessary. It is, of course, well known that the various types of synthetic rubbers differ in the degree to which they are able to resist deterioration by such fluids. When considering the changes of synthetic rubber stocks in various fluids, it must be remembered that, in addition to the type of basic polymer used, the ingredients added during the compounding of the stock also influence the changes in properties. As one example of this, the work of Juve and Garvey[5] on the compounding of Hycar OR stocks may be mentioned. They showed that certain softeners would cause increased swelling while others caused shrinkage and hardening due to loss of the softener by evaporation and extraction. From the point of view of the users of the synthetic rubbers, therefore, it is more valuable to study the compounded stock as it is made available for use on the engine than to judge by the reactions of the basic polymers. The stocks listed in Table 2 illustrate the type of changes shown by synthetic rubbers in fluids such as might be encountered on the aircraft engine. These synthetic rubber vulcanizates are not to be considered as typical of the basic polymer, but rather as representative of stocks approved for various uses on Ranger engines.

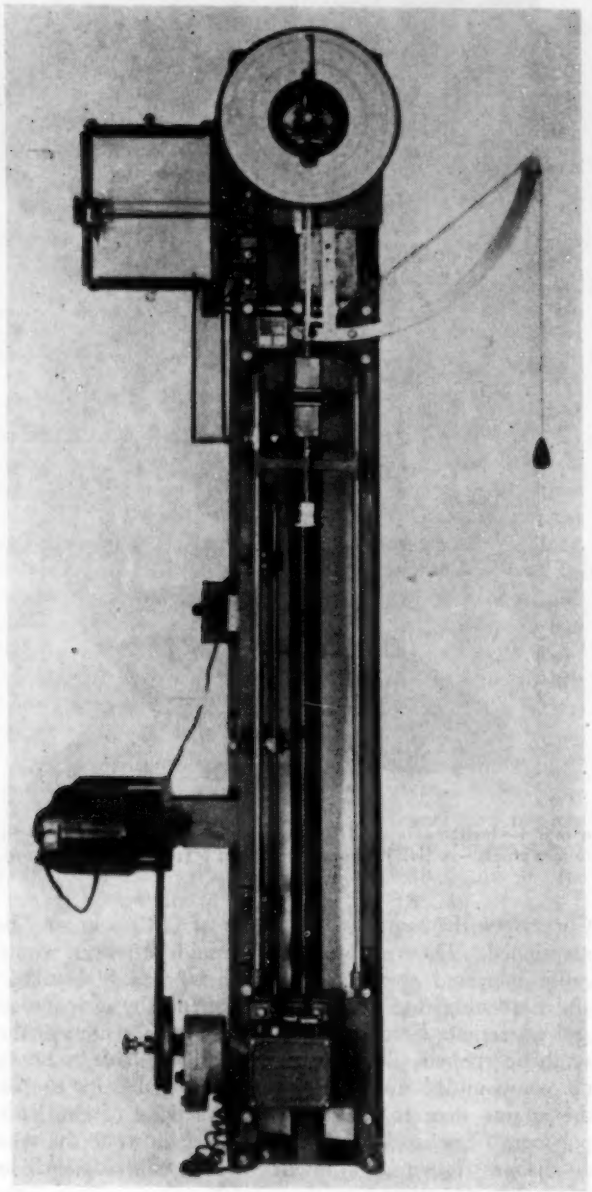
The differences in fluid resistance shown in Table 2 by

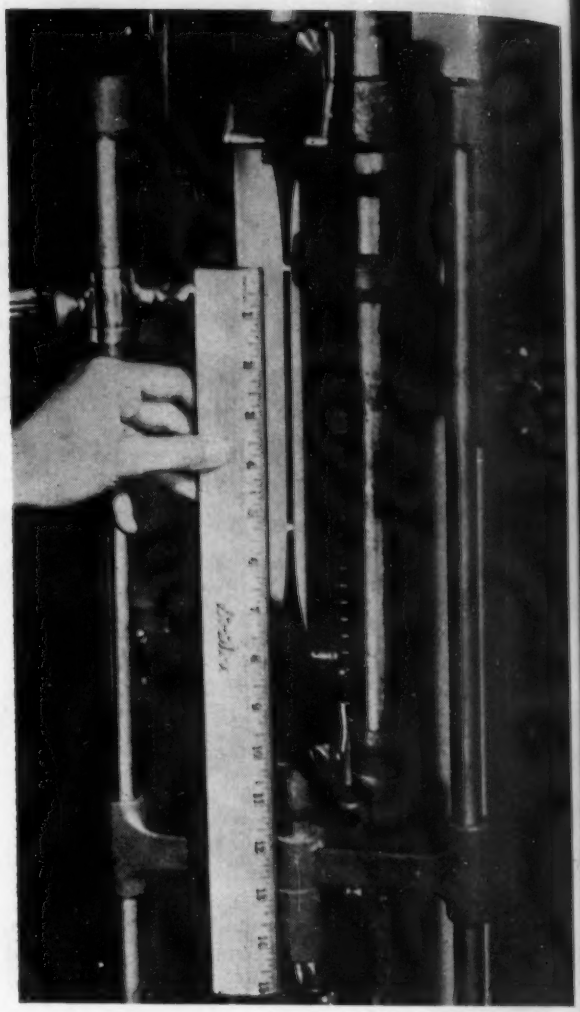
[5] See *Industrial and Engineering Chemistry*, Vol. 34, November, 1942, pp. 1316-1319: "Volume Change of Hycar OR-15 Compounds on Immersion," by A. E. Juve and B. S. Garvey, Jr.

Table 2 – Fluid Resistance of Various Types of Synthetic Rubber Stocks

Base Polymer	Aircraft-Engine Oil [a]				65-Octane Gasoline [b]				Ethylene Glycol [c]			
	Volume Change, %	Hardness Change[d]	Tensile Strength Change, %	Elongation Change, %	Volume Change, %	Hardness Change[d]	Tensile Strength Change, %	Elongation Change, %	Volume Change, %	Hardness Change[d]	Tensile Strength Change, %	Elongation Change, %
Neoprene	+1.5	+10	+24	−54	+22	− 1	−18	− 8	+21	− 4	− 9	−57
Hycar OR	−16	+30	+13	−56	+ 1	+ 8	−22	−19	− 6	+13	+13	−50
Perbunan	−6.0	+ 7	+15	−40	+ 5	0	+10	−17	− 7	+ 3	+33	−17
Perbunan + Hycar	−9.0	+19	+35	−62	+ 3	− 5	−19	−25	− 3	+12	+33	−63
Buna N, Buna NXM, Buna S	+0.3	+ 5	− 7	−44	+23	− 9	−49	−36	+ 7.5	+ 5	−18	−43

[a] 70 hr at 300 F.
[b] One week at room temperature.
[c] 70 hr at 300 F.
[d] In Shore durometer "A" units.





■ Fig. 5 – Scott rubber tester for obtaining tensile strength and ultimate elongation values of synthetic rubber compounds – A (left): entire apparatus; B (above): used with standard tensile dumbbells; C (right): used with finished engine parts

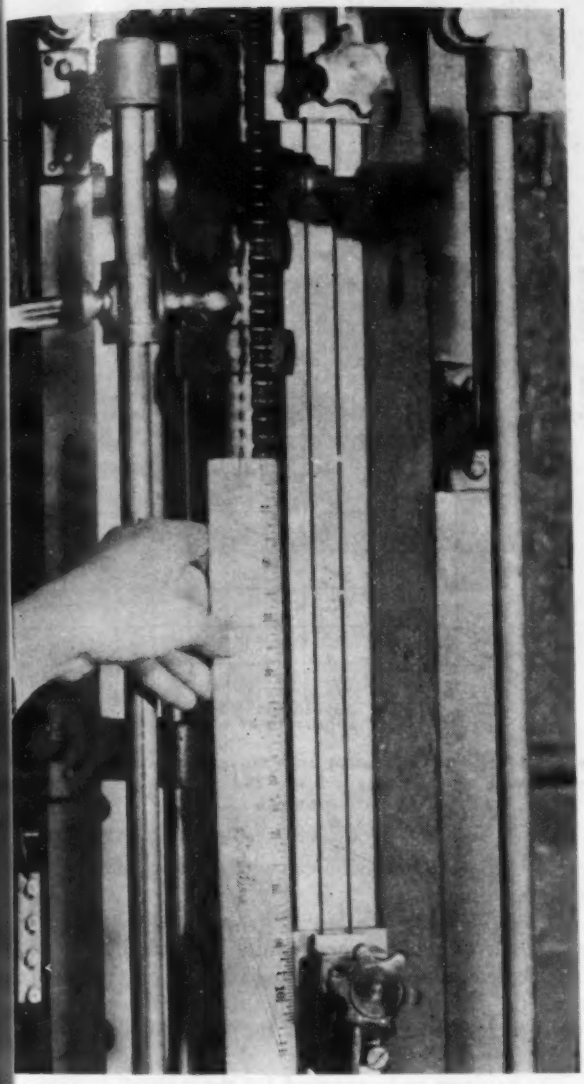
the various synthetic rubber stocks are greatly magnified when some of the variables affecting their changes in these liquids are considered. It was pointed out earlier in the paper that the oil resistance of these compounds is a function of the aniline point of the oil with which it comes in contact. The aniline point of the oil is defined as the critical solution temperature of a mixture of the oil and an equal volume of aniline. It is widely used as a measure of the aromatic content; however, it is affected not only by aromatics but, in a lesser degree, also by naphthenes and by the average molecular weight of all constituents, including the paraffins. A number of workers have shown that the aniline point of an oil will indicate its probable swelling effect upon a synthetic rubber stock. Carman, Powers, and Robinson[6] found that, in general, the logarithm of the per cent volume changes of synthetic rubbers in mineral oils varied inversely with the aniline point.

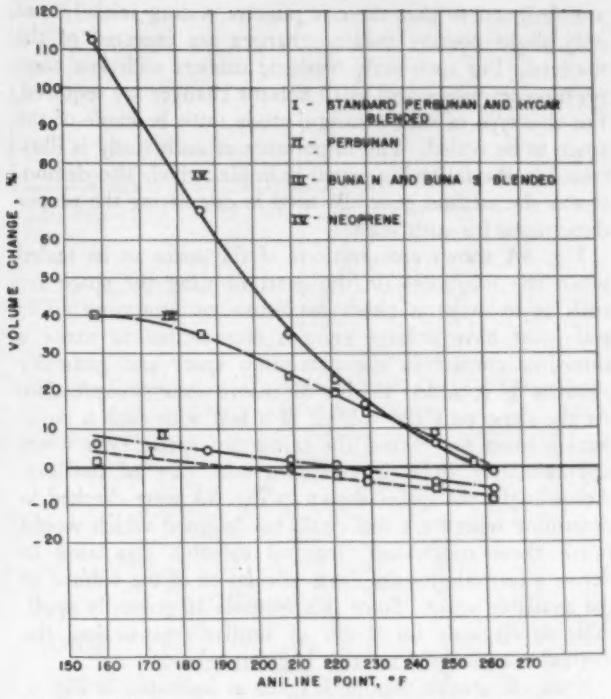
[6] See *Industrial and Engineering Chemistry*, Vol. 32, August, 1940, pp. 1069-1072: "Swelling of Synthetic Rubbers in Mineral Oils; Effect of Variation in Aniline Point of Oils," by F. H. Carman, P. O. Powers, and H. A. Robinson.

Fraser[7] worked with lubricating and hydraulic oils and concluded that the aniline point and the viscosity-gravity constant may be used as criteria of the swelling effect of the oils on Neoprene vulcanizates. Since the aircraft-engine manufacturer cannot control all the oils which may be used in the engine in its various fields of service, he must anticipate and prepare for oils of varying aniline point and, if possible, choose a synthetic rubber which will be satisfactory even when used with the worst type of oil that might be encountered. This necessity for resistance to such a wide range of oils considerably reduces the number of suitable synthetic rubber stocks.

To illustrate the effect of aniline point variation upon the volume changes of typical stocks available for aircraft-engine use, oils with aniline points varying from 158 to 262 F were prepared by mixing varying proportions of an aircraft-engine lubricating oil with Circo light processing oil. Some of the physical constants of these oils are listed in Table 3. Samples of the various synthetic rubber stocks were immersed in these oils at 300 F for 70 hr. The resulting volume changes are plotted as a function of the

[7] See *Rubber Chemistry and Technology*, Vol. 14, 1941, pp. 204-210: "Effect of Lubricating and Hydraulic Oils," by D. F. Fraser.





■ Fig. 6 – Volume changes of four typical synthetic rubber stocks plotted as a function of the aniline point of the oil – 70-hr immersion at 300 F

aniline point in Fig. 6. Since the viscosities and viscosity indexes of these oils are varying simultaneously with the aniline points, the volume changes could be plotted as functions of these variables also. The aniline point is probably the easiest and most rapid of these variables to determine and it does serve as means of predicting the effect an oil will have on a synthetic rubber stock.

It has been known for some time[8] that aircraft engines are being operated on aromatic fuel blends in certain localities. Fuel containing 20% by volume aromatics is common and an aromatic content as high as 40% is not considered improbable. The addition of aromatic constituents to gasoline has a marked effect upon the synthetic rubber parts which come in contact with that gasoline. To show the effect of varying the aromatic content upon the volume changes of typical synthetic rubber vulcanizates, a series of gasolines was prepared by adding 0, 8, 16, 24, 32, and 40% aromatics by volume to 65-octane gasoline. In all cases the aromatic components were benzene, xylene, and toluene in the following proportion: 1:3:4. The volume changes of typical stocks immersed in these fuels for 24 hr at room temperature are plotted as a function of the percentage aromatics in Fig. 7. While the function is linear in all cases, the slopes vary with the type of synthetic used.

It is apparent from Figs. 6 and 7 that analyzing the requirement of fluid resistance involves consideration not only of present operating conditions but also an estimate of what might happen under the worst possible conditions.

Design Analysis – The importance of proper design of synthetic rubber parts is often overlooked. It has been found that the failure of a synthetic rubber part to function satisfactorily has at times been attributed to faulty material when study has revealed that the trouble actually lay in the design of the parts forming the space to be sealed. Application analysis, therefore, should include a study of design, especially of those parts which are to function as seals against various fluids.

There are several theories regarding the manner in which rubber seals should function. Some seals are designed so that they depend upon the swelling action of the material to do the sealing. High-swelling synthetic rubbers are required for such seals. This type of sealing is open to the criticism that, upon swelling, many synthetics show a deterioration in other physical properties. There is a softening of the material and often loss in tensile strength and ultimate elongation. The other type of seals

[8] See War Department, Air Corps, Matériel Division Memorandum Report #Exp.-M-57-531-112: "Aromatic Fuel Blends in Tactical Aircraft." December, 1941.

Table 3 – Physical Constants of Oils Prepared by Mixing Varying Amounts of Circo Light Processing Oil with Aircraft-Engine Lubricating Oil

Processing Oil, % by volume	Aniline Point, F	Viscosity at 100 F, sec	Viscosity at 210 F, sec	Viscosity Index
0	257	1707	121	97.1
20	247	1052	89	92.5
40	229	646	68	86.9
50	221	516	62	84.9
60	209	394	55	79.3
80	186	248	46	52.8
100	158	157	41	16.9

are designed so that there is positive sealing initially and only slight positive volume changes are expected of the material. For such seals synthetic rubbers with low compression set values and small volume changes are required. For this type of seals a careful study must be made of the space to be sealed. The importance of such study is illustrated in the following actual example, which also demonstrates the method generally used in calculating the proper dimensions for such seals.

Fig. 8A shows cross-sections of the space to be sealed when the tolerances on the parts forming the space are such as to make it either maximum or minimum. The seal must have a large enough cross-section to make a three-line contact in the maximum space and generally about 5% is added to this to ensure some compression. At the same time the volume of a seal with such a cross-section must not exceed the minimum space, even when approximately 15% is added to take care of swelling. Accordingly, the spaces shown in Fig. 8A were checked to determine whether a seal could be designed which would fulfill these conditions. Integral calculus was used to derive a formula for the exact calculation of the volume of the available space. Since this formula is generally applicable to all seals for spaces of similar cross-section, the method used in its derivation is given below:

If the element of volume is taken as indicated in Fig. 9, then

$dv^1 = x\ d\ \theta\ dx\ dy$ and integrating θ from 0 to 2π

$$dv = 2\pi\ x\ dx\ dy$$

$$\frac{v}{2\pi} = \int dy \int x\ dx$$

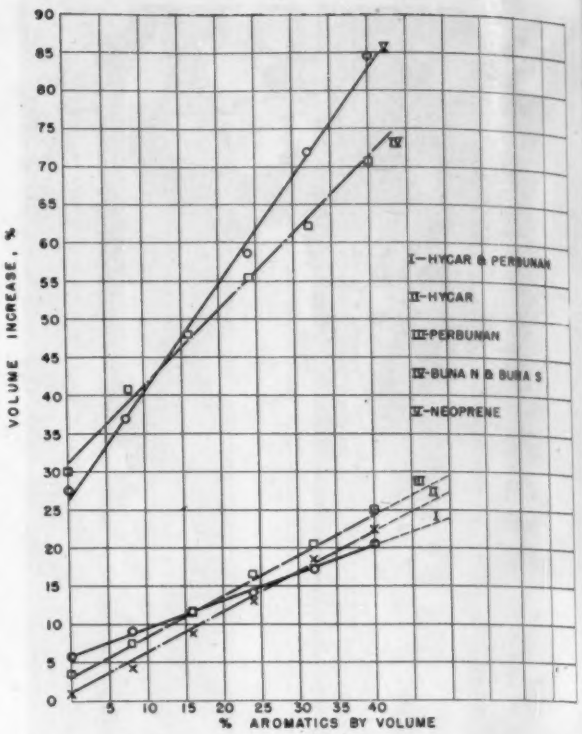
$$\frac{v}{2\pi} = \int_0^n dy \int_c^{\frac{y+b}{a}} x\ dx + \int_n^{y_2} dy \int_{m-\sqrt{R^2-(y-n)^2}}^{\frac{y+b}{a}} x\ dx + \int_{y_2}^{p} dy \int_{m-\sqrt{R^2-(y-n)^2}}^{x_2} x\ dx$$

From which it follows that

$$\frac{v}{\pi} = \frac{(y_2+b)^3 - b^3}{3a^2} - c^2 n - (m^2+R^2)\ (p-n) + x_2^2\ (p-y_2) + \frac{(p-n)^3}{3} + m\ R^2\ \frac{\pi}{2}$$

Using this formula the volume of the minimum space, Fig. 8A, was found to be 1.67 cc. However, the volume of a ring which would just make a three-line contact in the maximum space was 1.68 cc. Obviously, it was not possible to design any seal which would be satisfactory under all possible conditions in this case. It was, therefore, necessary to suggest a redesign of the parts forming the space for the seal. The new dimensions are illustrated in Fig. 8B. This new design did not present an ideal condition, but it represented the best compromise which could be made as far as the other parts were concerned. An analysis of the spaces available in this figure revealed that the volume of the minimum space was 4.10 cc. The diameter of the cross-section of a seal which would be under 5% compression in the maximum space was 0.133 in. This corresponded to a volume of 3.75 cc, permitting approximately 10% swelling of the seal when the space was a minimum.

It is evident from the above that to be ideal, from the point of view of sealing, this type of cross-section should present the maximum area with a minimum fillet-to-chamfer distance.



■ Fig. 7 – Effect of increasing aromatic content on volume changes of synthetic rubber stocks in fuels – 24-hr immersion at room temperature

Determination of Required Physical Properties and Temperature Resistance – The application analysis must also include determining the required physical properties of the synthetic rubber as received, the necessity for resistance to high and low temperature, and the compression set requirements.

The hardness to be specified for the material as received depends upon several factors. The general function of the part is first considered. For example, seals usually are required in the hardness range of 50 to 70 Shore durometer, while packings often function more satisfactorily when selected in a somewhat harder range. The changes likely to occur in the synthetic during service should also influence the selection of the original hardness value. If the part is to come in contact with a fluid which will cause it to swell appreciably, then softening will probably result and accordingly a fairly hard material should be chosen. On the other hand, if the part is not going to swell, but may be exposed to very high or to very low temperature, which will cause it to harden during service, a sufficiently soft material should be used initially.

The tensile strength and ultimate elongation requirements for the material as received are established not so much on the basis of the application requirements as upon values necessary to obtain a good quality synthetic rubber. As mentioned previously these values are most important from the point of view of indicating the deterioration of the rubber during aging.

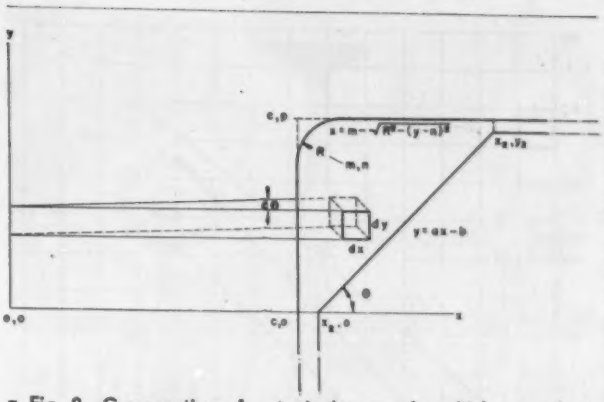
Many applications of synthetic rubbers on the aircraft engine require that cold flow be a minimum. Compression set is generally used to measure the cold flow properties of a synthetic rubber. In determining the compression set

requirement, the temperature of the service conditions should be considered, since the amount of compression set is a function of the temperature. This is shown very clearly in some data obtained by the General Motors Research Laboratories Division for the SAE-ASTM Sub-Committee V,[9] which are plotted in Fig. 10.

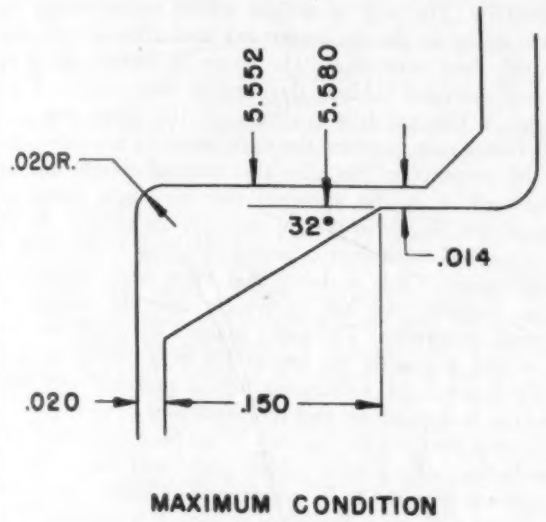
An estimate of the temperature requirements for the part is often a difficult one to make. Generally, the highest possible temperature which the part may reach under normal conditions is selected as the top requirement. As far as the cold resistance is concerned, all aircraft synthetic rubber parts are required to meet at least —40 F, and those for special applications specify —70 F.

The final step in the application analysis is the selection of a specification which will cover the requirements as determined in the preceding steps. The Aeronautical Material Specifications have been written so that once the fluid resistance and the initial hardness requirements are known, the other demands of the aplication, such as heat

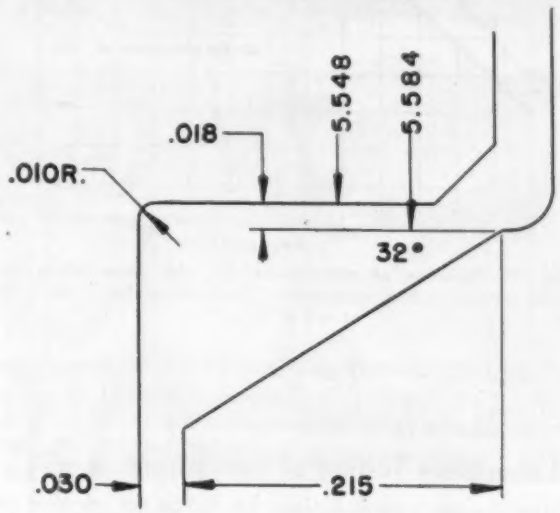
[9] These data are presented through the courtesy of W. M. Phillips and A. J. Kearfott of the Rubber and Plastics Laboratory, General Motors Research Division.



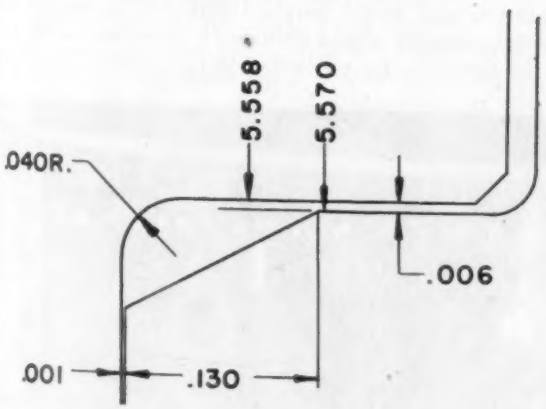
■ Fig. 9 – Cross-section of a typical space for which a synthetic rubber seal must be designed



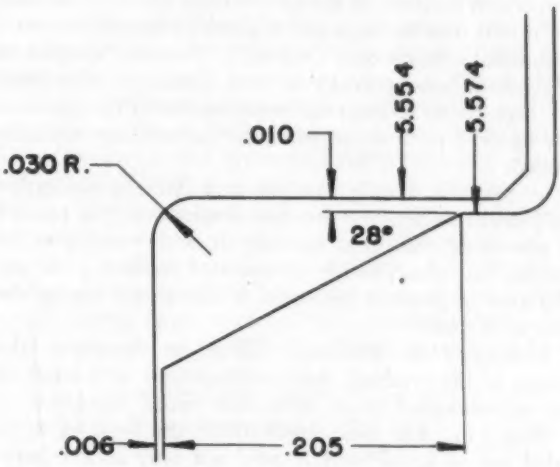
MAXIMUM CONDITION



MAXIMUM CONDITION

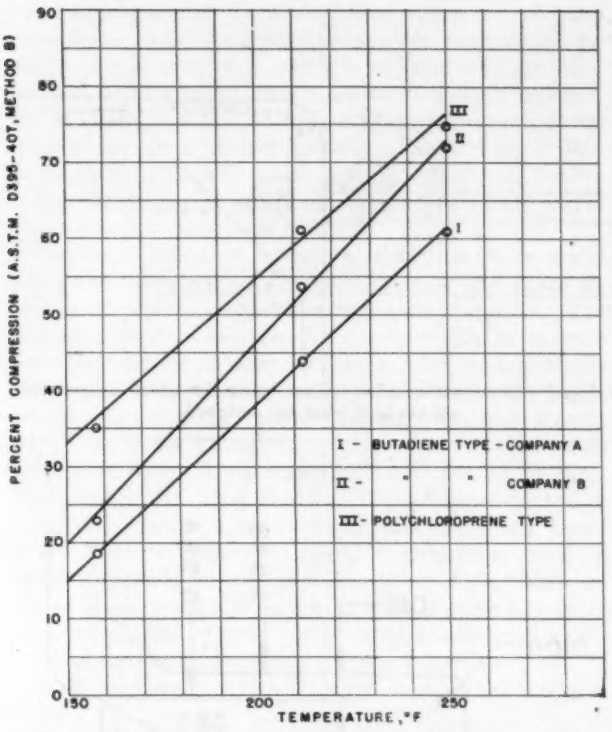


MINIMUM CONDITION



MINIMUM CONDITION

■ Fig. 8 – Cross-sections of a space to be sealed by a synthetic rubber part – A (left): as originally designed; B (right): as re-designed after calculation


PERCENT COMPRESSION (A.S.T.M. D395-40T, METHOD B)
TEMPERATURE, °F
I - BUTADIENE TYPE - COMPANY A
II - " " COMPANY B
III - POLYCHLOROPRENE TYPE

■ Fig. 10 – Variation of compression set with temperature for typical synthetic rubber compounds (Shore durometer = 60 ± 5) at end of 70 hr

and cold resistance and compression set, are automatically satisfied. The procedure involved in the selection of a suitable specification is outlined diagrammatically in Table 4.

■ Laboratory Testing of Synthetic Rubbers

After the application analysis has led to the selection of a specification for a particular part, the problem of checking various synthetic rubber compounds for conformance with the specified requirements remains. The laboratory testing consists primarily of routine checking according to the methods outlined in the specification, but in some cases special tests may be employed to check properties necessary for unusual applications. Generally, laboratory samples of the synthetic rubber stocks are first tested and after these have been found to meet the requirements of the specification, finished parts made from the material are similarly checked.

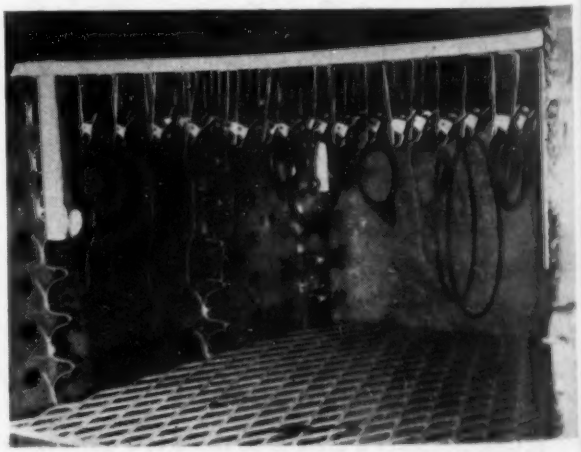
The hardness, tensile strength, and ultimate elongation of all stocks as received are first determined. As pointed out previously, these tests not only check the quality of the material, but also provide a means of following the deterioration in physical properties which occurs during the aging of the stock.

Checking Heat Resistance – Tests to determine the changes in the synthetic rubber compounds as a result of heat are conducted in an oven with forced circulation of air (Fig. 11). For most applications, the oven aging is carried out at 212 F and is used not only as a relative measure of the stock's resistance to that temperature but also as an accelerated test indicative of changes likely to occur in the synthetic rubber during storage over a period of time. For applications requiring special heat-resisting compounds, the tests are conducted at 300 F. Routinely, the changes in hardness, tensile strength, and elongation are used to judge the stock's resistance to elevated temperatures. The loss in weight which occurs during the oven aging is also an important indication of the compound's heat resistance. The losses in weight shown by typical synthetic rubbers during one week at 212 F are shown in Fig. 12. It is evident from this figure that great differences exist between the various stocks not only as far as the percentages but also the rates of weight loss are concerned. It is also apparent that no single determination of the change in weight can give an accurate picture of what will occur over a period of aging at the elevated temperature. Table 5 shows that there is no rigid correlation between the loss in weight and the changes in physical properties. The exact manner in which this loss of weight will affect the service life of a part is not definitely known. Its seriousness would probably vary with the type of application and the conditions of service.

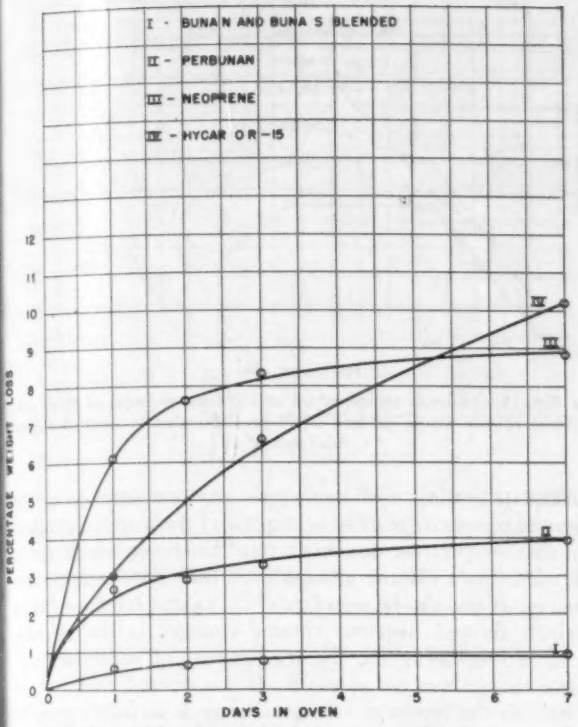
Table 4 – Procedure for Selection of a Specification for a Synthetic Rubber Part

Requirements						Specification
First	Second		Third			
Resistance to:						
Weather	Hardness	Soft				AMS-3207
		Medium				AMS-3208
		Hard				AMS-3209
Heat	Hardness	Soft				AMS-3281
		Hard				AMS-3282
Cold	Hardness	Soft				AMS-3284
		Hard				AMS-3285
Aromatic fuel	Hardness	Soft				AMS-3212
		Medium				AMS-3215
		Hard				AMS-3213
Oil	Swell	High	Hardness	Soft		AMS-3222
				Hard		AMS-3223
		Low	Hardness	Soft		AMS-3226
				Medium		AMS-3227
				Hard		AMS-3228
				Very hard		AMS-3229
Oil and gasoline						AMS-3221
Hydraulic fluid						AMS-3208
EP lubricant						AMS-3286
Coolant						AMS-3211

Testing for Cold Resistance – It has been mentioned previously that all synthetic rubber parts used on an aircraft engine are required to be resistant to at least —40 F, and for some applications flexibility at —70 F is required. The apparatus used in the standard SAE-ASTM method for the determination of cold resistance is illustrated in Fig. 13. In this method strips 4 x ¼ x 0.075 in. are placed in a loop



■ Fig. 11 – Interior of oven used for determining heat resistance of synthetic rubber compounds

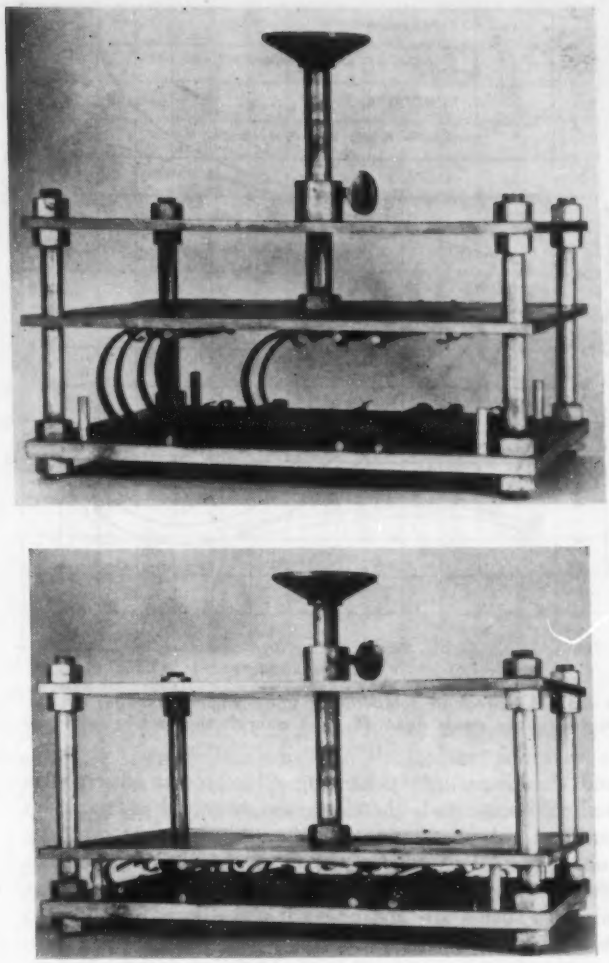


■ Fig. 12 – Rate of loss in weight during oven aging for one week at 212 F for four synthetic rubber stocks

position between jaws initially 2½ in. apart. After exposure to cold dry air at −40 or −70 F for 5 hr, the jaws are rapidly brought together until they are 1 in. apart. The sample must show no sign of cracking after the flexing. This is a rapid and convenient method for routine checking of rubber stocks and it is adaptable to the finished part in most cases. It is recognized, however, that this may not be the most sensitive method. A part may lose its functional properties at temperatures far above that at which it fails by this test. The use of a dyne tensiometer for evaluating the increasing modulus with decreasing temperature has recently been proposed.[10] This is a sensitive method which yields data that can be plotted to indicate the relation of stiffness to temperature.

One of the important factors to be borne in mind when considering the cold resistance of a synthetic rubber stock is whether resistance to any fluid is also required. Since many stocks owe their cold resistance to the plasticizers used in compounding, loss of this plasticizer during the aging will result in an impairment of its cold resistance. It is, therefore, necessary for most aircraft-engine applica-

[10] See "The Evaluation of Low-Temperature Properties of Several Synthetic Rubbers in Aviation Fuels Using the Dyne Tensiometer," by R. G. Chollar, G. I. Wilson, and G. K. Green. Paper presented at the Fall Meeting of the Division of Rubber Chemistry, American Chemical Society, New York, October, 1943.



■ Fig. 13 – Apparatus used for determining cold resistance of synthetic rubber stocks – A (upper): position during test; B (lower): position at conclusion of test

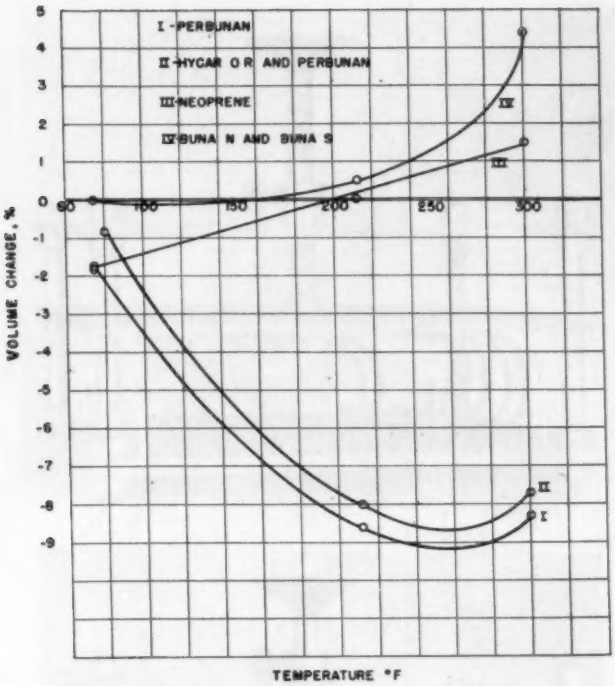
tions to measure the cold resistance of the synthetic rubber after fluid aging to ensure that the cold resistance will be maintained all during the service life of the part. Fortunately, the aromatic gasolines which often extract plasticizers generally swell the synthetic so that loss of plasticizer is compensated for by solvent plasticization.

Determination of Resistance to Various Oils – Testing the oil resistance of a synthetic rubber stock for aircraft-engine applications is done routinely by measuring the changes in volume, hardness, tensile strength, and elongation which take place during a 70-hr immersion of the sample in both aircraft-engine lubricating oil and rubber processing oil at 300 F. It has already been shown that the volume change of the stock will be a function of the aniline point of the oil used. The aircraft-engine lubricating oil

Table 5 – Changes Occurring in Four Synthetic Rubber Stocks During Oven Aging at 212 F

Stock	1 Day	2 Days	3 Days				1 Week			
	Weight Loss, %	Weight Loss, %	Weight Loss, %	Hardness Change *	Tensile Strength Change, %	Elongation Change, %	Weight Loss, %	Hardness Change *	Tensile Strength Change, %	Elongation Change, %
Neoprene	6.1	7.6	8.3	+ 9	+ 3.3	−50	8.85	+15	+ 9.5	−68
Hycar	3.0		6.6	+10	+12	−60	10.2	+14	+10	−69
Perbunan	2.65	2.9	3.3	+ 6	0	−25	3.95	+ 9	+ 5	−50
Buna N and Buna S	0.54	0.65	0.73	+ 5	0	−21	0.9	+ 5	+ 8	−29

* In Shore durometer "A" units.

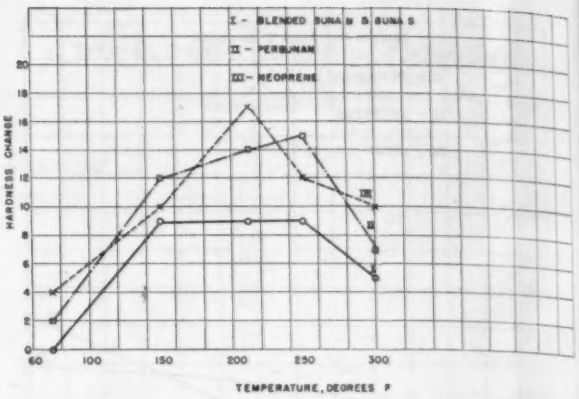


■ Fig. 14 – Effect of temperature upon volume changes of synthetic rubber stocks aged 70 hr in aircraft-engine lubricating oil

and the Circo light processing oil represent extremes in aniline points and, therefore, are expected to represent limits of volume change to be exhibited by the synthetic rubber part. The value of testing in the processing oil has often been questioned, since it has not been shown that oils with such a low aniline point will ever be encountered in service. This testing does, however, have three advantages: it fixes the upper limit for volume changes; it provides a type of accelerated aging test by which long-time service changes may be judged; and it presents a method of identifying the basic polymers used in the synthetic rubber stock, since the volume changes of the buna N stocks are characteristically lower than those for the polychloroprene types. The temperature of 300 F was chosen because it represents the upper limit of what is likely to be met in service by those parts required to be oil resistant.

An investigation of the effect of temperature upon the changes in typical synthetic rubber stocks when immersed in various oils was conducted to determine whether the routine testing procedures give an adequate picture of what may happen during the service life of a part which must be oil resistant. Various synthetic rubber compounds were aged in the oils described in Table 3 at room temperature, 100, 150, 212, 250, and 300 F for 70 hr.

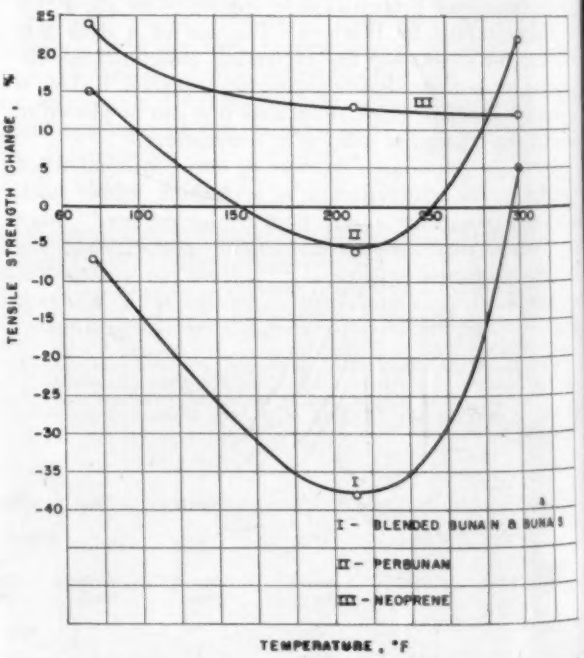
The effect of temperature upon the volume changes of these synthetic rubbers in the aircraft-engine lubricating oil is shown in Fig. 14. It is at once apparent that no generalizations can be made regarding the effects of temperature, since they vary with the type of stock tested. It is significant to note that a stock which shows a positive volume change at 300 F may shrink when tested at lower temperatures. Most oil seals require small positive changes in volume, and leakage may occur if the material shrinks appreciably. Therefore, if the material for such a seal is selected on the basis of volume changes at 300 F, but in service the temperature of the oil encountered is much lower, the part might not perform satisfactorily. To ensure



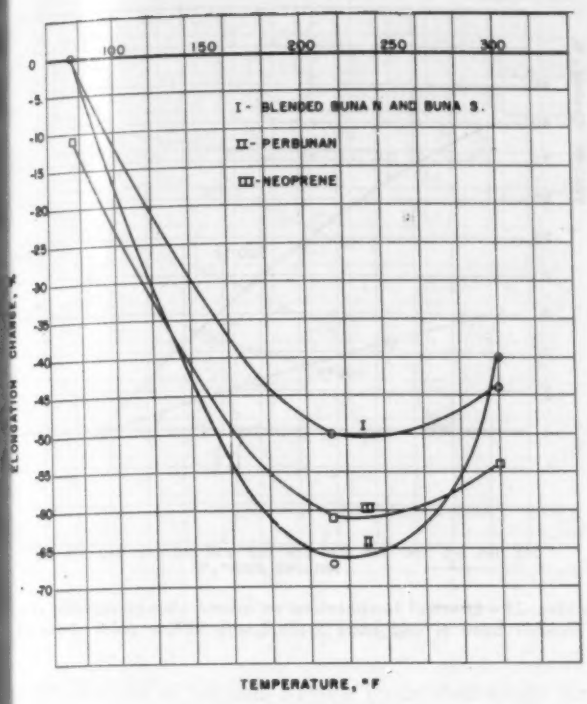
■ Fig. 15 – Effect of temperature on hardness changes of three synthetic rubber stocks at end of 70-hr immersion in aircraft-engine lubricating oil

adequate sealing, then, either an accurate estimate of the service temperature must be made and the material checked at that temperature or a stock must be chosen which shows a satisfactory volume change over the entire temperature range. It can also be seen from Fig. 14 that for those stocks which showed negative volume changes in the aircraft-engine lubricating oil, the temperature of maximum volume change was not at 300 F. If it is desired, therefore, to establish the limits of volume change in an oil, it may be necessary to use a testing temperature other than 300 F.

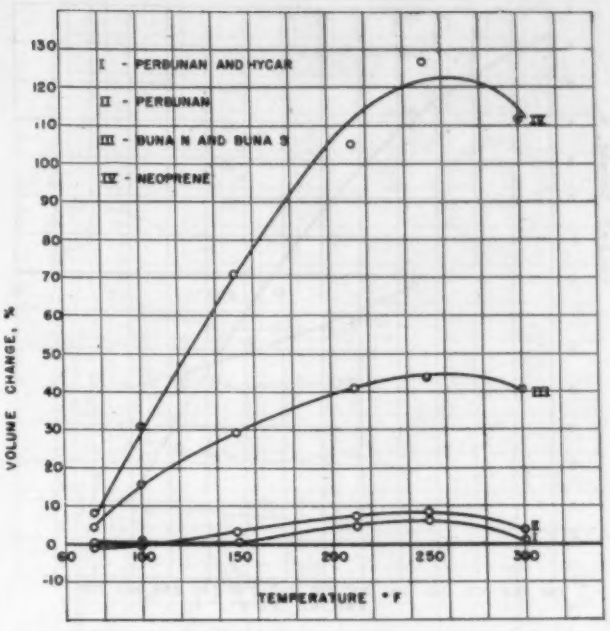
Fig. 15 illustrates the effect of temperature upon the changes in hardness occurring during 70-hr aging in the aircraft-engine oil. All three stocks hardened during this oil immersion but the relative amounts of hardening were a function of the temperature. The temperature of maximum hardness change for all of them occurred between 200 and 250 F rather than at 300 F. This suggests that for applications where the softness of the material affects the functional characteristics of the part, it might be more valuable to test at several elevated temperatures and at a



■ Fig. 16 – Effect of temperature on tensile strength changes of three synthetic rubber stocks after 70-hr immersion in aircraft-engine lubricating oil



■ Fig. 17 – Effect of temperature on changes in ultimate elongation of three synthetic rubber stocks at end of 70-hr immersion in aircraft-engine lubricating oil



■ Fig. 18 – Effect of temperature on volume changes of four synthetic rubber stocks aged 70 hr in rubber processing oil

temperature as close to the actual service conditions as possible.

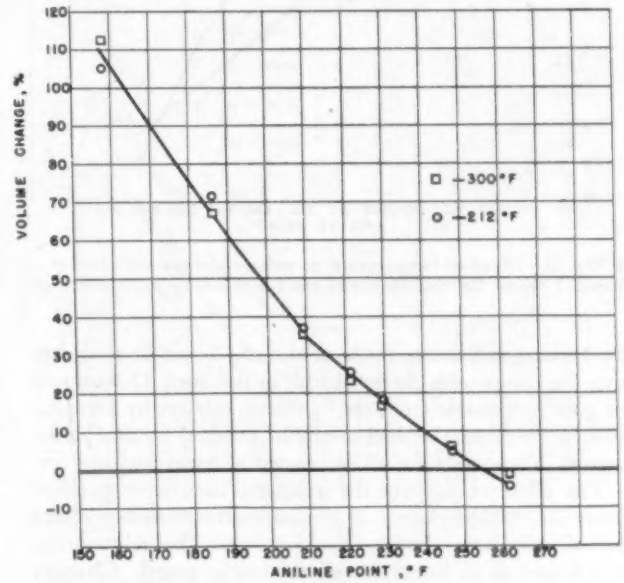
The effect of temperature upon the changes in tensile strength and ultimate elongation occurring during aircraft-engine lubricating oil aging for 70 hr is illustrated in Figs. 16 and 17. It is evident from these figures that testing at 300 F may not reveal the maximum changes that occur in these properties as a result of oil aging. Not only does the temperature influence these changes, but the manner of its effect varies with the stock tested, so that the comparative values of the stocks are a function of the temperature of testing.

Fig. 18 shows the effect of temperature upon the volume changes of typical synthetic rubber stocks when immersed in Circo light processing oil for 70 hr. These curves also illustrate that testing at 300 F may not reveal the maximum in the changes occurring. The comparative values of the stocks remain the same, however, regardless of the temperature of testing.

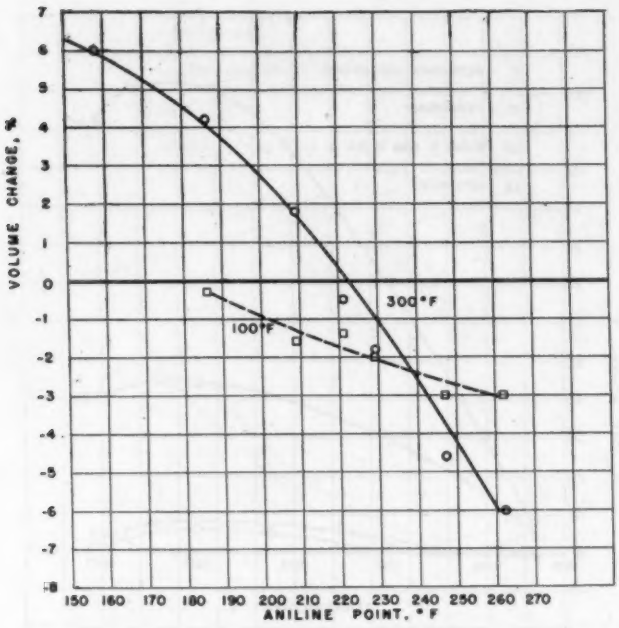
It must be emphasized that in all these studies of the effect of temperature upon the property changes occurring during oil aging, the determinations were made at the end of 70 hr. This immersion period was chosen because it is the one used routinely when testing for conformance to the Aeronautical Material Specifications. If, however, these changes have not reached equilibrium at the end of the 70-hr period, then the conclusions drawn might have to be revised according to the immersion period chosen for the testing.

The influence of temperature on the variation of volume change with the aniline point of the oil was studied for four of the stocks. The results are summarized in Figs. 19 to 22 inclusive. Except for the Neoprene stock, the temperature of testing changes the type of functional relation between the volume change and aniline point entirely. The slopes of the curves at the higher temperatures are generally steeper, indicating that the aniline point has a greater significance as far as volume changes are concerned at the higher temperatures. The accelerating effect of temperature upon the functional relationship is illustrated also in the curves relating hardness changes to aniline points of the oils. This is shown for the blended buna N and buna S stock in Fig. 23.

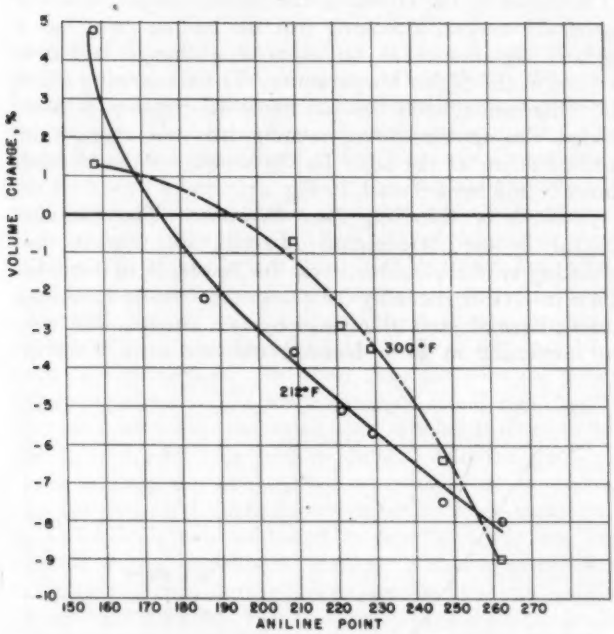
Methods of Checking Fuel Resistance – The tests included in the Aeronautical Material Specifications for checking synthetic rubber stocks for resistance to aromatic fuels involve determining the changes in volume, hardness, tensile strength, and ultimate elongation after 24- and 168-hr immersion in 40% blended aromatic fuel. Tests in



■ Fig. 19 – Effect of temperature on volume change variation of a Neoprene stock with aniline point of the oil



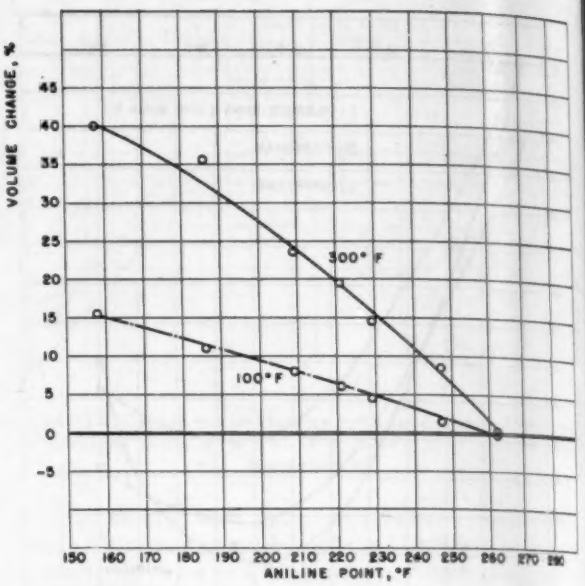
■ Fig. 20 – Effect of temperature on volume change variation of a Perbunan stock with aniline point of the oil



■ Fig. 21 – Effect of temperature on volume change variation of a blended Hycar OR and Perbunan stock with aniline point of the oil

65-octane gasoline are made at the end of 24 hr to ensure that the compounds do not shrink in this fuel. The amount of gum extractable from the synthetic rubbers by aromatic fuel, as well as the fuel-insoluble residue, is also determined. These tests are all conducted at room temperature.

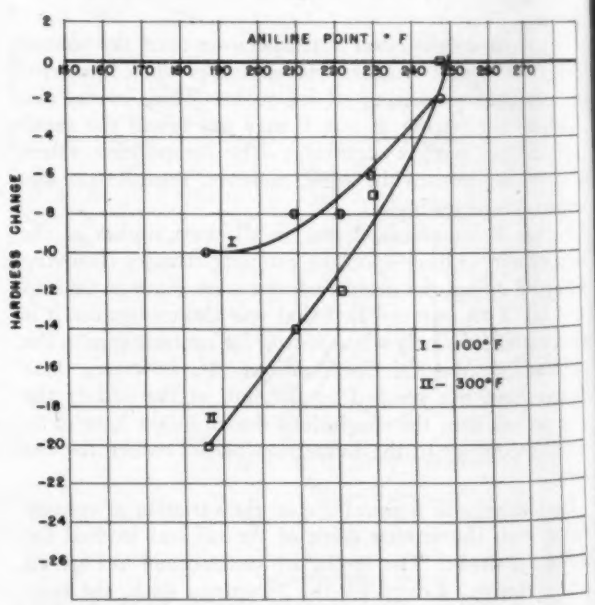
The effect of varying the aromatic content of gasoline upon the volume change of typical synthetic rubber stocks has already been shown (See Fig. 7). The relationship was found to be linear for all the stocks tested. Changes in other physical properties, however, are not as simply related to the aromatic content. After a week's immersion,



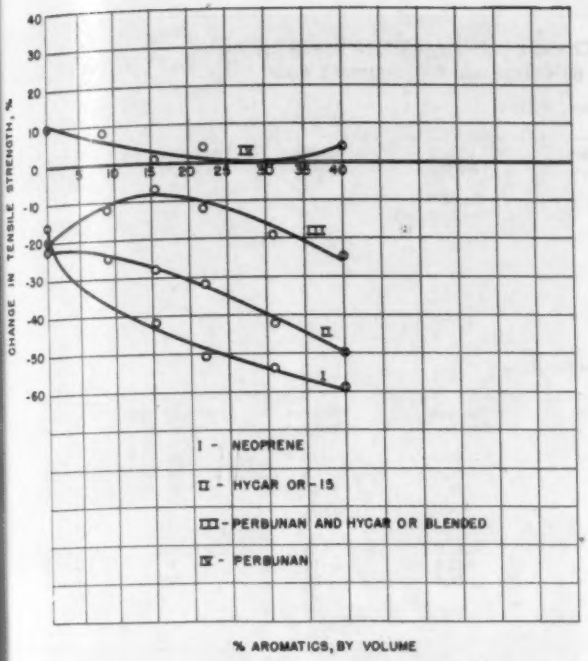
■ Fig. 22 – Effect of temperature on volume change variation of a blended buna N and buna S stock with aniline point of the oil

the compounds tested showed changes in tensile strength which varied in a more complicated manner with increasing percentage of aromatic constituents in the fuel. This is illustrated in Fig. 24, from which it is also evident that the type of function varies with the material tested.

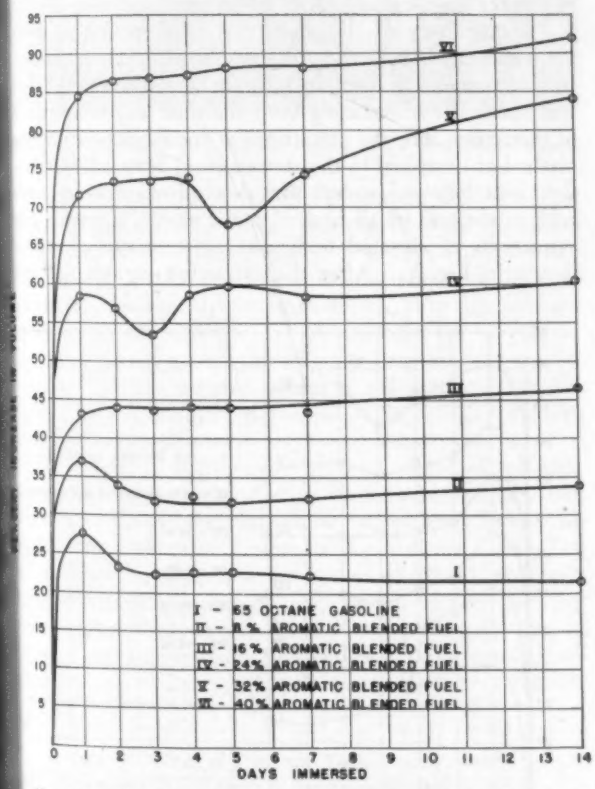
Daily determinations of the volume changes of five different synthetic rubber compounds immersed at room temperature in fuels varying from 0 to 40% in concentration of aromatic compounds were made. The results are summarized in Figs. 25 to 29, inclusive. With the exception of the Neoprene stock, which would not be used for applications requiring an aromatic fuel resistant material, it is evident from these curves that when testing in the 40% aromatic fuel, the specified 24- and 168-hr periods will give an adequate picture of the extremes of volume



■ Fig. 23 – Effect of temperature on hardness changes of a buna N and buna S stock in oils of varying aniline points

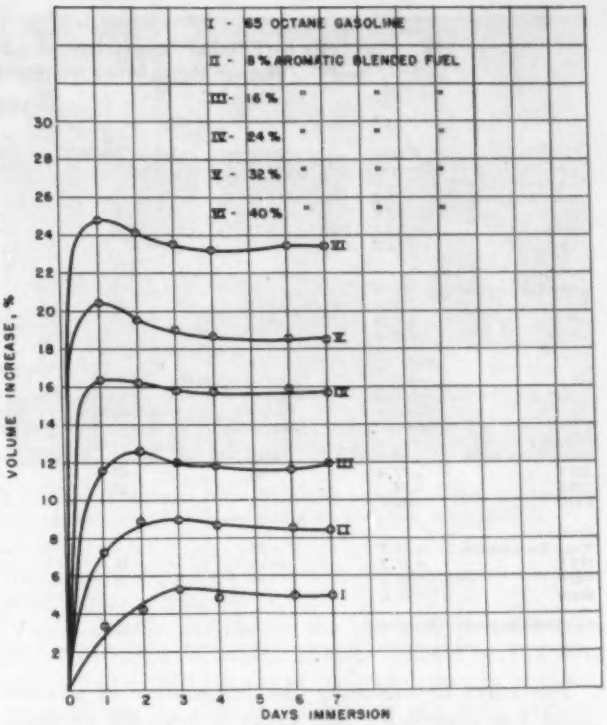


■ Fig. 24 – Effect of percentage aromatics on tensile strength changes of synthetic rubbers in fuels at end of one week's immersion at room temperature


I - 65 OCTANE GASOLINE
II - 8% AROMATIC BLENDED FUEL
III - 16% AROMATIC BLENDED FUEL
IV - 24% AROMATIC BLENDED FUEL
V - 32% AROMATIC BLENDED FUEL
VI - 40% AROMATIC BLENDED FUEL
DAYS IMMERSED

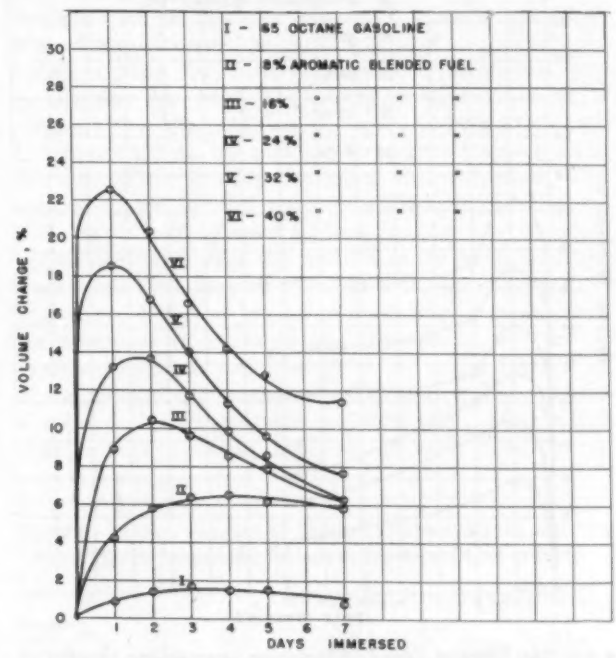
■ Fig. 25 – Effect of period of immersion upon volume changes of a Neoprene stock in fuels of varying aromatic content

changes likely to occur, since the maximum in the curves occurs at the end of 24 hr. In some of the fuels with lower concentrations of aromatics, however, the maximum volume change is often not reached before the end of 48 hr. Again excepting Neoprene, it appears that increasing the



■ Fig. 26 – Effect of period of immersion upon volume changes of a Perbunan stock in fuels of varying aromatic content

concentration of aromatic components hastens the time at which the maximum volume change will occur. For those stocks studied, checking the volume change in 65-octane gasoline at the end of 24 hr was an adequate guarantee against shrinkage. Since it is noted, however, that the volume change has in all cases decreased at the end of 168 hr, it appears that stocks which swell only slightly during the first 24 hr should be checked again at the end of a week to ensure that no shrinkage has occurred.



■ Fig. 27 – Effect of period of immersion upon volume changes of a Hycar OR stock in fuels of varying aromatic content

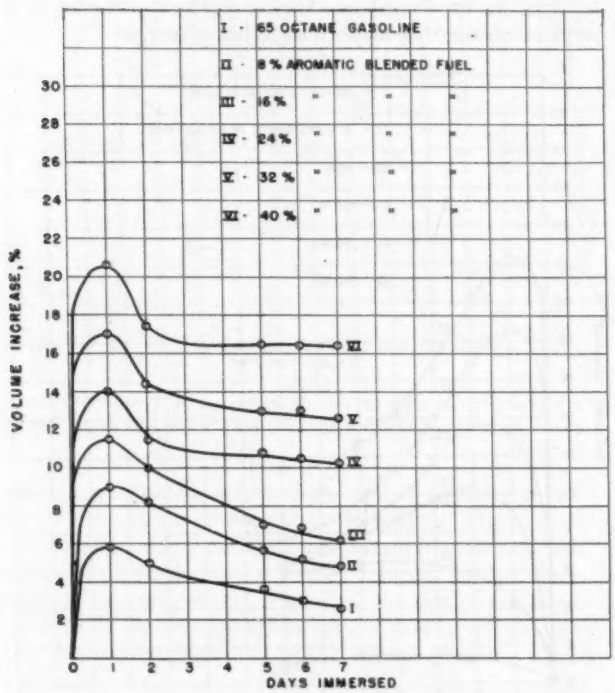
Table 6 – Effect of Temperature upon the Changes in Physical Properties of Synthetic Rubber Stocks When Immersed in 65-Octane and 40% Aromatic Fuel

	65-Octane Gasoline Aging							
	24 Hr				168 Hr			
	Volume Change, %	Hardness Change, °	Tensile Strength Change, %	Elongation Change, %	Volume Change, %	Hardness Change, °	Tensile Strength Change, %	Elongation Change, %
Neoprene:								
Room Temperature	+27.5	−10	−18	−23	+22.2	− 1	−16	− 8
115 F	+22.9	− 8	−24	−17	+22.0	0	−45	−24
140 F	+24.1	− 2	−29	−17	+23.8	− 4	−28.5	−23
Vapor	+23.5							
Perbunan:								
Room Temperature	+ 3.4	−11	− 1	0	+ 4.9	0	+10	−17
115 F	+ 2.8	− 5	− 7	0	+ 1.4	+ 1	+ 5	−17
140 F	+ 6.0	− 4	+ 3	0	+ 4.9	−11	− 1	0
Vapor	+ 9.4							
	40% Aromatic Fuel Aging							
	24 Hr				168 Hr			
	Volume Change, %	Hardness Change, °	Tensile Strength Change, %	Elongation Change, %	Volume Change, %	Hardness Change, °	Tensile Strength Change, %	Elongation Change, %
Neoprene:								
Room Temperature	+84.5	−19	−61	−54	+88	−16	−60	−46
115 F	+77.4	−16	−61	−54	+78.4	− 8	−59	−54
140 F	+74	−11	−57	−46	+86	−11	−68	−46
Vapor	+80							
Perbunan:								
Room Temperature	+24.7	−22	− 1	−17	+23.5	−10	+ 3.5	−33
115 F	+21.3	−18	−18	−17	+22.6	−10	− 8	−33
140 F	+24.3	−17	− 8	0	+27.9	−25	−32	−17
Vapor	+18.8							

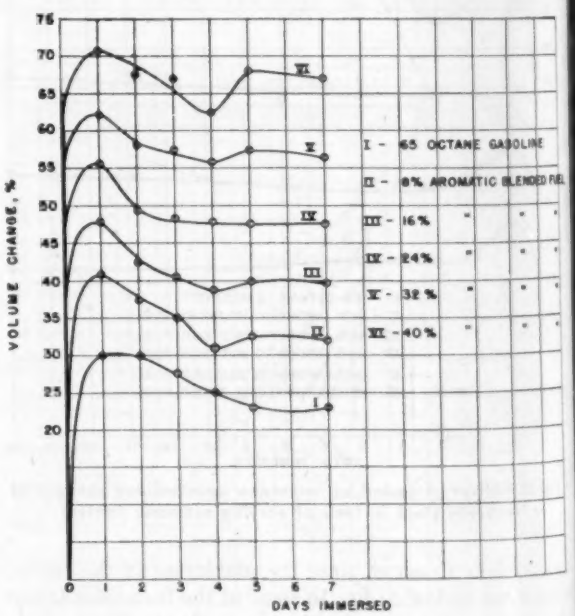
° In Shore durometer "A" units.

The effect of increasing the temperature of gasoline aging was studied for two stocks in both the 65-octane gasoline and in 40% aromatic blended fuel. The apparatus used in conducting these tests is illustrated in Fig. 30. Measurements of changes in volume, tensile strength, and ultimate elongation were made at the end of 24 and 168 hr. The effect of subjecting these stocks to only the vapors of these two fuels was also checked. The results of these studies are summarized in Table 6. It is evident from this table that temperature has no great nor consistent effect upon the changes occurring in the stocks during the gasoline aging. It may also be concluded that the routine checking at room temperature will give a sufficiently accurate picture of the performance of those parts which may encounter gasoline at higher temperatures during service.

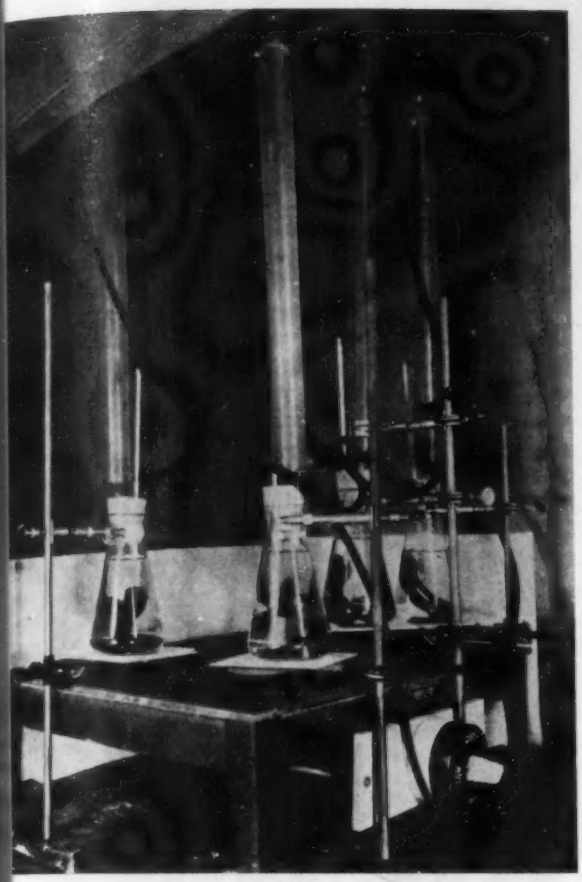
The adequacy of the 48-hr extraction period at room temperature for indicating the amount of gum extractable from a synthetic rubber stock by the aromatic (40%) fuel was checked by increasing both the time and temperature of extraction. For the extractions at room temperature, the method as described in the Aeronautical Material Specifications was followed, except that determinations were made both at the end of 48 hr and at the end of a week. The extractions at elevated temperatures were carried out as shown in Fig. 31. After the extraction periods (48 and



■ Fig. 28 – Effect of period of immersion upon volume changes of a blended Perbunan and Hycar OR stock in fuels of varying aromatic content



■ Fig. 29 – Effect of period of immersion upon volume changes of a blended buna N and buna S stock in fuels of varying aromatic content



■ Fig. 30 – Apparatus for gasoline aging at elevated temperatures

168 hr) the remainder of the procedure followed that described in the specification. The results of the extraction tests are summarized in Table 7. It is apparent from this table that at constant temperature the length of time of extraction has a relatively small effect. The temperature at which the extraction is carried out, however, may be an important factor in determining the amount of gum extracted. Neither variable causes a sufficiently great difference in the amount of fuel-insoluble residue to make that quantity significant.

The difference between the relative effects of temperature upon the amounts of gum extractable from the two stocks tested may be explained, in part at least, by the fact that different plasticizers were used in the two cases. The solubility of the one plasticizer in aromatic fuel is obviously a more critical function of temperature than is the solubility of the other. It may be concluded from the data in Table 7 that a 48-hr extraction period at room temperature does not necessarily reveal the amount of gum extractable

Table 7 – Effect of Time and Temperature upon the Aromatic Fuel Extract of Synthetic Rubber Stocks

Stock	48 Hr		168 Hr	
	Gum Content, %	Fuel-Insoluble Residue, %	Gum Content, %	Fuel-Insoluble Residue, %
	At Room Temperature			
Neoprene	8.1	0.03	8.15	0.03
Perbunan	3.0	0.035	3.4	0.07
	At Elevated Temperatures			
Neoprene	8.7	0.03	9.2	0.03
Perbunan	6.4	0.04	6.46	0.15

■ Fig. 31 – Apparatus for determining amount of gum extractable from synthetic rubber stocks at elevated temperatures

from a part, if gasoline at elevated temperatures is encountered during service.

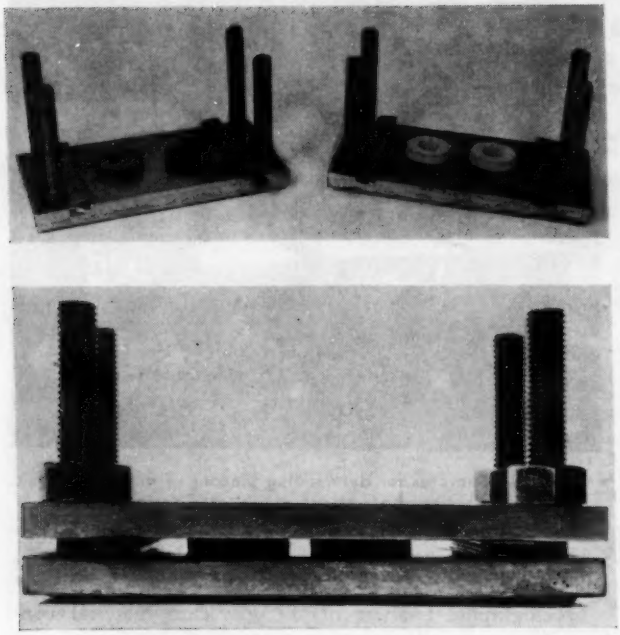
Spectographic analysis of the extracted material confirmed its essentially organic nature. A slight trace of zinc was also detected. Because of its organic nature, it is not expected that the gum in the gasoline will have harmful effects upon the performance of the fuel, if the quantities are kept relatively small.

Determination of Coolant Resistance – Tests for resistance of synthetic rubber stocks to coolants are checked routinely by immersing the stock in 97% ethylene glycol for 70 hr at 300 F. The adequacy of the 70-hr test period at this temperature for predicting the volume changes of the stocks was checked by determining the changes in volume both at the end of 70- and 168-hr immersion. The values obtained for the six stocks checked are summarized in Table 8, from which it is evident that for most synthetic rubber compounds, the 70-hr period is sufficient to indicate the extent of the volume change. For a Neoprene stock of the type used in this study, however, a longer period of immersion would need be used.

Compression Set Tests – It has been mentioned previously that for many aircraft-engine applications, it is necessary that the cold flow of the parts be a minimum. Laboratory testing for cold-flow properties involves determination of the compression set of the compound. The apparatus used for this purpose is illustrated in Fig. 32. The sample is compressed a given percentage of its initial height (this percentage is a function of the hardness of the stock) and then kept at an elevated temperature, usually 212 or 250 F for 70 hr. Its recovery after removal from the apparatus is measured and the amount of "set" is expressed either as a percentage of the original height or of the amount compressed. A stock with a low compression set is highly desirable for most seals. Hose which must be held by clamps should also be made from material

Table 8 – Volume Changes of Synthetic Rubber Stocks After 70- and 168-Hr Immersion Periods in Ethylene Glycol at 300 F

Stock	70 Hr Volume Change, %	One Week Volume Change, %
Neoprene	+20.9	+40.4
Hycar OR	– 6.2	– 4.5
Perbunan	+ 6.1	+ 5.0
Blended buna N and buna S	+ 7.5	+ 7.2
Blended Perbunan and Hycar	– 3.2	– 3.5



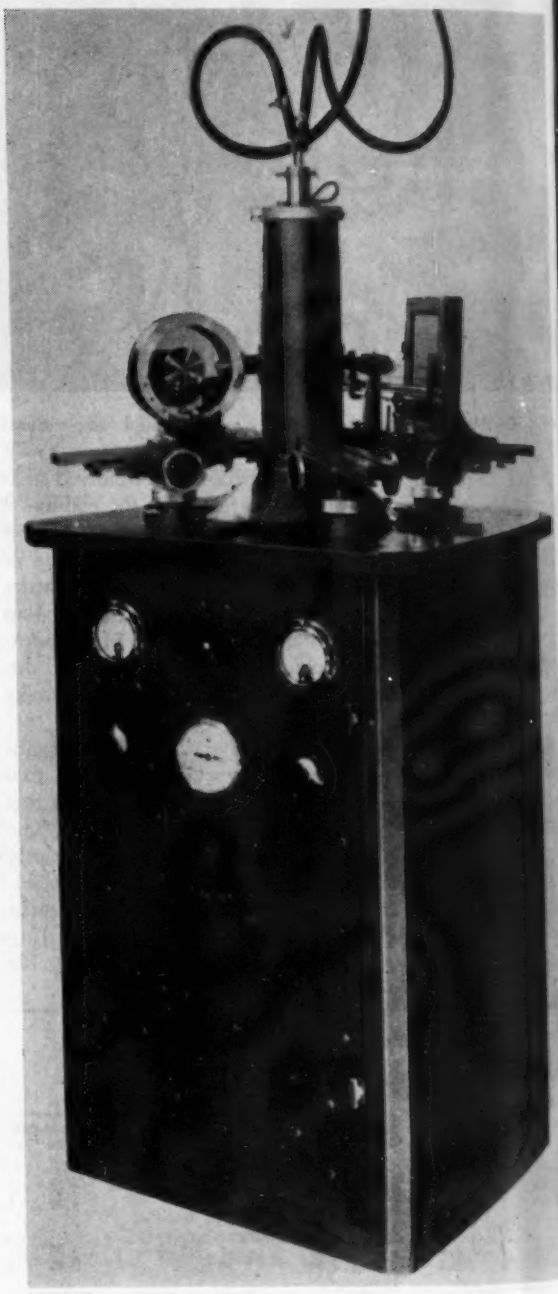
■ Fig. 32 – Apparatus used in determining compression set of synthetic rubber parts

with as low a compression set as possible to ensure satisfactory service performance. To obtain stocks with low compression values, not only must the type of polymer be carefully selected, but the compounding ingredients, temperature and time of cure must also be rigidly controlled. Regulating the curing time and temperature to obtain best compression set values often means sacrificing the original tensile strength and elongation properties. It is now recognized that for many aircraft applications, it is worth the sacrifice of such properties to obtain good cold-flow characteristics.

X-Ray Diffraction Analysis – A study was made of the usefulness of X-ray diffraction analysis in evaluating various synthetic rubber stocks for aircraft-engine applications. All of the testing which is included in the checking of synthetic rubbers by specification requirements is essentially physical in nature. No insight into the chemical composition or changes in chemical composition which may occur during various types of aging is given. It was expected that X-ray diffraction might be used as a chemical check to supplement the physical tests generally employed.

X-ray diffraction has been used to study natural rubber both as the crude product and in the vulcanized state. A general review of this work has been presented by Gehman.[11] Recently Field[12] has shown that it is possible to obtain a quantitative measure of the crystalline hydrocarbon present in stretched rubber samples by comparing the intensities of the diffraction spots and the amorphous halo appearing in the X-ray diagrams. He also showed that the physical properties and performance of vulcanized rubber are related to the amount of crystalline material formed upon stretching. Clark[13] discusses very fully the fundamentals of X-ray analysis of structure and includes descriptions of the experimental techniques. He has also reported[14, 15] studies of the effects of various compounding ingredients as indicated by X-ray diffraction analysis.

Sebrell and Dinsmore[16] and Sebrell[17] have discussed the use of X-ray diffraction for the study of synthetic rubbers. Since many of the synthetic rubbers do not show the crystalline fiber pattern which is characteristic of rubber upon stretching, they conclude that the structure of the synthetic rubbers is different from that of natural rubber and that



[11] See *Chemical Reviews*, Vol. 26, April, 1940, pp. 203-226: "The Contribution of X-Ray Research to the Knowledge of Rubber," by S. D. Gehman.

[12] See *Journal of Applied Physics*, Vol. 12, January, 1941, pp. 23-34: "An X-Ray Study of the Proportion of Crystalline and Amorphous Components in Stretched Rubber," by J. E. Field.

[13] See *Rubber Age*, Vol. 48, February, 1941, pp. 321-327: "X-Ray Studies of Rubber. Part I: Fundamentals," by George L. Clark.

[14] See *Rubber Age*, Vol. 49, July, 1941, pp. 255-257: "X-Ray Studies of Rubber. Part III: A Comparison of Two High Gum Compounds," by George L. Clark.

[15] See *Rubber Age*, Vol. 49, September, 1941, pp. 399-405: "X-Ray Studies of Rubber. Part IV: Finely Divided Crystalline Fillers," by George L. Clark and Robert L. LeTourneau.

[16] See SAE Transactions, Vol. 49, September, 1941, pp. 368-379: "Properties of Some Synthetic Rubbers," by L. B. Sebrell and R. P. Dinsmore.

[17] See *Industrial and Engineering Chemistry*, Vol. 35, July, 1943, pp. 736-750: "The Second Mile," by L. B. Sebrell.

they owe their rubber-like properties to the operation of a different mechanism from that of the natural product.

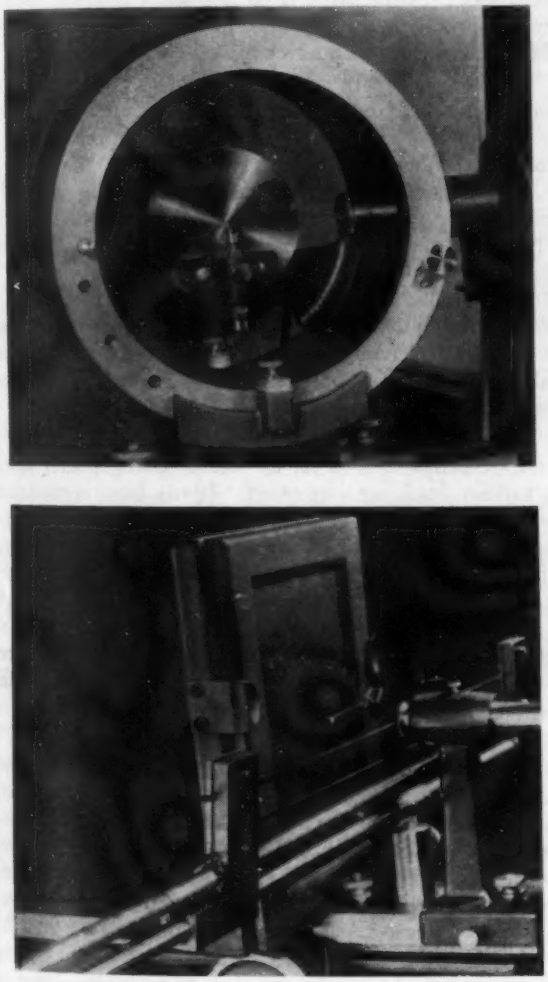
The X-ray diffraction apparatus used in the work reported here is pictured in Fig. 33. Employing cobalt radiation, forward reflection pictures of stretched and unstretched samples were taken. By means of the circular camera, powder pictures were obtained for identification of the crystalline fillers. Typical synthetic rubber stocks were studied as received and after various types of aging. Finished parts for use on the engine were also checked.

As is well known, the diffraction patterns of both natural and synthetic rubbers when unstretched are similar to that of a liquid, displaying an amorphous halo. When stretched, however, natural rubber, Neoprene, and butyl rubber exhibit a crystalline structure. The buna N and buna S stocks remain amorphous even upon stretching. It is thus possible to make some identification of the basic polymers used in the synthetic rubber stocks through a study of their X-ray diffraction patterns taken when the compounds are stretched.

By means of the X-ray diffraction analysis, the crystalline constituents used in compounding the synthetic rubber stock may also be identified. This is illustrated in Fig. 34, in which A represents pure zinc oxide and B a synthetic rubber stock in which zinc oxide has obviously been used as a compounding ingredient. It is, of course, possible to identify the material used in the stock even without the use of a comparative picture. If the distances of the lines from the central spot are measured, the interplanar spacings can be calculated. Checking of these values with tables listing the values of various compounds identifies the material present. When more than one compound is identified, it is also possible to compute the relative percentages of each which are present.

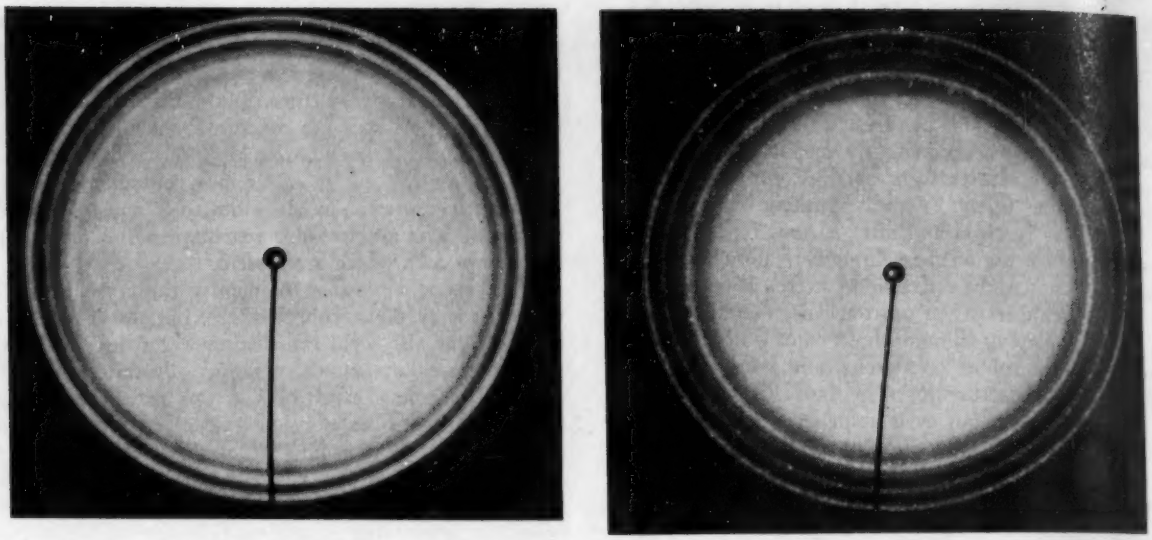
It is not only possible to identify the crystalline fillers used in the synthetic rubber stocks, but the X-ray diffraction patterns also yield information regarding the particle size of these compounds. Fig. 35 illustrates this. The smooth continuous bands of Fig. 35A reveal that the particle size of the filler is below 10^{-3} cm, while the spotted arcs of Fig. 35B indicate that in this stock some of the crystalline filler present has a much larger particle size. Clark[15] working with natural rubber and commenting upon the importance of the particle size of fillers points out that for a given weight, the smaller the particle size the greater the active surface for bonding to the rubber matrix. He concludes that this would be expected to exert a considerable influence on the behavior of the filler in rubber deformation and hence on the physical properties of the rubber mix. Further study and experimentation are required before the significance of the particle size of fillers in synthetic rubbers can be understood. It is logical to expect, however, that the physical properties and, in turn, the service performance of a stock will be influenced by this variable.

Information may also be gained from the X-ray diffraction pattern regarding the behavior of the crystalline compounding ingredients when the stock is stretched. In Fig. 36 the preferred orientation of the filler is evident, that is, the intensities of the arcs are no longer uniform, but exhibit regions of maximum and minimum brightness. This phenomenon indicates that, during the stretching process, the particles are rotated and aligned in a preferred direction with respect to the direction of stretching. Clark[15] explained similar orientation of zinc oxide in stretched rubber stocks on the basis of a powerful bond existing between the zinc oxide grains and the rubber, which prevents the rubber from tearing away from the oxide surfaces during stretching and tends to pull all the grains into parallel alignment. It is interesting to note in Fig. 36 that the orientation of the filler is apparent even though there is no indication of a fiber pattern for the synthetic rubber.



■ Fig. 33 – X-ray diffraction apparatus used in studying various synthetic rubber stocks – A (left): entire apparatus; B (upper): powder camera; C (lower): forward transmission technique for stretched parts

■ Fig. 34 – X-ray diffraction powder pictures – A (left): zinc oxide; B (right): synthetic rubber stock containing zinc oxide

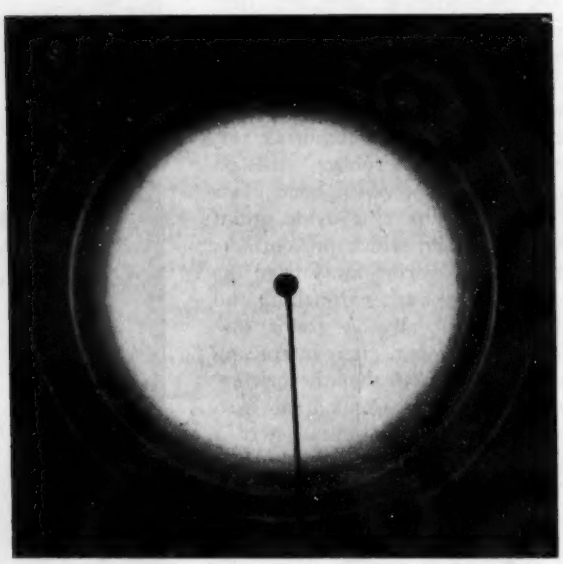


■ Fig. 35 – X-ray diffraction forward transmission pictures – A (left): synthetic rubber stock with small-grained filler; B (right): synthetic rubber stock with larger-grained filler

It may be possible, therefore, by X-ray diffraction analysis to check upon various fundamental differences in synthetic rubber stocks which would not be apparent by physical methods of testing. It is not suggested as a method for routine testing, but only as a tool which may be used to determine the reason for differences in service performance of parts which, according to physical testing, should be the same.

X-ray diffraction patterns of commercial synthetic rubber stocks made before and after aging in aromatic blended fuel, aircraft-engine lubricating oil, and processing oil, and at elevated temperatures yielded no information regarding the type of changes occurring either in the synthetic rubber or in the fillers as a result of the aging.

Adaptation of Standard Tests to Finished Parts – The testing, both routine and special, which has been described to select a synthetic rubber stock suitable for a particular engine application has been done mainly on standard-sized specimens prepared from laboratory samples. Before final approval of the material, however, the properties of the finished parts themselves must be laboratory tested.



■ Fig. 36 – Forward transmission X-ray diffraction pattern of a stretched synthetic rubber compound showing preferred orientation of the crystalline filler

Each finished part presents its own problem as far as the necessary adaptation of the standard testing method is concerned. The tensile strength and ultimate elongation values for finished parts which are in the form of thin rings are most conveniently obtained by using the special jaws on the tensile tester as illustrated in Fig. 5B. The Shore durometer, as mentioned previously, is adaptable to the finished part, although for some parts it is difficult to obtain reproducible results. Parts which are not too thin and whose cross-sections are uniform are used in the form received for determinations of compression set values. It is necessary only to vary the height of the spacers used according to the thickness of the part.

There has been much discussion regarding the importance of sample size as far as volume changes after aging in various fluids are concerned. It has been argued that the volume changes of very thin sections from finished parts cannot be expected to compare with those obtained when regular laboratory samples are used. Juve and Garvey[5], for example, have reported on the volume changes of several Hycar OR stocks in Univis 40. According to their results, varying the size of the specimen from 1 x 2 x 0.080 in. to ½ x ½ x ⅜ in. caused differences both in the magnitude of the volume change and in the rate of change.

It is admitted that great differences in the ratios of area to volume of the specimens used for volume determinations will cause variations in the volume changes noted, these differences varying with the time, temperature, and fluid used for the immersion. The variations in specimen size between standard laboratory samples and samples obtainable from typical finished parts, however, are generally not great enough to cause serious differences in the volume changes noted. Using the same apparatus as employed in the routine methods for testing, a study of the effect of specimen size upon volume change was made. The finished part, a small grommet, and a standard laboratory

sample were aged for 70 hr in aircraft-engine oil at 212 F. The volume changes of these samples (each done in quadruplicate) are listed below:

Volume Change, %	
Laboratory Sample	Grommet
−1.4	−1.6
−2.0	−0.5
−1.6	−1.2
−1.4	−0.9
Average −1.6	−1.05

It can be seen from these values that the difference in average values is well within the experimental error involved in routine testing.

There are, however, normal and expected differences between a stock compounded and cured in a laboratory and one which is obtained in large quantities from a production run. Variations inherent in synthetic rubber processing techniques and materials tend to produce deviations between the values obtained with purely laboratory compounds and those with finished parts which are made from production-run stock. That even with all these natural variations it is possible to use the same methods of laboratory testing and the same specification values for the finished parts as for the laboratory samples is illustrated in Table 9. It is evident from the data in this table that, in so far as it is possible to test, the limits established for laboratory samples are applicable to finished parts of various sizes.

Occasionally, it is found necessary for some applications to devise various tests which will simulate service conditions more closely than those outlined in the specification. For example, for a certain important oil seal, the effect of stretching on the changes in physical properties had to be determined. Accordingly, seals were stretched to the same degree as in their service applications and aged, along with unstretched seals, in aircraft-engine lubricating oil for 70 hr at 300 F. The resulting changes in tensile strength and ultimate elongation are listed below:

	Unstretched	Stretched
Tensile strength change, %	−52	−40
Elongation change, %	−46	−36

These data indicate that for routine testing, a sufficient indication of serviceability can be obtained from checking the unstretched rings. Tests were also made on these seals to determine whether stretching had any influence upon the cold resistance of the part. None could be demonstrated. Numerous other examples might be cited of special tests for particular applications, but the value of such tests lies in their very specialized nature and hence their general usefulness is limited.

The final check upon the suitability of a synthetic rubber compound selected for an aircraft-engine application is, of course, its performance on the engine. After it has been demonstrated through engine testing that the material chosen for the part is satisfactory, further laboratory checking ensures that the synthetic rubber in the parts received continues to meet the requirements of the specification.

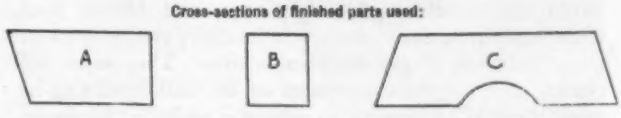
Summary

An outline of the procedure involved in selecting a suitable synthetic rubber for aircraft-engine applications has been presented. This procedure was divided into two main classifications: the application analysis, and the laboratory testing of synthetic rubber stocks and finished parts.

Table 9 – Comparison of Values Obtained on Laboratory Samples and on Various Finished Parts When Tested According to Specification

Tests	AMS-3213	Laboratory Samples	Finished Part (A)	Finished Part (B)	Finished Part (C)
I. Properties as received:					
Hardness	80 ±5	75	75	75	77
Tensile strength, psi	1000, min.	1840	2050	2180	2000
Elongation, %	150, min.	250	200	175	200
II. Fuel aging, 65-octane 24-hr:					
Volume change, %	No shrinkage	+5	+3	+3	+3
Aromatic fuel 24-hr:					
Volume change, %	0 to +25	+25	+29	+25	+28
Hardness change	0 to −15	−14	−10	−9	−13
Tensile strength change, %	−50 max.	−35	−48	−48	−50
Elongation change, %	−50 max.	−20	−50	−29	−50
168-hr:					
Further volume change, %	+8 max.	−2	−8	−3	−5
Hardness change	0 to −15	−14	−10	−9	−15
Tensile strength change, %	−50 max.	−48	−47	−33	−46
Elongation change, %	−50 max.	−40	−25	−48	−29
Aromatic fuel extraction:					
Gum content, %	10 max.	9	9	9	9
Fuel-insoluble residue, %	0.5 max.	0.07	0.03	0.05	0.04
Oven aging:					
Bend test	No cracks	No cracks	No cracks	No cracks	No cracks
Compression set, %	50 max.	35	No test possible	41	No test possible
Cold test, −40 F	No cracks	No cracks	No cracks	No cracks	No cracks

Cross-sections of finished parts used:



It was pointed out that analyzing the requirements of the application involves consideration of various factors which must be examined not only from the point of view of present operating conditions but also as they may be influenced by the least favorable conditions which might be encountered in service. The necessary general physical properties and the requirements for resistance to high and low temperatures must be determined for each application and a specification covering these requirements selected. The importance of proper design was emphasized, especially for parts such as seals.

Laboratory testing of synthetic rubber stocks and finished parts was discussed. This included descriptions both of routine testing of the material for conformance with specification and of special tests designed to supplement the information available through the routine checking.

The special laboratory testing was carried out on synthetic rubber stocks typical of those approved for use on the Ranger engines. The following conclusions are based upon the data presented:

1. Synthetic rubber vulcanizates differ widely both in the percentages and in the rates at which they lose weight during oven aging for one week at 212 F.

2. The changes in physical properties of synthetic rubber stocks which occur during oil aging are a function both of the aniline point of the oil and of the temperature of testing. The functional relationships with temperature are not linear, but exhibit maxima.

3. The volume changes of synthetic rubber stocks are a

concluded on page 280

OPERATING

"Dog power" used by author in Alaska on field studies of fuel and lubrication problems in winter operation of motorized equipment

DURING the past winter, the Standard Oil Co. of Calif. sent the author to Alaska to make an investigation of fuel and lubrication problems in internal-combustion-engine-powered equipment. The investigations included diesel- and gasoline-powered trucks, tractors, power units, passenger cars, and aircraft, operated in Nome, Anchorage, Fairbanks, Whitehorse, and on the Alaska Highway. It was learned that considerable difficulty was experienced with some petroleum products under war conditions between the months of October, 1942, and March, 1943, when minimum atmospheric temperatures ranged between —30 and —79 F for weeks at a time. This paper will discuss some of the experiences of the author during his work there in attempting to solve the problems of operating automotive equipment at —50 F.

In addition to the low temperatures involved, operating automotive equipment in Alaska is quite a problem due to the remote location and lack of facilities for transporting fuels, lubricants, and repair parts. In fact, a person working in that location has to be a frontiersman, a mechanic, and an operator, all in one. The newspapers tell us that last year was the most severe winter that the North country had experienced in almost 30 years. It was particularly serious to us Americans as a large amount of construction work had to be carried through regardless of weather conditions. Due to the fact that this was the first time that such a concerted effort had been made to operate under these conditions, operators were caught without previous experience and suffered many handicaps. When these difficulties developed, many persons thought that it was simply necessary to ask the advice of some sourdough on the proper method of operating internal-combustion engines, but it was soon learned that they had followed the practice of shutting down most of their equipment during the winter and going back to the States to enjoy themselves. This, combined with the fact that you could place all of the normal inhabitants of Alaska, including the Eskimos, in the University of California stadium and still have a considerable number of seats left, indicates that there was not a large number of people from whom to obtain advice. The Army engineers, constructors, and civilian operators deserve considerable credit for one of the largest pioneering feats in our history. It also required some pioneering on our part in developing the subzero fuels and lubricants which will be described below.

[This paper was presented at a meeting of the Northern California Section of the SAE, San Francisco, Dec. 13, 1943.]

■ ■

THE operation of automotive equipment at —30 to —79 F brought to light a variety of fuels and lubricants problems.

To investigate some of these difficulties, Mr. Goodale made a trip to Alaska. This paper discusses some of the experiences he had there in attempting to solve the problems of operating internal-combustion engines at —50 F.

■

THE AUTHOR: R. WAYNE GOODALE (M '35) is supervising engineer in charge of field research and development of fuels and lubricants for Standard Oil Co. of Calif. He started with that company in 1929 as an engineer in the producing department. For three years he was group leader for Pacific Coast companies participating in API-SAE road tests on detonation and vapor lock. Subsequently, he expanded to field testing of fuels and lubricants for airplanes. Last year he was a consultant to the Engineer Board at Fort Belvoir, Va., on cold weather testing of fuels and lubricants.

■ Diesel Fuels

In view of practices followed in civilian life, large quantities of fuels supplied to Alaska for use in diesel engines, pot-type heaters and burners, had a pour point of only 0 F. Due to the low temperatures experienced on construction jobs in the inland and northern regions, serious difficulty was experienced during the winter months. Briefly, the heater and burner problems involved either the flooding of stoves that resulted in serious fires, or the starvation of stoves that permitted the temperature of living quarters and supply houses to fall to extremely low temperatures and resulted in much damage to canned goods. The high pour point fuel was also unsatisfactory in all types of diesel engines due to the plugging of screens and filters with wax. It was not uncommon to experience a complete failure of fuel flow between the tank and transfer pump of an engine. Auxiliary heat from torches, fires, or storage in heated enclosures, and the idling of equipment 16 hr to permit its use for 8 hr, were practiced throughout the territory. Frequently, engines would stop during the idling period due to the formation of wax in the fuel line, and from 2 to 4 hr were required to start them again. One organization that operated all types of equipment along the Alaska Highway between Fairbanks and Whitehorse obtained stove oil throughout the winter from the local supplier and operated engines and heaters without difficulty. The success of these operations soon became known, and by the end of the season, stove oil was being purchased in large quantities at Anchorage, Whitehorse, and Fair-

AUTOMOTIVE EQUIPMENT at -50F

by R. WAYNE GOODALE
Standard Oil Co. of Calif.

banks, and was found to be entirely satisfactory. It is interesting to note that five makes of diesel engines: namely, Caterpillar, General Motors, Cummings, International, and Buda were operated successfully with this fuel. We have determined that at the temperatures involved cetane number is of minor importance and that the most important single requirement of a diesel fuel supplied to Alaska is low pour point. Arrangements have therefore been made to supply a fuel this winter of adequate cetane number and with a pour point not above —40 F.

Gasoline

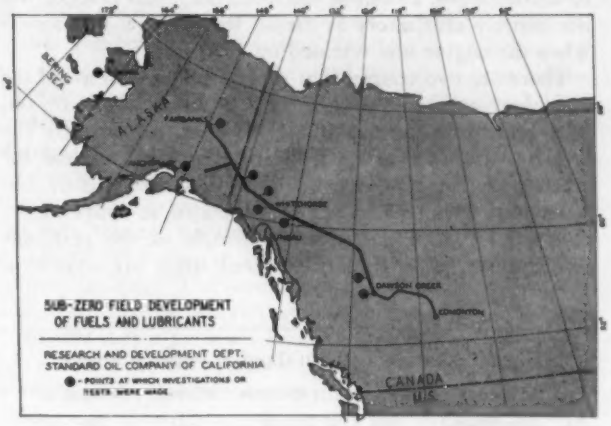
Gasoline is such a light product that there were no problems such as pour point that confronted its use at low temperatures. However, any gasoline has to meet conflicting requirements in that it has to be volatile enough to permit starting but involatile enough to prevent vapor locking. With these limitations the gasoline was found satisfactory if, at extremely low temperature, auxiliary heat or starting aids were used to start the engines.

Standard Ban-Ice

A common problem that existed in both gasoline and diesel fuel systems was ice that formed in the fuel tanks. This was due to the fact that fuels will dissolve about 0.005% water. Even though the fuel may be dry when it leaves the refinery, this amount of water will frequently be absorbed within a short time from condensation or laxity in handling. The ice frequently plugged the fuel lines, filters, or, in many cases, accumulated in the transfer pump of diesel engines so that when the engine was started cold, the safety pin of the pump shaft was sheared off. To overcome this difficulty, a product has been developed known as Standard ban-ice that has been distributed throughout the North country, for use this winter. This material, when used in the proportions of a fraction of 1%, is effective in preventing the freezing of water down to temperature of —50 to —60 F. Fig. 1 shows the proportions required for lowering the freezing point of the entrained water to the desired temperature. This fluid is added to the gasoline or diesel fuel in the tanks of vehicles at the time they are filled. As a precaution it should be mentioned that separated water should be drained off from fuel supplies.

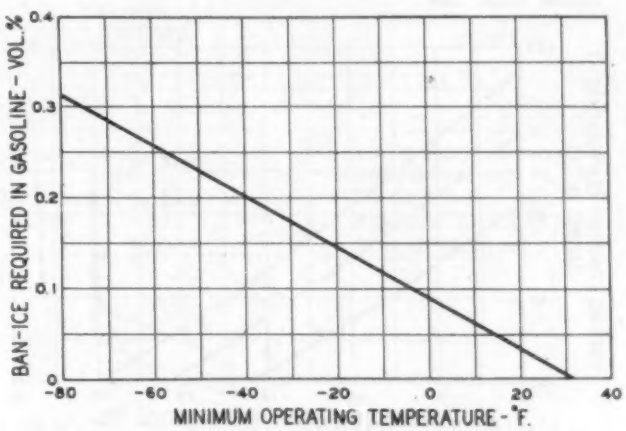
Crankcase Oils

One petroleum product that caused considerable trouble in starting engines was the crankcase oil. With the present



military activity in the North, where several thousand diesel- and gasoline-powered units must be operated the year round, the inconvenience and expense of keeping engines warm or the loss of time in heating them are not acceptable. The difficulty with the crankcase oil can be attributed to two characteristics; first, too high a viscosity to permit cranking of the engine, and second, too high a pour point to permit fluid flow at the temperatures involved. For this reason, at atmospheric temperatures between 0 and —60 F, the following practices were resorted to by many operators:

1. Engines were idled continuously during the periods when not in use. In many instances, this resulted in 16 hr of idling each day. Now that a starting fluid is available,



■ Fig. 1 — Proportion of Standard ban-ice required to prevent freezing of water in gasoline-water-ban-ice mixtures

the waste resulting from this practice is directly chargeable to the crankcase oil.

2. The oil was drained when the engine was shut down, heated, and replaced in the crankcase the next morning.

3. The crankcases were heated with a fire pot, or an open fire was built under the crankcase. Many units were destroyed by fire through this practice.

4. The units were covered with large tarpaulins and heaters placed underneath during the shutdown period. This was time-consuming and required portable equipment.

5. The equipment was placed in heated storage. Facilities of this type were available for only a limited number of vehicles.

6. The crankcase oil was diluted with various light oils or kerosene but, as will be discussed later, this practice was not entirely satisfactory as the oil was too low in viscosity when the engine was warmed up again.

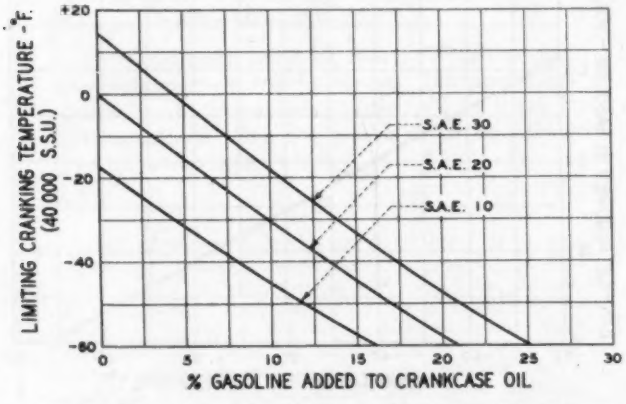
There are two courses that can be followed to avoid the difficulties experienced; namely, dilute the SAE 10 oils with gasoline or develop an oil of lower viscosity than the SAE 10. In order to select the most suitable method for overcoming the crankcase oil difficulties, temperature histories were obtained on vehicles operated at temperatures as low as −44 F. A typical example of the pertinent temperatures[1] of a gasoline-powered truck are shown in Table 1.

Table 1 – Temperatures in Gasoline-Powered Trucks

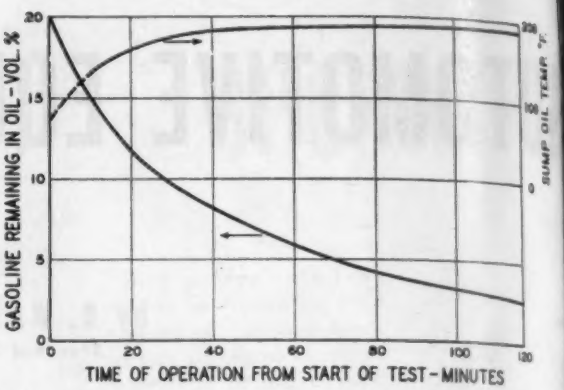
	Maximum	Minimum	Average
Atmosphere, F	−12	−44	−29
Jacket, F	170	150	159
Crankcase, F	174	131	155
Transmission, F	155	80	102
Transfer Case, F	142	103	123
Front Differential, F	43	5	15
Forward Rear Differential, F	63	39	53

It will be noted that the coolant and crankcase oil attained levels approaching those experienced in service at high atmospheric temperatures due to the thermostatic control of the jacket. For this reason, it was decided that the use of permanent diluents or an oil lighter in viscosity than SAE 10 would not provide adequate protection for the engine under heavy-duty operating conditions. It was therefore recommended that the practice as followed by military aircraft of diluting engine oils with gasoline prior to shutting down be followed by ground equipment also. This practice requires less diluent to reduce the viscosity and pour point to the desired values and has the advantage in that the gasoline will evaporate after starting and the engine will operate on the original grade of lubricant after it has warmed up. Also, gasoline is readily available and the type used is not critical. See Fig. 2. To determine the length of time required to evaporate the gasoline from an 80-20% mixture in an engine crankcase, tests were conducted in the laboratory with a Buick engine. Fig. 3 shows that in 30 min, half of the gasoline was evaporated, and although the rate of evaporation was slower following that period, only 3% was left after 100 min. This amount is considerably lower than the average amount of dilution found in passenger-car crankcases under normal operation in cold climates. Experience has shown that the gasoline may be poured into the crankcase through the oil fill pipe just prior to shutting the engine down at night, and after the engine has idled for 10 or 15 min, the oil and fuel will be mixed sufficiently to provide easy starting the next morning. To prevent excessive evaporation of the gasoline the engine should be idled a few minutes to permit the oil to cool before adding the diluent.

[1] Data obtained by J. Gardner Lewis, GMC Truck & Coach Division, General Motors Corp.



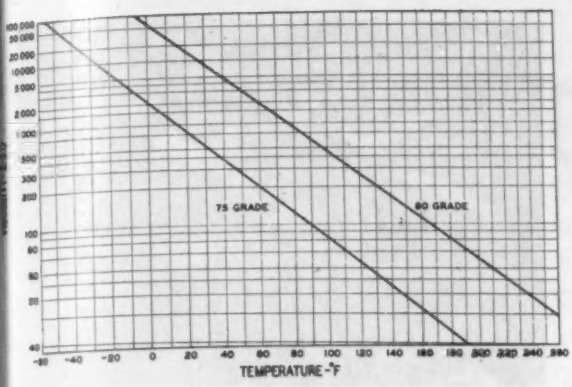
■ Fig. 2 – Effect of gasoline dilution on limiting cranking temperatures



■ Fig. 3 – Evaporation of gasoline from 80% 10W oil and 20% gasoline – 1941 Buick engine

■ Gear Lubricants

Several methods were utilized in Alaska during the winter months to protect the engines of automotive vehicles and permit relatively easy starting; however, no device was utilized to keep the gear cases warm except to provide warm storage. For this reason, it was necessary to apply torches or fires to gear cases, build fires under them, or tow the trucks, even after the engine had been started in order to loosen the gear lubricants sufficiently to permit driving the vehicles. Towing was the most common practice but resulted in many failures of drive shafts and axles due to the excessive torque produced by pulling the vehicles while the wheels were sliding. Considerable test work was done using gear lubricants of lighter viscosities than SAE 80 and with diluents added to the SAE 80 grade. With gear lubricants of this type, it was found that the viscosity limit of 100,000 SSU previously established in laboratory tests was an adequate criterion for satisfactory shifting of automotive vehicles. It will be recalled from the temperature history shown in Table 1 that the temperatures of gear cases of a typical truck varied from 5 to 155 F. This indicated that a lighter product than the SAE 80 could be utilized and still provide adequate protection



■ Fig. 4 – Viscosity-temperature characteristics of 75- and 80-grade gear lubricants

at the maximum temperatures of gear cases operated in subzero atmospheric conditions.

As a result of the above investigations, a gear lubricant lighter than the SAE 80 but one that would have extreme-pressure characteristics was developed and is now distributed throughout the North. Its physical characteristics are as given in Table 2.

Table 2 – Hypoid Lubricant – Subzero Type

Viscosity at 100 F, SSU	81
Temperature at which Viscosity is 100,000 SSU, F	−55
Pour Point, F	−55
Channel Point, F	−65
Load Carrying Capacity, SAE Machine, 1000 rpm	300
Foaming	None
Turbidity	None

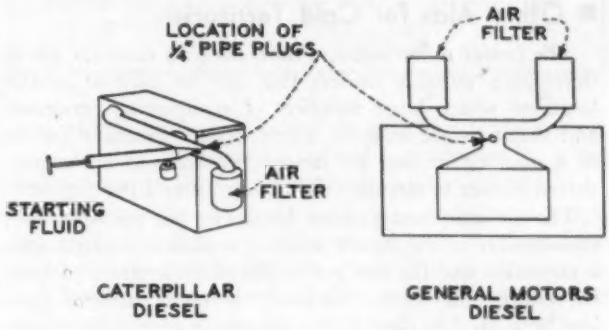
A comparison of the viscosity-temperature characteristics of this lubricant with the familiar 80 grade is shown in Fig. 4.

■ Chassis and Wheel Bearing Grease

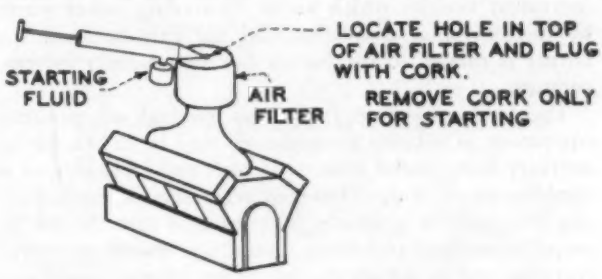
The wheel bearings and the chassis of vehicles experienced the same problem of being inadequately warmed as did the gear cases. No device was utilized to keep these points warm except to provide warm storage. A grease for use under these conditions must be sufficiently plastic to permit its use in grease guns at extremely low temperatures, as well as to provide adequate lubrication. The experience of many operators showed that the grade No. 0 grease generally used in that territory was not satisfactory under extremely cold conditions. A No. 00 grease was therefore developed that had outstanding low-temperature plasticity characteristics and sufficient extreme-pressure properties to make it an all-purpose grease for use in wheel bearings, universal joints, track rollers and idlers, spring shackles, drag links, king pins, and other chassis bearings on passenger cars, trucks, tractors, and special mechanized equipment.

■ Starting Fluid

As mentioned previously in this paper, starting diesel equipment below 20 F and gasoline equipment below zero is very difficult. Data have been submitted in numerous reports that show there is a relationship between the temperature at which diesel engines can be started and the cetane number of the diesel fuel. For example, Porter[2] shows that the minimum temperature for satisfactory starting with a 50 cetane number fuel is approximately 28 F, while with a 40 cetane number fuel it is approximately 35 F. These data show that at temperatures below approximately 20 F the difference in cetane number is not important because even with a high cetane fuel considerable time is consumed in applying heat or continuous cranking before starting can be accomplished. As the result of experimental work conducted in Alaska, a starting fluid which we call Standard starting fluid, was developed that permits the starting of automotive-type diesel engines in less than 10 sec at temperatures of −50 F. This material is injected into the intake manifold during the starting operation by atomizing with a special spray gun as shown by Figs. 5 and 6. During the starting tests made in Alaska, attempts to start the engine without the assistance of the starting fluid failed. At subzero temperatures it was necessary to continue supplying the starting fluid for a period up to 30 sec to catalyze combustion until the combustion chamber was warmed up sufficiently to burn the diesel fuel. Although less complete data are available for gasoline engines, cold-room tests have been conducted in which successful starts were made at temperatures as low as approximately −40 F. It should be pointed out that in all the tests the engines were equipped with crankcase oils diluted with gasoline to reduce the viscosity low enough to permit cranking. The reason for the success of this starting fluid compared to others that have been used in the past is the fact that it has a sevenfold wider range of inflammability than pure hydrocarbons. Also, the material is very unstable under the pressures existing in diesel engines and self-ignition takes place at extremely low temperatures. In starting equipment in the North country, it was found that from 1½ to 2 oz of fluid was all that was required for a start even at extremely low temperatures. Operators of



■ Fig. 5 – Diagram of inducting starting fluid



■ Fig. 6 – Diagram of inducting starting fluid in car or truck engine

[2] See SAE Transactions, Vol. 51, October, 1943, pp. 356-361, 368: "Cold-Starting Tests on Diesel Engines," by H. R. Porter.

■ Fig. 7 – Applying Standard starting fluid to Buda diesel engine

■ Fig. 8 – Applying Standard starting fluid to GM diesel engine in an Allis-Chalmers tractor

diesel equipment in that territory estimated that the use of this fluid during the winter months would save from 1½ to 2 hr at each start in comparison to applying heat or continuous cranking of the engine.

Because of the time consumed in starting cold equipment, and due to the waste of valuable materials in idling engines all night, in addition to the loss due to fires resulting from the use of open flames for heating purposes, Standard starting fluid is being distributed throughout the North for use during the coming winter.

Photographs of applying this product in Alaska are shown in Figs. 7 and 8.

In all cases, even though a diluted crankcase oil and the proper starting fluid is used, the battery must be warm enough to crank the engine. As the output from a storage battery falls rapidly at temperatures below zero, the batteries must be maintained at normal temperatures for satisfactory service.

■ Other Aids for Cold Territories

The heater manufacturers have done an excellent job in developing portable heaters that can be used at central locations where large numbers of equipment, personnel, and materials are located. These devices generally consist of a gasoline or fuel oil hot-air furnace and an engine-driven blower to circulate the hot air around the engine.

The old-fashioned gasoline fire solder pot has been used considerably in the North where a vehicle is covered with a tarpaulin and the fire pot is placed underneath to keep the entire unit warm. Ordinary kerosene lanterns have also been used in place of the fire pot in areas where there is no wind.

A heater has also been developed that is installed on individual vehicles which keeps the cooling jacket warm by thermosyphon circulation and the exhaust from the burner is carried to the battery box to maintain its temperature.

The final answer to the proper method of operating equipment in subzero temperatures may lie in the use of auxiliary heat, special winterized fuels and lubricants, or a combination of both. However, regardless of mechanical aids that may be available, it is believed that the use of proper winterized petroleum products to ensure successful operation and to permit starting under extreme conditions in any location, even though auxiliary heat may not be available, justifies the development work that has been done. The contrast between the difficulties experienced last year and the success this winter that has been reported up to this date, shows the advantage of close cooperation between the military services, engine manufacturers, operators, and the petroleum industry.

Synthetic Rubber Applications on Aircraft Engines

continued from page 275

linear function of the aromatic content of the fuels in which they are immersed. For some of the stocks tested, increasing the aromatic content of the immersion fuel hastened the time at which the maximum change in volume was noted.

4. The aromatic fuel extract of synthetic rubbers is a function of the temperature and, to a much less extent, the time of extraction. These variables influence the gum content much more markedly than they do the fuel-insoluble residue. The influence of temperature on the amount of gum extractable from a synthetic rubber stock varies with the basic polymer and compounding ingredients.

5. X-ray diffraction analysis may be used as an aid in identifying the basic polymer and the crystalline fillers used in the synthetic rubber stocks. Particle size of the crystalline ingredients may also be estimated from the X-ray diagrams.

Testing of finished parts made from synthetic rubber was also described and a comparison of the values obtainable from both standard laboratory samples and different-sized finished parts was given.

■ Acknowledgments

The writer gratefully acknowledges her indebtedness to C. E. Carrigan for his helpful comments and suggestions throughout the preparation of this paper and to Dr. Paul F. Spremulli for his help, particularly in the X-ray diffraction studies. The assistance of Miss Sally Levitow in carrying out many of the laboratory tests is also acknowledged with appreciation.

AIRCOOLED ENGINES for Automotive Vehicles

by CHESTER S. RICKER
Detroit Editor of Aviation

MODERN developments in aircooled aircraft engines and the experience of the Army with them in tanks that have been operated in the blistering heat of the desert have made aircooled engines for automobiles much more practicable now than they were only a few years ago.

In the final analysis, however, whether or not aircooled engines will be used in post-war cars, Mr. Ricker explains, will depend on the price of aluminum castings, for aluminum cylinders, heads, and pistons must be used in any successful aircooled automobile engine.

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AIRCOOLED engines for automotive vehicles is a very timely subject. If future automotive vehicles are to be lighter in weight and more economical to operate, aircooled engines will undoubtedly be a contributing factor. Today, over 150,000,000 hp in civilian and military planes is now available from direct aircooled engines. A month ago, the U. S. Navy announced that in the future all their aviation engines would be of the aircooled type.

Aircooling, or had we better say direct cooling, eliminates a lot of the plumbing. It makes all-weather, year-round operation easier. Designs now available for bus applications reduce the overall powerplant weight over 40%.[1] Due to the higher operating temperature differences, these engines can be direct cooled with about two-thirds the air required for a liquid-cooled engine. Further, they can be buried in the rear of a bus, truck, or car and still be cooled as easily as if they were under a hood in front. Tests over many years have also shown that direct-cooled engines have also been unusually economical with fuel. Franklin's road record stands at over 80 mpg, and aircooled aircraft engines are getting below 0.40 lb of fuel per hp-hr in service flights.[2]

Why is direct aircooling more desirable for automotive vehicles today than in the past? Twelve years have passed since the last Franklin car was built. In that period, aircraft-engine cooling has been improved 600%. See Table 1. In this war, the U. S. Army tanks which have battled in the deserts of North Africa in blistering heat have had aircooled engines. How the engineers at the Office of the Chief of Ordnance – Detroit have accomplished this is covered in the latter part of the paper.

In the automotive field there have only been three well-known vehicles with direct-cooled engines since the Franklin. They include the German volkswagen, the Crosley, and the White Horse truck. All of them have flat opposed-cylinder engines, the first and last have four cylinders, the Crosley two. Specifications of the volkswagen and the White Horse truck are given in Appendixes I and II. In the past decade, while the automotive vehicle engineers have been refining and improving liquid-cooled engines, aircraft-engine aircooling research in this country and England has made tremendous strides. For example, just one research cooling project of the NACA has taken eight years but has contributed invaluable information as to the best application of cooling fins to an engine cylinder.[3]

In approaching the problem of direct-cooled automotive engines it seems logical first to review what has been done in aircooling them; then to check the airplane-engine developments to see how well they fulfill the requirements of automotive vehicles.

One of the most interesting developments during the past 10 years has been the volkswagen, which was turned

[This paper was presented at the SAE War Engineering-Annual Meeting, Detroit, Mich., Jan. 10, 1944.]

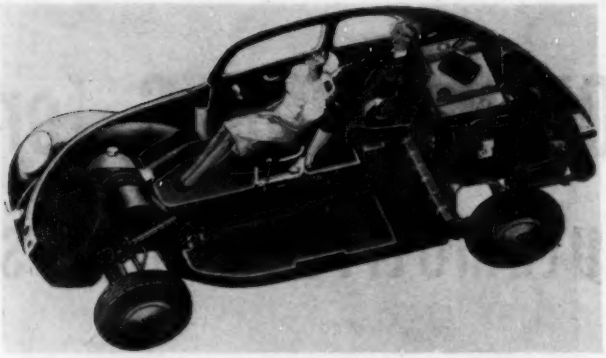
[1] Lycoming Division, Aviation Corp.

[2] Glenn L. Martin at a meeting of the Engineering Society of Detroit.

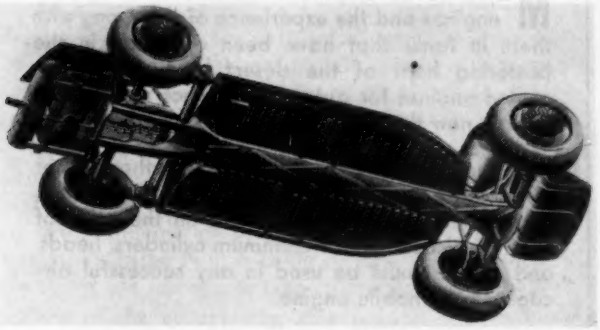
[3] See NACA Technical Report No. 676, 1939: "Surface Heat-Transfer Coefficients of Finned Cylinders," by H. H. Ellerbrock, Jr., and A. E. Biermann.

Table 1 – Ten-Year Development in Wright Fin Design

	1930	1935	1940	
Bmep, psi	138	172	184	208
Fin Pitch, in.	0.375	0.218	0.118	0.118
Fin Depth, in.	0.875 (head)	1.500 (head)	0.648 (cylinder)	0.843 (cylinder)
Cooling Area, Head, sq in.	....	....	2400	2400
Cylinder, sq in.	....	....	1170	1320
Total, sq in.	620	1900	3570	3720
Cooling Area, sq in. per cu in. displacement	3.2	9.4	19.2	18.3
Increase, % times 1930	100	294	600	574
Cooling Area, sq in. per bhp	....	....	29.4	27.8



■ Fig. 1 – Phantom view of German volkswagen showing rear engine, trunk space, and seating arrangement[4]



■ Fig. 2 – Bottom view of the volkswagen chassis showing the fin cooling of the crankcase[4]

into the Nazi "jeep" for the war period. (See Figs. 1, 2, and 3.) Perhaps I will be forgiven for analyzing this enemy product first, for it represents the most thoroughly tested aircooled engine passenger vehicle today and illustrates almost every aircooling problem.

Personally, I do not think this small European type of car would meet the American taste but it suggests what we may expect in future designs. The specifications will give a pretty good idea of what performance can be obtained from a car like this. It weighs about half as much as one of the "big three" and has 45% of the power per ton of weight. See Table 2.

I have prepared this table to give a perspective of the size of engine that may be needed in post-war cars. It seems to be the opinion of many engineers that cars must have the comfort and elbow room of the "big three" today. If they are reduced in weight by new designs and the horsepower is kept around 90 or 100, their road performance as indicated by hp per ton will be as good or better than the figure given here for Cadillac – that is, 76. For buses, there does not seem much likelihood of change because state regulations determine the maximum allowable size and weight. If the payload capacity can be increased by a reduction in engine weight, then aircooled

Table 2 – Horsepower per Ton Vehicle Weight

Hp per Ton	Vehicle	Weight, lb	Bhp
76	Cadillac[6]	3,950	150
61.7	("Big three" average)[6]	2,950	91
58	Willys[6]	2,160	63
35	U. S. jeep[7]	3,120	55
33	Volkswagen[4]	1,430	23.5
24.6	Crosley[8]	975	12
16	M-3 Tank[7]	30,000	250
16	White Horse[9]	5,000 (loaded)	40
13	Bus[10]	30,000	200

Table 3 – Specific Horsepower Developed by 1942 Automobile Engines

Make	Compression Ratio	Hp per Cu In.	Fuel Octane Number Required
Buick	6.0–6.7	0.44–0.52	74-80
Cadillac	7.25	0.43	80
Chevrolet	6.5	0.42	74
Chrysler	6.6–6.8	0.43–0.48	74-80
Crosley	5.6	0.34	74
De Soto	6.6	0.49	74
Dodge	6.7	0.46	74
Ford Six	6.7	0.40	74
Ford V-8	6.2	0.41	74
Hudson	6.5–7.2	0.48–0.53	74-80
Lincoln	7.0	0.43	80
Mercury	6.4	0.42	74
Nash	6.5–6.8	0.44–0.45	74-80
Oldsmobile	6.5	0.43	74
Packard	6.85	0.43–0.46	74-80
Plymouth	6.8	0.44	74
Pontiac	6.5	0.38–0.41	74
Studebaker	6.5	0.47	74
Willys	6.5	0.47	74

engines would be of interest. They should be from 100 hp for light city buses to 200 hp for intercity parlor coaches.[10]

The White Horse offers a very interesting example of an aircooled engine applied to a truck. It is a very special kind of door-to-door delivery vehicle with the power unit attached directly to the rear axle. This arrangement permits the interchange of complete power units in less than an hour.[9] It was brought out in 1939, and over 2000 were built and in use prior to the war. The engine is particularly interesting because it is a smaller version of the well known 4-cyl Franklin 65 hp aviation engine. In this respect it is of interest because it indicates the possible source of well-tried aircooled engines for post-war cars, buses, and trucks. These flat engines lend themselves to rear-engine installations because of their low center of gravity and since it is possible to obtain trunk or carrying space above them.

Suppose we want 80 bhp from an aircooled engine for our post-war car. If we expect to get the same output as from the liquid-cooled engines in 1942 models we will have to obtain 0.434 hp per cu in., which was the average. That means we need a 184 cu in. displacement engine, or one slightly smaller than Studebaker's Champion. With a car the same weight as the Champion we would have a hp-per-ton factor of 63.5, which is just above the present "big three" value of 61.7. The highest and lowest values of specific hp 0.53 and 0.34 would give us 80-hp engines from 151 and 236 cu in. engines, respectively. In Table 3, the specific horsepower per cubic inch is given for 1942 cars, also their compression ratios and fuel octane number.

We will undoubtedly have smaller, more efficient engines for post-war cars. The question here is, will direct aircooled engines give as good performance as liquid-cooled ones. There are three viewpoints on the question: first on a comparison of maximum bhp per cubic inch; second, the kind of fuel used and average cruising horsepower; and third, whether they are rated on average cruising horsepower.

The answers to the first two are given in Table 4, where 18 recent aircooled-engine performances are listed. These range from cylinders of 15 to 202.5 cu in. displacement and bores from 2¾ to 6⅛ in. Compression ratios range from 5.6 to 9.3 and fuel antiknock values from 65 to 100 octane.

Operating under comparable fuel and compression ratios, the specific output of aircooled engines equals that of similar liquid-cooled ones. Twelve years ago 0.300 hp per cu in. was considered good performance for the last Franklin passenger-car engine. Only one engine among those

listed in Table 4 is below this figure and that is because the maximum hp is at governed speed.

It will also be observed that for 73-octane fuel the specific outputs are between 0.30 and 0.40 while for 80-octane they step up from 0.309 to 0.57 and if aviation 100-octane gas can be used the specific hp range is between 0.60 and 0.70. It is interesting to observe that all investigators call attention to the fact that high outputs made possible by aviation fuel and correspondingly high compression ratios, are accompanied by cooler engine operation.

Part of this better cooling of high specific hp engines is also due to their sizes, that is why the bore is listed in Table 4. "Heat loss to the cylinder walls is a function of cylinder volume, and varies as the ratio of the cylinder surface to its volume or inversely as the diameter of the cylinder."[13] This is shown in the comparison of two 100-octane engines, one with 4.25-in. bore and 49.6 cu in. volume, the other with 6.125-in. bore and 202.5 cu in. volume. The smaller engine carries a 9.3 to 1 compression ratio, the latter only 6.7 (the latter engine is also supercharged). This indicates how much more heat must go out through the walls of the smaller engine to allow the higher compression. For that reason it seems logical to expect higher compressions will be used with the smaller-diameter aircooled-engine cylinders, such as are required for automobile engines.

Comparison of specific take-off horsepower in an airplane engine and that for maximum brake horsepower in an automotive engine only indicates the power which may be obtained from a given displacement. This has been covered in the two previous points. In service, engines rarely run at maximum horsepower but at something less.

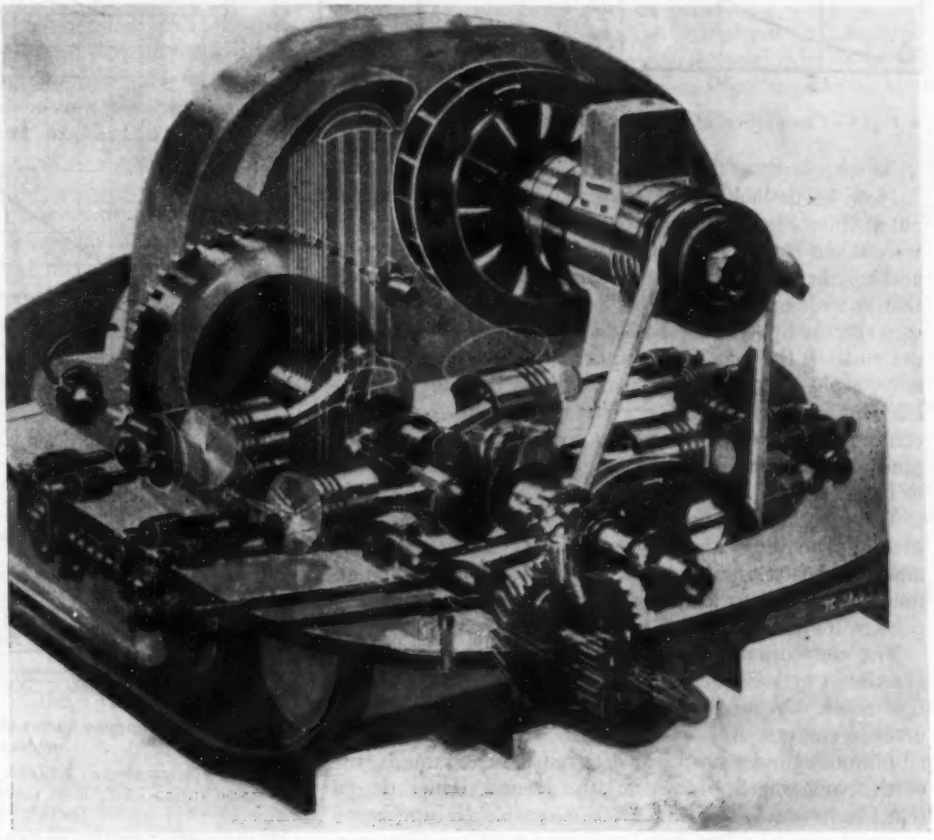
Airplane engines having 0.38 to 0.42 specific horsepower at take off usually cruise at specific horsepowers between 0.25 to 0.30. These cruising values represent 66.0 and 71.5% of the maximum power. No cooling trouble is experienced with values of 0.30 to 0.35 specific horsepower if cylinders are properly baffled.[14]

It is particularly interesting to compare these figures with the aircooled engines in tanks. Under continuous operation at full throttle, 200-hr tests have been run with engines developing 0.309 hp per cu in. This was obtained on 80-octane fuel and was at 75% of take-off horsepower, when used in a plane. This figures out at 0.411 maximum hp per cu in. or in close agreement with automotive engines today.

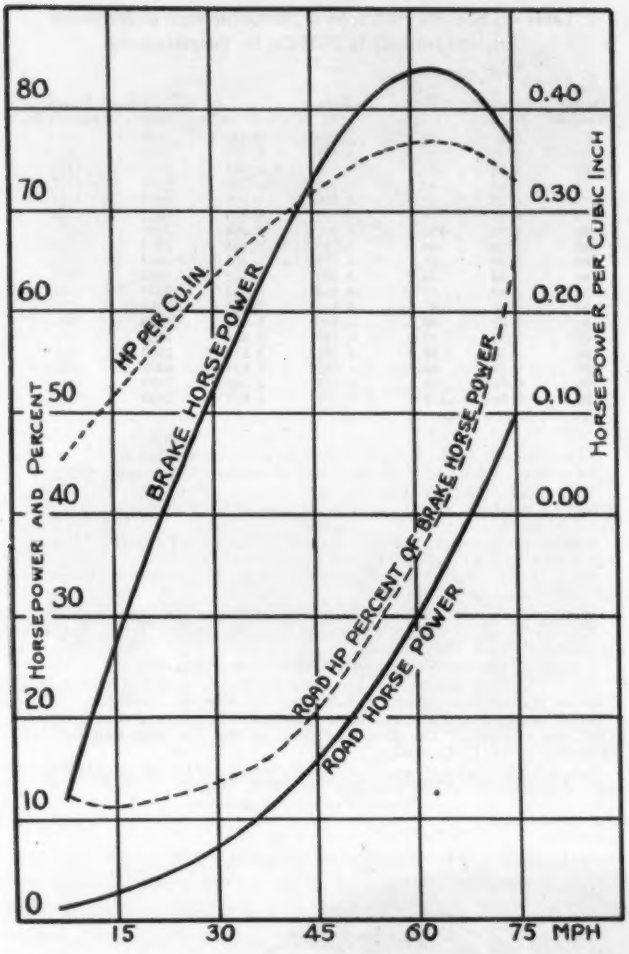
Table 4 – Specific Horsepower per Cubic Inch of Aircooled Engines from 60 to 2603 Cu In. Displacement

Fuel Octane Number	Compression Ratio	Displacement per Cylinder cu in.	Bore, in.	Hp per cu in.	Rpm	Source, Footnote No.
65	5.6	15.0	2.760	0.391	3000	(4)
65	5.7	36.2	3.625	0.276	2600	(9)
65	6.00	73.5	4.125	0.397	2450	(11)
73	6.30	44.0	4.000	0.386	2300	(12)
73	6.10	49.6	4.250	0.326	2000	(12)
73	6.20	73.5	4.125	0.409	2450	(11)
80	5.70	108.1	5.000	0.309	2200	(5)
80	7.00	44.0	4.000	0.455	2500	(12)
80	6.80	73.5	4.125	0.431	2450	(11)
80	7.00	49.6	4.250	0.452	2500	(12)
80	7.00	49.6	4.250	0.453	3500	(12)
80	7.00	49.6	4.250	0.436	2550	(12)
80	7.00	49.6	4.250	0.571	3500	(12)
87	7.50	73.5	4.125	0.454	2450	(11)
100	9.30	49.6	4.250	0.621	3500	(12)
100	9.30	49.6	4.250	0.671	4000	(12)
100	6.90	186.0	6.125	0.646	2300	(4)
100	6.70	202.5	6.125	0.654	2590	(4)

[4] Courtesy of A. T. Robinson, Wright Aeronautical Corp.
[5] Courtesy of the Office of the Chief of Ordnance – Detroit.
[9] See SAE Transactions, Vol. 52, January, 1944, pp. 1-12: "The Effect of Wartime Fuel Developments upon Post-War Automobiles," by A. T. Colwell.
[11] See *SAE Journal*, Vol. 51, July, 1943, pp. 39-41, 56-57: "Engines for Tanks," by Lt.-Col. R. J. Icks.
[12] See *Motor Annual*, Vol. 76, October, 1941, p. 202 of "Specifications of 1942 Cars and Trucks."
[13] Courtesy of George H. Scragg, White Motor Co.
[14] Courtesy of E. W. Allen and H. E. Fox, GMC Truck & Coach Division, General Motors Corp.
[15] Ranger Aircraft Engines, 6-cyl catalog specifications.
[16] See SAE Transactions, Vol. 51, September, 1943, pp. 334-343: "Intake Systems for Aircraft Engines," by Carl T. Doman.
[17] See Proceedings, Institution of Automobile Engineers, Vol. 1, 1906-1907, pp. 223-302: "The Effect of Size on the Thermal Efficiency of Motors," by H. L. Callendar.
[18] See SAE Transactions, Vol. 30, February, 1935, pp. 50-57: "Current Problems in Airplane Engines," by R. F. Gagg.



■ Fig. 3 – Phantom view of the volkswagen engine showing the arrangement of the cylinders and cooling system[5]



■ Fig. 4 – Comparison of road and brake horsepowers at various car speeds[15]

Now let us look at the horsepower curves of a typical automobile, Fig. 4. The road and brake horsepower curves were taken from a previous SAE paper and the percentage and specific horsepower curves computed from them. Note that at wide-open throttle this engine does not reach quite 0.40 specific hp. On the basis of road horsepower, 0.22 hp per cu in. is the maximum required to keep the car rolling at 75 mph. Hence it may be safely said that any automobile engine designed to operate at 0.35 hp per cu in. continuously can be successfully aircooled, for aircraft engines are now being operated for hours developing 0.45 or better and do not overheat.

The design and materials used in aircooled engines differ greatly from those used in liquid-cooled engines: cylinder and fin construction, cylinder material, cylinder arrangement, air circulating systems, valve location, valve gear, pistons, lubrication and oil cooling, and accessory drives.

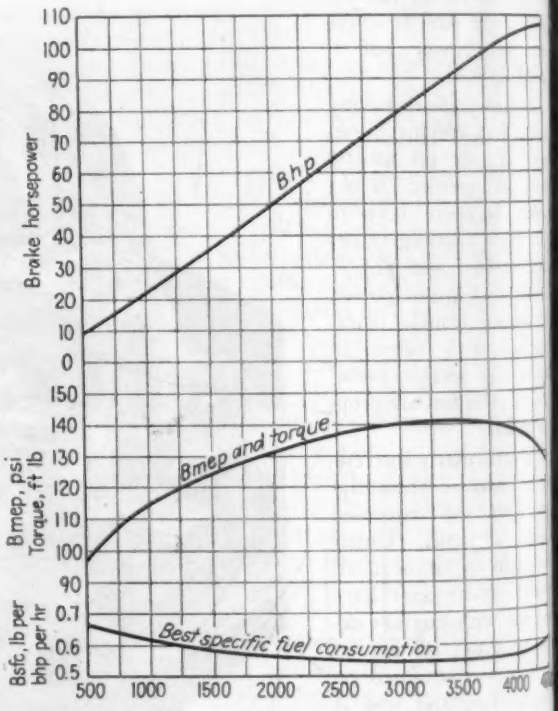
The conventional aircooled cylinder construction is an aluminum cylinder head into which a steel cylinder barrel is screwed. On smaller engines the cylinder barrel may be made from cast iron. On the last Franklin engine the aluminum cylinder head was detachable. The same is true of the volkswagen. However, the latest cylinder design that I have seen is one off Carl T. Doman's drafting board in which the cylinders, crankcase, and head are all cast in an aluminum block. "A high-grade cast-iron liner 0.030 in. thick is inserted into the bore with a 0.012-in. spring fit."[16] The performance curves of this engine are given in Fig. 5. This engine according to its designer has 3.375-in. bore and 2.812-in. stroke. It is designed to give 100 hp at 3800 rpm with 7 to 1 compression ratio using 73-octane fuel. "There are many features of the design which are not as yet settled. Nevertheless, in general, it is designed to be built in large production using the same machinery and tools as followed in the automotive industries on water-cooled engines. The design is based on the assumption that the price of aluminum will be reduced to a point where the aluminum block casting for the engine will cost no more than the cast-iron block used in water-cooled engines. The engine has been designed to keep the distance between cylinder centers comparable with conventional automotive practice."[16] In this design aircraft air-cooling experience is being applied but the cylinder arrangement and method of fabrication is engineered so that it may be installed in a conventional automobile.

In discussing the sleeved cylinder bore inside an aluminum casting, E. W. Allen[10] suggested three possible alternatives to the pressed in cast-iron sleeve: plating the finished aluminum with "Holochrome," metal spraying the inside with a ferrous alloy, or even hydrogen furnace brazing a very thin sleeve in a high-silicon aluminum-alloy casting using an aluminum brazing compound.

Aluminum plays a most vital part in the success of modern high-powered aircooled engines because of its greater conductivity compared with iron and steel.

[15] See SAE Transactions, Vol. 49, October, 1941, pp. 432-441: "Engineering for Better Fuel Economy," by H. T. Youngren.

[16] Courtesy of Carl T. Doman, Aircooled Motors Corp.



■ Fig. 5 – Performance curves of proposed Franklin aircooled automotive engine[16]

Bore: 3⅜ in.; stroke: 2 13/16 in.; displacement: 151 cu in.; compression ratio: 7.0:1; inlet valve diameter: 1⅝ in.; exhaust valve diameter: 1¼ in.; lift (both): ⅜ in.; carburetor: side draft, 1¼-in. venturi; spark advance: 30 deg at 3800 rpm; fuel: 73 octane; oil SAE 20; inlet valve opens at 20 deg BTDC, closes at 60 deg ABDC; exhaust valve opens at 60 deg BBDC, closes at 20 deg ATDC

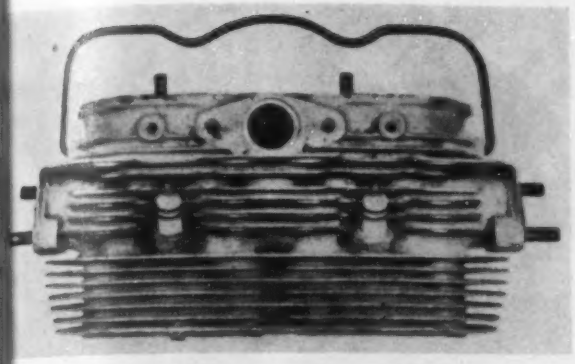


Fig. 6 – Side view of volkswagen aluminum dual cylinder head. Note exhaust outlets on each side of head and central intake port[5]

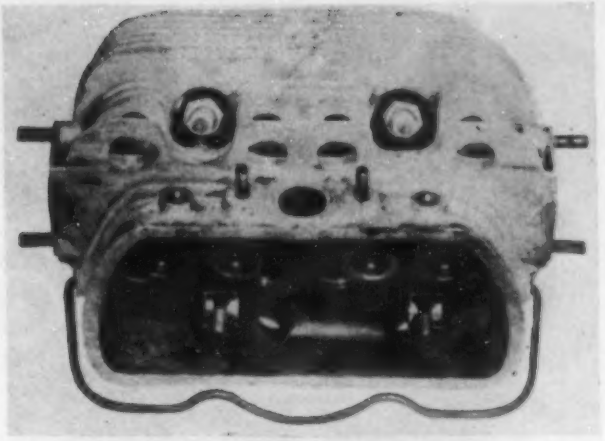


Fig. 7 – Top view of dual cylinder head used on volkswagen engine, showing valves and rocker arm shaft brackets[5]

According to Kaye and Laby the conductivity of the following metals based on cast iron as one are:

Cast iron 1.0; steel 1.3; magnesium 3.0; aluminum 4.0; copper 8.0.

Use of aluminum is one reason for the great improvement in cooling in recent years. In every case it will be observed that ferrous alloys have been used for the cylinder barrels since none of the high-conductivity alloys have the wearing qualities so essential to long cylinder life.

The first Franklin engines had cast-iron cylinders with integral heads. In the last model a detachable aluminum head with cast-iron barrel was used. This is the construction followed in the volkswagen engine, the cylinders of which have a dual head that is bolted to the separate cast-iron cylinder barrels. Four capscrews, equally spaced around the cylinder bore clamp cylinder and head together. A gasket seals the joint. The upper part of the head is extended to enclose the overhead valves and the rocker arm shaft. Holes are drilled through the fins to allow inserting the hold-down screws. A side and top view of this cylinder construction are shown in Figs. 6 and 7.

For current aviation engines the practice is to screw the cylinder barrel into a heated aluminum head while the temperature is about 450 F. One such assembly has cast fins on both cylinders and head which are unmachined. This design is unique in having deeper fins on the exhaust side of the cylinder barrel than on the intake, thus reducing the center distance between the adjacent cylinders. However, this introduces a machining and assembling problem so as to get head and cylinder in proper relationship when they are screwed in place.

The next advance was to use aluminum heads with cast fins and steel cylinders with very thin, 0.015- to 0.030-in. thick fins and spaced about 0.10 in. apart. The exhaust passage and rocker arm box are well finned but the intake is not. The cylinder head usually carries two spark plugs – one in front, the other diametrically opposite. Such a cylinder is shown in Fig. 8 (left), and was the first type used in tank engines.

To increase the cooling area, a new fin pattern was developed with deeper ribs on the head and a deep fin cast-aluminum muff shrunk on the cylinder, as shown in Fig. 8 (right).

Power demands continued to increase until it became necessary to contour mill the fins on the aluminum heads as well as turn them on the steel barrels. Sodium-cooled hollow exhaust valves also were an important contribution to higher allowable compressions, more power, and longer service life. Such a cylinder assembly is shown in Fig. 9. The use of valve-seat inserts, which are necessary with aluminum heads, is clearly shown in this illustration. Note the way the cylinder threads are masked from the combustion chamber by an extension of the cylinder wall. The threads are of the butress type, to assume the axial thrust of the high operating pressures. Another combination of aluminum cylinder head and steel cylinder with turned fins is the Ranger shown in Fig. 10.

Improved cooling has been obtained by shrinking a forged aluminum-alloy finned sleeve over a plain steel barrel. This is standard practice on Pratt & Whitney radial engines. The aluminum forging permitting very close thin fins to be turned on its outside. An alternative construction is the casting of aluminum on the steel cylinder barrel.

This construction is now used on some engines by both Ranger and Continental. Such a cylinder is shown in Fig. 11.

The arrangement of fins, their thickness and spacing are indicated by NACA research.[3] Briefly, the tests were to determine "effect of fin width, fin space, fin thickness, and cylinder diameter on the heat transfer." "Tests were made with free air, with and without baffles, also with devices to create turbulence and with blower cooling."[3] The latter is of particular interest to automotive-engine designers.

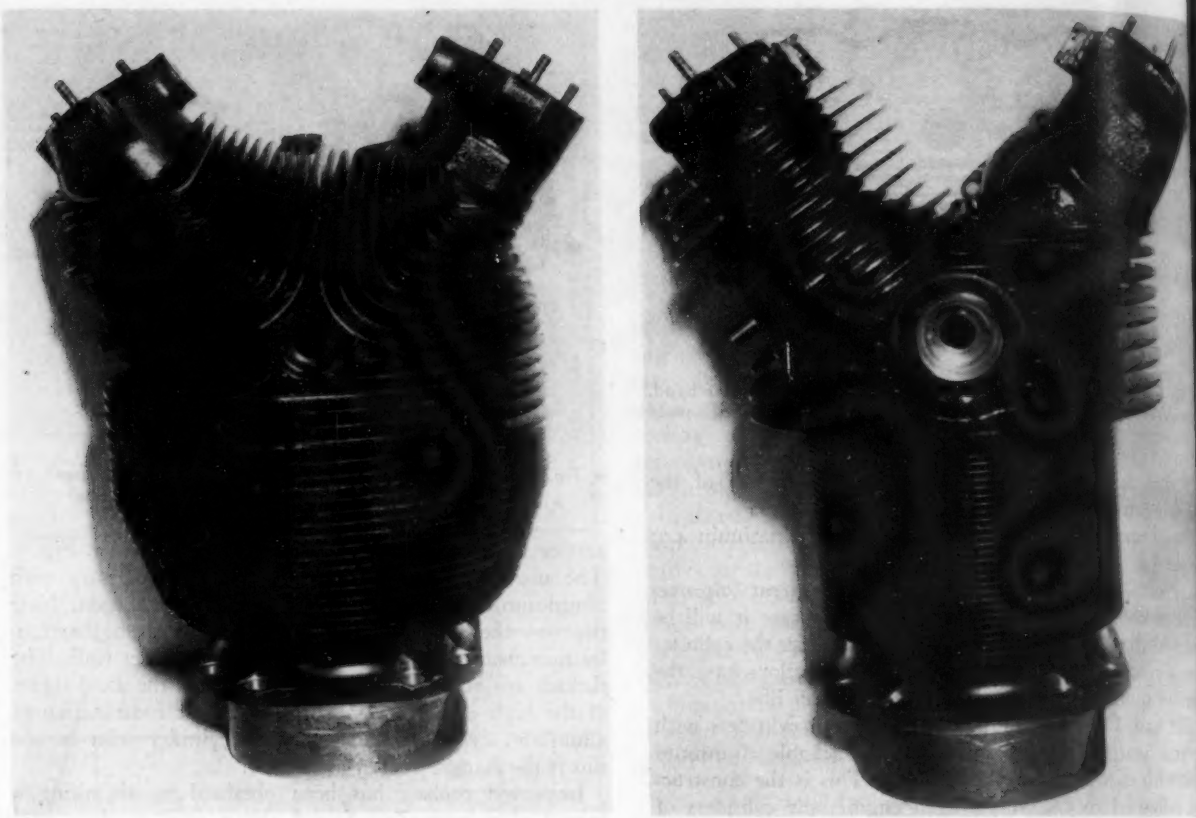
"*Turbulence* had little effect on heat transfer."[3]

"*Angular directing* of air gave the highest heat-flow rates and was most effective when fin plane to airstream angle was between 30 and 60 deg." In plotting the results at 40 and 150 mph air velocity with the angle of attack 45 deg, it was found that best results at 150 mph were obtained with about 5/32-in. fin spacing, but at lower speeds the cooling increased as the fins were spread apart. See Fig. 12. The efficiency of angularly directed airstream was discovered in applying air-deflecting vanes to the tank engines.

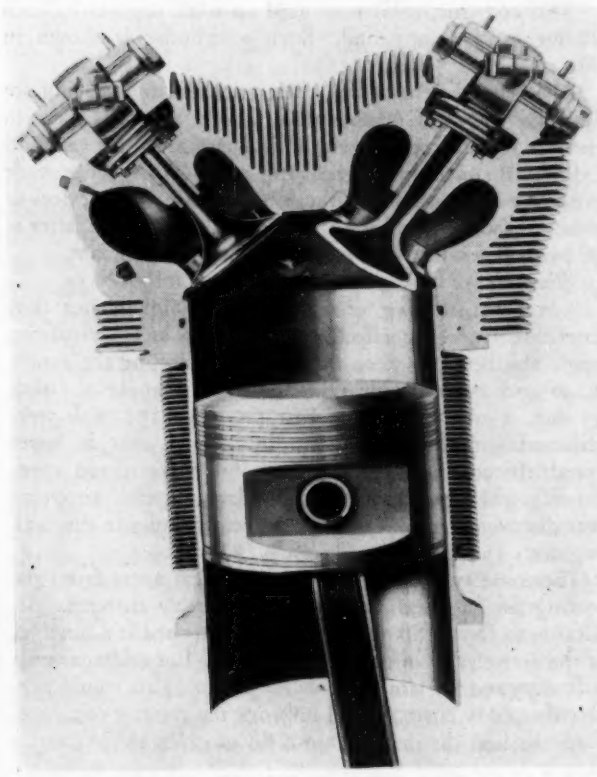
There are two ways that air takes heat away from the cooling fins: by *laminar* flow and by *eddy* currents. According to Dr. T. Stanton,[17] the heat absorbed is a function of the air velocity in mph or $H = fV^n$. For eddy currents $n = 0.73$ and for laminar flow $n = 0.50$. This would suggest that eddy currents will improve the cooling efficiency.

As the heat dissipated from a fin of given section varies

[17] See NACA Technical Note No. 572, 1936: "Performance of Air-cooled Engine Cylinders Using Blower Cooling," by O. W. Schey and H. H. Ellerbrock, Jr.



■ Fig. 8 – Alternative types of cylinders made by Continental Motors Corp. for use on the Wright-Continental 9-cyl radial aircraft engines. Cylinder at left has turned steel fins; cylinder at right, a cast aluminum muff shrunk over the steel barrel. 20% more power is obtained from the latter cylinder



■ Fig. 9 – Wright Whirlwind cylinder construction is clearly disclosed in this cross-sectional view[4]

as the length, while the weight varies as the square of the length, the cooling fins should preferably be kept short.[18] This applies to airplane engines particularly, but it is not so important in automotive applications where extra weight only affects cost. This may be entirely offset by lower cost of cooling fan.

Dr. A. H. Gibson recommends 5/16 in. as the minimum spacing of cast-aluminum fins, and the fins tapered not only for foundry convenience but to get the heat out to the tips. "The pitch should not be less than 0.15 in. and 0.20 in. is better, otherwise there will be interference of the boundary layers of the cooling air with consequent loss in cooling efficiency when the air has to flow between fins for a distance of more than 6 in. from the leading edge."[19] These tests were made with various fin spacings and air velocities from 30 to 150 mph.

Some other fundamental facts are that:[20]

"At air speeds from 30 to 88 fps, the heat loss was sensibly proportional to the mean temperature difference between the fin surface and air."

"Roughness of fins did not affect heat dissipation."

"Cast-aluminum fins improved 10% over unpainted ones when coated with black stove enamel."

This leads us to the question of how much cooling surface should be provided. One convenient empirical basis

[18] See "Aircraft Engines," by A. W. Judge, 2 vol., 1940-1941, Van Nostrand.

[19] See NACA Technical Report No. 488, 1934: "Heat Transfer from Finned Metal Cylinders in an Air Stream," by A. E. Biermann and Benjamin Pinkel.

[20] See Proceedings, Institution of Automobile Engineers, Vol. 15, 1919-1920, pp. 243-302: "The Air Cooling of Petrol Engines," by A. H. Gibson.

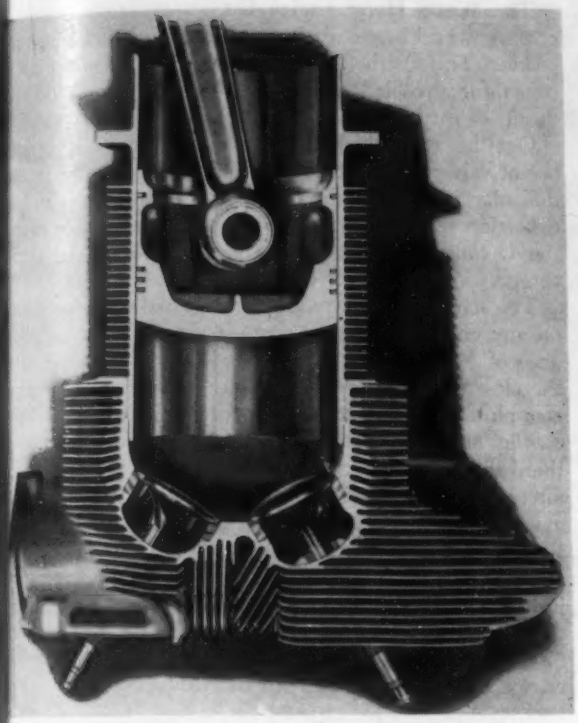


Fig. 10 – Cross-sectional view of the Ranger inverted cylinder engine. The aluminum valve gear housing is bolted to the cylinder heads of this engine. One housing covers all six cylinders[11]

is the ratio of internal cylinder surface to total external cooling area. In Table 5, several engines of recent origin are compared with some English engines built over four years ago. This seems to indicate that with present-day performance we should have at least a 15 to 1 ratio. All of the engines listed have ferrous-alloy cylinder fins. More cooling or fewer and lighter fins would be required if aluminum were used.

Another analysis of cooling areas is given in Table 6, where the distribution of cooling to head and barrel is tabulated. This ratio varies from 1.66 to 2.31. The most common relationship, the square inches of cooling area to cubic inches of piston displacement is quite consistent in the range given or about 20 for high performance. I have my doubts that it would hold in cylinders of less than 4-in. bore, such as we need for automotive engines. This is the one diameter where the area and circumference of a circle are equal. Under 4 in., the circumference decreases less rapidly than the area, while above 4 in. the area increases more rapidly than the circumference. This must be considered in connection with the stroke-bore ratio and the amount of cooling area that will be available on the cylinder barrel.

If we relate the cooling area to the bhp, the effect of

Table 5 – Ratio of Internal Surface to Cooling Fin Area

Engine Make	Bore, in.	Stroke, in.	Compression Ratio	Estimated Internal Area, a, sq in.	Total Cooling Area, b, sq in.	Ratio $\frac{b}{a}$
Wright Radial..	6.125	6.875	6.7	241.74	3720.0	15.4
Wright Radial..	6.125	6.312	6.9	231.74	3570.0	15.4
Franklin.....	4.625	4.000	7.0	83.40	1073.5	12.9
Franklin.....	4.625	4.000	7.0	83.40	1333.0	16.0
Franklin.....	4.625	4.000	7.0	83.40	1410.0	16.9
Franklin.....	4.250	4.250	7.0	67.70	889.7	13.1
Franklin.....	4.250	4.250	7.0	67.70	1098.0	16.2
Jupiter (1939)..	5.750	7.500	5.3	191.80	1252.00	6.6
Rapier (1939)..	3.500	3.500	6.0	51.20	454.0	8.9

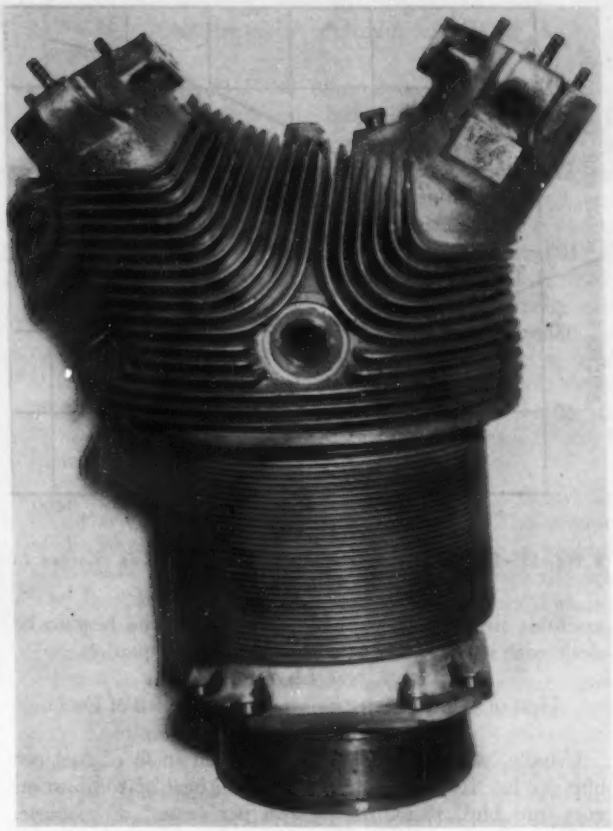


Fig. 11 – For Continental aircraft engines it has been found economical to cast an aluminum sleeve around a steel cylinder instead of shrinking on an aluminum muff. Cooling fins are then turned in the aluminum. This procedure may find a useful application in motor-vehicle engines

cylinder size can easily be discerned. The area of a 4.25-in. bore engine is around 50 sq in. per bhp, on a 4.625-in. bore about 40 sq in., and with a 6.125-in. bore around 30. Thus the smaller-bore engine requires a greater cooling area per hp, and it is harder to obtain the extra surface.

Fundamentally we should go to the fuel to determine the amount of heat that must be handled by the cylinder

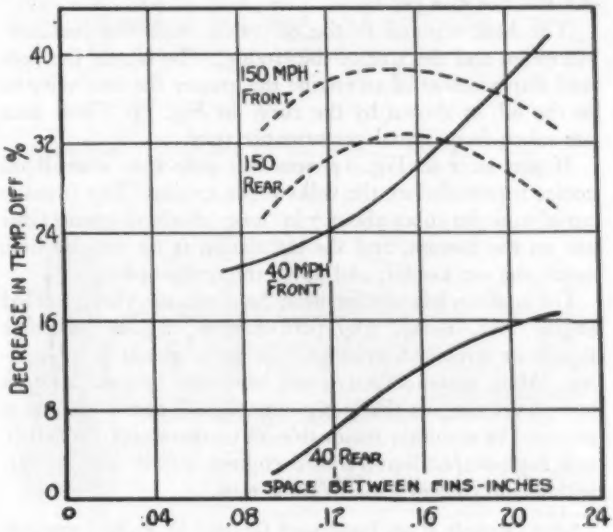
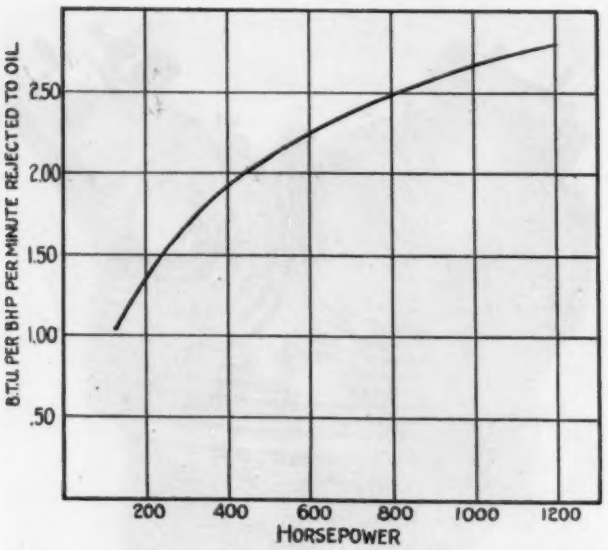


Fig. 12 – NACA test curves with air directed against the fins at 45 deg to the plane of the fin showing optimum fin spacing[3]



■ Fig. 13 – Relationship of engine output to the Btu's rejected to the oil per bhp[21]

assembly fins and the lubricating system. The heat to be dealt with is easily evaluated by the simple equation.

$$\text{Heat of Combustion} = \frac{\text{Lb of Fuel}}{\text{Time}} \times \text{Btu of Fuel}$$

Usually, we have the fuel value given in lb of fuel per bhp per hr. If we assume 25% of the heat of combustion goes into bhp, 18,400 as the Btu per pound of gasoline, 75% as the heat of the bhp that goes into the cylinder walls, and from 3 to 10% of the bhp heat to the oil we have the following values:

Heat per pound of fuel rejected to cylinder walls = 57.5 Btu per bhp per min.

Heat per pound of fuel rejected to oil varies from 2.3 to 7.5 Btu per bhp per min.

Hence, for a 1200-bhp engine using 0.5 lb of fuel per bhp-hr, the fuel consumption would be 10 lb per min and the total heat rejected: $57.5 \times 0.5 \times 1200 = 34{,}500$ Btu per min. The heat rejected to the oil would then be $6.0 \times 0.5 \times 1200 = 3600$ Btu per min. In actual service tests this engine rejected 3400 Btu per min to the oil or 2.8 Btu per min per hp.

The heat rejected to the oil varies with the fuel consumption and the size of the engine. The bigger the bore and displacement of an engine the greater the heat rejected to the oil, as shown by the curve in Fig. 13. These data are taken from actual performance tests.

If you refer to Fig. 14, you will note that a small oil cooler is provided on the volkswagen engine. This is made up of nine flat tubes about 7 in. long. Both oil connections are on the bottom, and the circulation is up on one side, across the top header, and down the opposite side.

Oil cooling has always been held against the aircooled engine but, today, high-performance engines, whether liquid- or aircooled, must provide some means of oil cooling. Most automobile engines normally operate at such low power outputs that a big aircooled oil pan is all that is required to maintain reasonable oil temperatures. Whether new high-output liquid-cooled engines will be able to run without oil coolers remains to be seen.

[21] See "Methods of Oil-Temperature Control," by F. M. Young and W. R. Ramsaur. Presented at the SAE National Fuels & Lubricants Meeting, Tulsa, Okla., Oct. 7, 1938.

The White Horse engine has an ingenious system of cooling the oil by circulating it through an intake manifold jacket. This method may eliminate the need of an oil cooler for aircooled engines and provide a most desirable means of getting heat to the manifold.

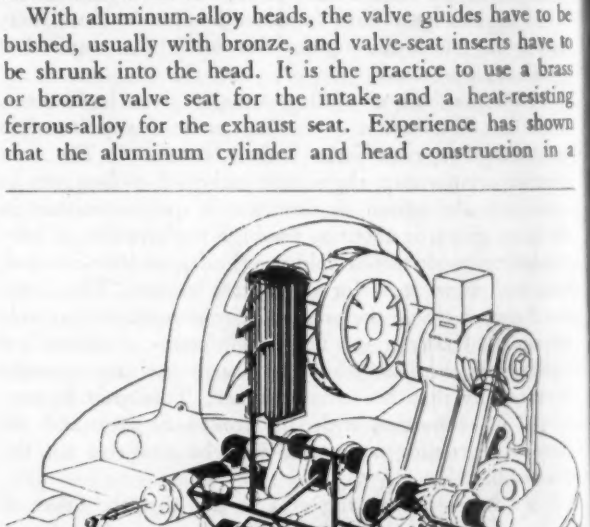
One thing on which practically all engineers agree is the use of valves in the head. There are three reasons for this, difficulty in cooling the L-head cylinder evenly and the distortion which follows, the need of a detachable head if the L construction is used, and the difficulty in getting compression ratios over 6 to 1.

The narrowest fins used on individual cylinder barrels, on any engine I have seen, are ⅜ in. wide. If an allowance of ⅛ in. between cylinders is made for baffles, then the closest cylinder center distance will be cylinder diameter plus 0.875 in. On a water-cooled engine this is about 0.25 in. on adjacent pairs of cylinders in some designs where there is no water between the cylinders. Thus an in-line aircooled 6-cyl engine might be about 3⅜ in. longer than a water-cooled one of the same bore and stroke. On the Ranger in-line engine and on some of Carl T. Doman's, the fins are practically eliminated between adjacent cylinders. If all the cylinders are cast in one aluminum block this problem would disappear. The illustration of the "Ranger" engine, Fig. 15, shows how close an in-line engine can be built even with individual cylinders.

■ Cylinder Heads

The cylinder head integral with the cast-iron cylinder was one of the limiting factors in power output on the early Franklins. It was frequently possible to get 800 F head temperatures with this design. When an aluminum head was substituted, the compression ratio was raised from 4.7 to 5.1 because the head temperature was kept down well below 600 F.

With aluminum-alloy heads, the valve guides have to be bushed, usually with bronze, and valve-seat inserts have to be shrunk into the head. It is the practice to use a brass or bronze valve seat for the intake and a heat-resisting ferrous-alloy for the exhaust seat. Experience has shown that the aluminum cylinder and head construction in a



■ Fig. 14 – Lubricating system of the volkswagen, which shows the oil cooler very prominently[5]

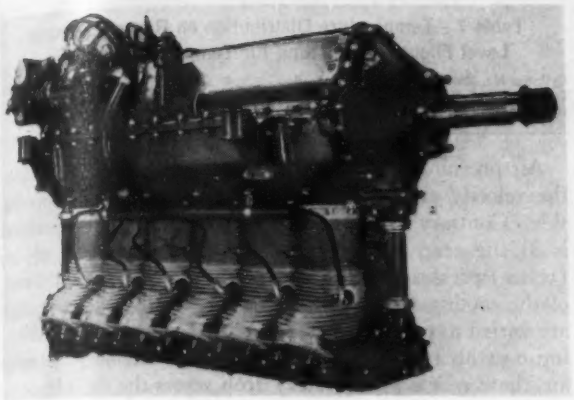


Fig. 15 – Inverted Ranger in-line engine on which the fins are practically eliminated where the adjacent cylinder heads almost touch one another[11]

60 mph air slip stream should have 0.28 to 0.35 sq ft of cooling surface per bhp and that cast-iron cylinders require an additional 50% in area. This indicates how much more desirable aluminum cylinders are for aircooled engines.[18]

A greater percentage of the heat content of the fuel burned goes out through the cooling system in the smaller-bore engines, as shown by the following percentages:[18]

Bore, in.	4	6	8	10	12
Total Heat, %	29	22	19	17	16

For automotive work in contrast to airplane service, the maximum torque should be between 1000 and 2000 rpm instead of around the normal airplane cruising speeds of about 2500 engine rpm or higher. This introduces the question of whether more heat is lost to the cooling system at low or high speeds.

The success of aircooled engines in aircraft has been due to an adequate supply of cooling air at comparatively low temperatures. This is not true of either automotive vehicles or tanks. They must be designed to cool properly whether the vehicle is moving fast or slow. It means circulation of an adequate volume of air by a fan of some type. Since the horsepower required to drive the fan is negative, it should be kept as low as possible. How serious this is, was clearly shown in the 1931 Franklin SAE paper,[22] where the fan efficiency and cooling were improved simultaneously. Bhp was increased from 67 to 87, or 30%, while the net horsepower, after deducting fan and other resistance hp losses, was only stepped up 46%. The fan horsepower being reduced from 14.5 to 6.70 or 54%. In both cases the fans delivered 46 cu ft of air per min per bhp. This compares very closely with the results obtained in tank cooling, where approximately 40 cu ft per min per maximum bhp are required.

Fan horsepower must be kept down to a minimum. That is why the maximum finning that can be used without interfering with production or increasing costs should be used. This indicates the need for using copper fins although they add weight. Fins do not absorb horsepower unless they obstruct the airflow. What is good average fan horsepower practice cannot be given now because there has not been enough automotive development along this line. However, tank fan hp varies from 34 to 6.2% of the maximum bhp of the engines used.[7] Compare this with 7.2% of bhp for the earlier Franklin and 5.0 for the 1931 model.

Lt.-Col. R. J. Icks also pointed out one of the most important advantages of direct aircooling in his SAE paper, *the 100% higher air temperature difference:* "One of the major obstacles to the use of liquid-cooled engines in combat vehicles is the greater airflow required and consequent difficulty involved in their cooling. The reason for this is that whereas the fins of aircooled engine cylinders in operation will rarely drop below the perfectly satisfactory temperature of 350 F, 225 F is usually considered the practical limit for radiator temperature in the cooling system of a liquid-cooled engine. Assuming operation at an ambient temperature of 125 F, the water-cooled engine permits only a 100 F temperature difference, while the aircooled engine has a temperature difference of 225 F in which to work."[7]

Cooling depends on the volume of air passed and the temperature difference across the cylinders. The velocity of the air will depend directly on the difference in pressure and inversely as the area of the openings through the fins. The Ranger method of cooling in-line cylinders is shown in Fig. 16. These engines operate successfully with 3-in. (water) pressure drop across the cylinders.[23] Doman considers 6-in. differential pressure enough for any automotive engine. This figure is substantiated by NACA tests on a 4-cyl Continental aviation engine.

Trying to increase cooling with a fan is expensive from a power standpoint. It takes four times the cfm to double the cooling effect by increasing the quantity of air delivered. This requires greater air pressure if the openings are not increased. Such pressure increases as the square of the change, hence four times the pressure will be needed to double the cooling. Fan horsepower, therefore, varies as the sixth power of the change in cooling so to double the cooling would take 63 times the original fan horsepower

Table 6 – Analysis of Cooling Areas of Aircraft Engines

Bore, in.	Stroke, in.	Head Total Area, sq in.	Head Fin Area, sq in.	Barrel Total Area, sq in.	Barrel Fin Area, sq in.	Total Area, sq in.	Ratio of Head Area to Fin Area	Sq In. Cooling Area per cu in. Displacement	Sq In. Area per Bhp
4.625	4.000	671.3	583.1	402.1	345.1	1073.4	1.86	16.0	43.4
4.625	4.000	930.9	842.7	402.1	345.1	1333.0	2.31	19.9	43.9
4.250	3.500	589.1	518.7	320.6	270.0	889.7	1.77	17.9	55.0
4.250	3.500	1098.1	987.7	Barrel and		1098.1	....	21.7	49.0
4.625	4.000	1410.0	1261.0	head together		1410.0	....	21.0	39.8
6.125	6.875	.....	2400.0	1320	1320	3720.0	1.82	18.3	27.8
6.125	6.312	.....	.....	1170	1170	3570.0	2.05	19.2	29.4

[22] See SAE Transactions, Vol. 26, 1931, pp. 339-345: "Development of the Franklin Direct Aircooled Engine," by E. S. Marks and C. T. Doman.

[23] See SAE Transactions, Vol. 32, October, 1937, pp. 473-482: "Features of the In-Line Aircooled Aircraft Engine," by A. T. Gregory.

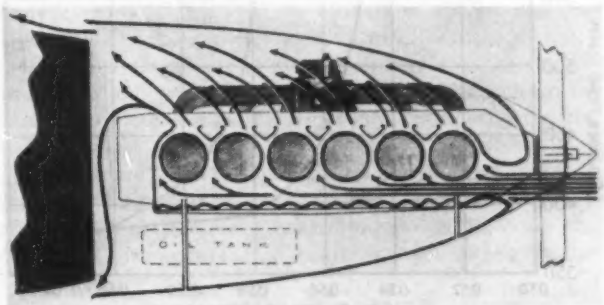
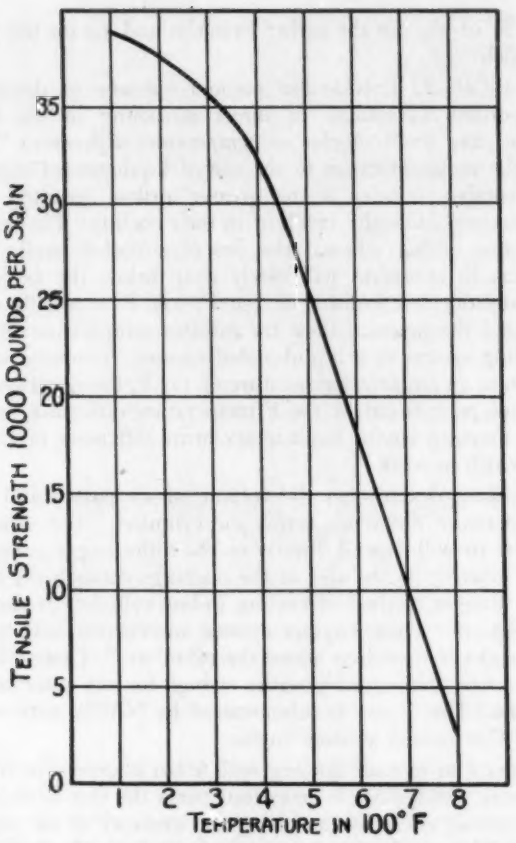


Fig. 16 – How Ranger aircooling is arranged is shown in this diagrammatic illustration[11]



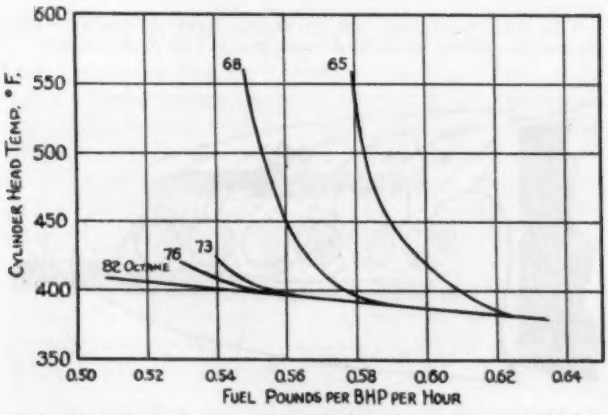
■ Fig. 17 – Characteristic temperature – strength curve of a typical aluminum-alloy aircraft-engine cylinder head showing why operating temperatures must be kept below 450 F

if it were initially unity. Hence, any cylinder that cannot be cooled properly with 5 to 6 in. of water pressure should be redesigned or the air passages and baffles should be modified to increase their effective areas or proper direction of airflow.[24]

■ Baffling Cylinders

Baffling cylinders have played as important a part in successful cooling of cylinders as the finning methods. On the Ranger engines the cylinders are air jacketed so that the airstream scours the fins and is directed at the hottest portions as desired.[23]

[24] Charles T. Grimes, Aircooled Engines, Inc.



■ Fig. 18 – Effect of fuel economy and octane ratings on cylinder-head temperatures

Table 7 – Temperature Distribution on Ranger Cylinders Level Flight With Ground Air Temperature of 63 F[11]

Cylinder No. (front)	6	5	4	3	2	1 (rear)
Cylinder Head, F	366	365	355	380	370	350
Cylinder Base, F	210	190	200	205	215	215

Air pressure is built up on one side of the cylinders and the velocity of the ship sets up a suction on the opposite side. Contrary to current impressions the highest pressure is at the rear cylinder. In tests where there is 5½-in. (water) pressure at the entrance there will be 7 at the rear of the cooling air duct. Therefore, the air-jacket openings are varied to equalize this difference in pressure. Developing 0.54 hp per cu in. at full throttle, with 50 F entering air, there was a 3 in. (water) drop across the cylinders and an air temperature rise of 80 F. Under these conditions the maximum cylinder-head temperature was 450 F and the barrels 225 F. In level flight the figures given in Table 7 show how well carefully metered air jackets can control uniformity of cylinder temperature.[11] In this respect also see the Tank section later in the paper.

Carl T. Doman, on his proposed 150 cu in. engine, is putting an air manifold on each side of the cylinder block and directing air against the part of the cylinder he wants to cool most. The fins, as you will observe, are both horizontal and vertical. This eliminates the common hot spot in the rear of all cylinders with cooling air directed from one side.

On the tank engines, turning vanes were applied both in front and behind the cylinder blocks. These not only directed the air so it would cool the cylinders to better advantage but has the effect of increasing the effective area and thus increasing the cfm without requiring increased fan hp. See Tank section. In any rear-engine installation these methods should be carefully considered, for a very much smaller fan may be required to do the job than if they are ignored.

■ Valves and Valve Gears

Camshafts on top of cylinders do not fit into the economical engine design but valves in the head are a must with all aircooled engines. The use of a single camshaft in the crankcase, with push-rods and rocker arms to operate the valves seems to be the best practice. A larger intake than exhaust valve works well in these designs. Valves at an angle and also with stems parallel to the cylinder axis are used. The latter is the least expensive and used on the smaller low priced engines, while the former is current practice on big radials.

In engines under 200 cu in. displacement, engineers generally concede that hollow sodium-cooled exhaust valves are not necessary. The same kind of valves used in water-cooled engines should work perfectly with automotive-engine aircooling.

The aluminum cylinder expansion being greater than that of the steel valve push-rods used to be a big problem in the Franklin days. All kinds of tricks were used to keep the valve clearances constant whether the engine was hot or cold. Now the use of hydraulic valve tappets which maintain a constant valve clearance have solved that problem. These have been used on Franklin aircooled motors for years, although they were originally developed for water-cooled engines. They not only assure uniform valve action but also make an overhead valve engine quiet, which it must be to meet water-cooled competition.

The valve seat problem is little different from that of the water-cooled engine except that a seat must also be used for the intake valve as well as the exhaust. Valve seats are inserted in hot cylinders instead of being frozen and inserted in cool ones.

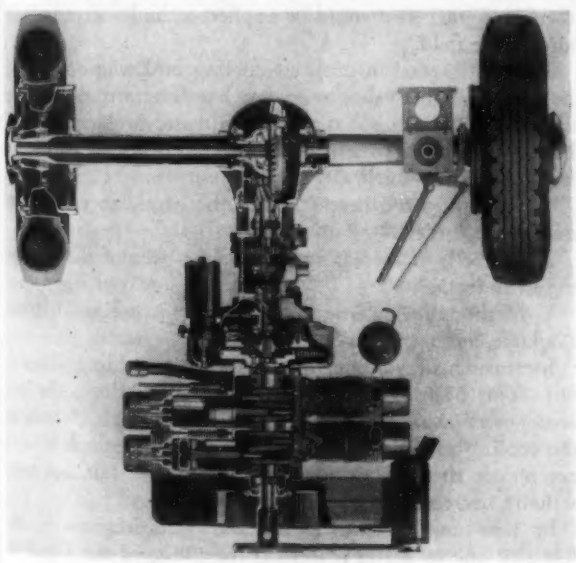
High power, high torque, and high rpm are being obtained by using a stroke-bore ratio of less than one. Doman uses as low as 0.825 ratio. This gives big valves in proportion to the displacement. It also allows valves to be separated by an adequate bridge at the center of the cylinder head. Proper cooling of this point and the spark plugs seem to be the most likely reasons for success or failure of any engine, while low cylinder-barrel temperatures mean long life of pistons and rings and low oil consumption.

Pistons and Rods

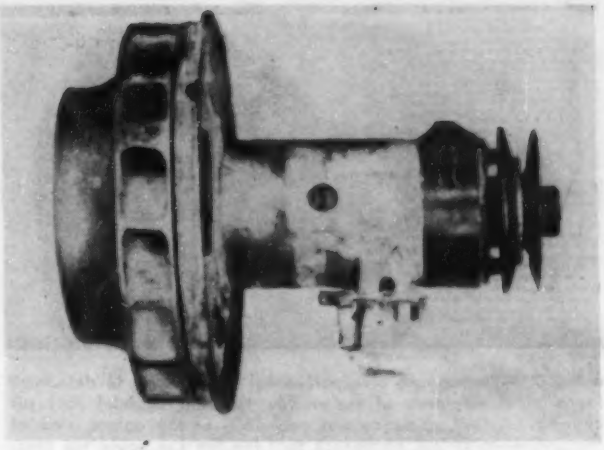
Aluminum pistons are a must in aircooled engines not only because of weight saving but for temperature control. Floating piston pins are almost universal in application to aircooled engines. Connecting rods are usually very short on aircraft engines to keep weight low, but for automotive-vehicle engines, most engineers consider they should be around 1.75 times the stroke.

Temperatures

It is generally considered satisfactory to operate engines with cylinder temperatures around 350 F and heads at the hottest point around 450 to 500 F. These seem awfully high to a liquid-cooled engine man until one checks up the actual temperatures on the inside of the water-cooled engines. They run from 266 to 302 F in liquid-cooled aircraft designs compared with 392 to 464 F for comparable aircooled engines.[18] In 1936, experiments run on water-cooled engines showed cylinder-wall temperatures at various engine speeds and from road load to full load ran from 260 F to 680 F.[25] This brought out one remark made by an engineer using liquid-cooled engines who said, "I think these big aircooled engines have a lot cooler exhaust valve than we do."



■ Fig. 19 – Sectional view of the White Horse power unit showing the relation of the engine to the other units and its general construction[9]



■ Fig. 20 – Side view of the volkswagen fan, which is carried on an extension of the electric generator shaft. The generator and fan are belt driven from the crankshaft[5]

These temperatures, however, introduce a most important material problem, namely, the tensile strength of aluminum alloys at these operating temperatures. This is something the liquid-cooled engine designer does not have to consider with his cast-iron cylinders. A typical aluminum alloy at 100 F has a tensile strength of about 39,000 psi (Fig. 17). At 400 F, the strength is about 15% less, at 500 F, the strength drops another 24%. While at 600 F, it is only half as strong as at normal temperature. That is why the maximum operating temperature limit is usually set well below 500 F.

Engine temperatures are also very responsive to the octane rating of the fuel. To keep the temperatures below 450 F, it is desirable to use 73 or better octane fuel. This permits higher compression and thus reduces the heat that has to be dissipated by the cooling system. Also, the cooling effect of rich mixtures is very evident from the curves shown in Fig. 18.

Accessory Drives

The automotive-vehicle engine requires entirely different accessories from an aircraft engine. These might be listed as follows:

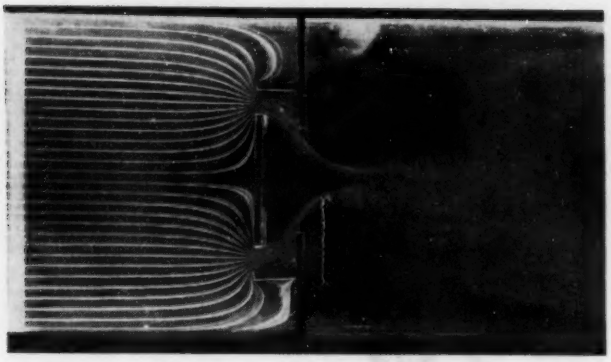
Cooling fan; float-feed carburetor; battery ignition distributor; electric generator; electric starter; oil cooler (probably); air compressor (bus engines).

Cooling fans may be centrifugal or axial flow. They may be belt or direct driven. See Figs. 19 and 20. They may be in front or behind the cylinders. They may be eliminated or assisted by an exhaust-induced vacuum such as suggested by Carl Doman.[26] One very interesting fan application to a horizontal opposed-six of 175 hp mounts it horizontally above the cylinders and drives it with multiple V belts, thus keeping the overall height of the engine unit down and permitting a sealed flywheel and clutch to be used in a conventional bell housing at the transmission end.[1]

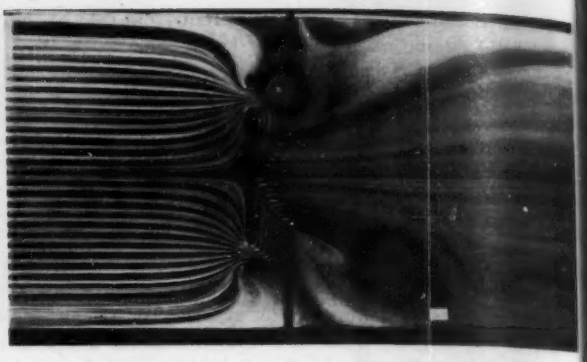
The carburetor and intake manifold should be kept out of the cooling airstream and as well insulated from the engine heat as possible if maximum power is to be obtained. It has occurred to me that with a rear-engine

[25] See SAE Transactions, Vol. 39, August, 1936, pp. 328-332: "Cylinder Temperature," by Macy O. Teetor.

[26] See "Rear-Engine Car," by Carl T. Doman. Presented at a meeting of the Metropolitan Section of the SAE, New York City, Oct. 7, 1943.



■ Fig. 21 – Photograph of experimental device used to determine the original efficiency of the airflow through the inlet tank air opening. Actual efficiency was only 26% of the actual area of the orifice—a serious bottleneck that the fan size could not overcome



■ Fig. 22 – Photograph of diverted airflow obtained by the use of turning vanes in the air inlet opening. Volume of air through the same opening was tripled without increasing the size or horsepower of the fan

mounting, the carburetor might be mounted in the body and connected to the engine by a nonmetallic flexible tube. Battery ignition presents no problems and can usually be driven off an extension of the oil pump drive shaft.

Driving a generator on the flat-type engines is not difficult if a belt is used. This same belt, if multiplied by two or three, may serve to drive the cooling fan, as is conventionally done now. A triangular drive permits a simple belt adjustment as well. Generator capacities should be from 80 to 125 amp at 12 v on engines designed for motor-coach operation. Also 10-cfm air compressors should be provided for this service.

Where engines are to be used at the rear end or beneath the floor they must be well sealed like the White Horse to protect them from dirt and mud. The crankcase ventilation also must be given careful consideration, particularly in opposed-cylinder types, where there is crankcase compression. Air should be taken as high above the road as possible, provided the ducts carrying air to the engine do not restrict it the way they do in a tank installation.

■ Aircooled Engines in Tanks[5]

The successful cooling of radial aircraft engines in our Army tanks represents the most difficult aircooling job that has been done during the war. Tank aircooled engines are completely buried so that all cooling must be fan induced. This is far worse than anything that will be encountered in any automotive vehicle. If aircooling is successful in tanks it follows that applying the same principles to motor-vehicle engines should give excellent results.

Cooling engines in tanks presents several difficult problems:

1. Both inlet and outlet air openings are restricted by military requirements.
2. All air must be circulated by a fan.
3. Air passages are not straight but very tortuous.
4. Outside temperatures in which tanks must be designed to operate are as high as 120 F.
5. No air velocity head induced by vehicle speed is available to help in the cooling.

The original 9-cyl Wright Whirlwind radial aircooled aircraft engines, designed nearly 10 years ago, have been converted for use in tanks. These engines have a bore of 5 in. and a stroke of 5.5 in., with a piston displacement of 997.3 cu in., compression ratio of 5.7 to 1 for 80-octane fuel, are supercharged, and weigh, complete with accessories, 955 lb. They are rated at 400 hp at 2400 rpm, but are held to 2200 rpm by a governor. A maximum cylinder-head temperature of 450 F and a cylinder-barrel temperature of 325 F is specified. The original engine had steel cylinder barrels with turned steel cooling fins.

With 5.7 to 1 compression ratio it is possible to run this engine on 80-octane fuel and obtain 300 bhp.

The tank engine cylinders have aluminum heads, shrunk and screwed onto finned steel cylinder barrels. This makes it possible to shrink an aluminum muff over the cylinder barrel and thus increase the cooling of the barrel. Although not yet in production, Continental Motors Corp., who are now manufacturing this Whirlwind tank engine, claims that the aluminum muff increases the continuous maximum power available over 20% at the same engine speed. Changes are also made in valve timing, and necessarily certain parts are strengthened as might be expected with this steady output.

According to the Wright Aeronautical Corp., these tank engines require 12,000 cfm of air to cool properly under full load. As first installed in tanks, Barber-Coleman tests showed that only 6250 cfm of air passed through the tank at full load. With the corrective devices they installed and an improved fan, the volume of air was approximately doubled. Furthermore, all the additional equipment had to be designed so it could be applied to tanks already built and in the field.

Lessons learned in properly cooling tank engines should be of inestimable value to anyone handling a motor-vehicle aircooled-engine installation. Three basic problems had to be solved, each of which should be considered in aircooling an automotive-vehicle engine:

1. Obtaining the necessary volume of air in circulation over the cylinders.
2. Methods of increasing the cooling area of the fins on the cylinders.
3. Methods to increase heat transfer to cooling air from available cooling surfaces.

Increasing air volume usually means providing a larger fan. This absorbs more horsepower and reduces the net horsepower available at the wheels. Although increasing the cooling area adds to the cost of the cylinders, it does not reduce the net horsepower like a bigger fan, but may actually decrease the horsepower required.

In the tank engine application, according to V. R. Abrams, one of the engineers at the Office of the Chief of Ordnance-Detroit, it was found that only about half the air required was obtained. To correct this lack of air volume every parasitic impediment to airflow should be simultane-

ously improved. Take for example the orifice through which the air entered the tank, which was a square hole with a table-top cover. Due to the orifice characteristics of such an opening in a plate, the orifice coefficient was reduced to approximately 0.3.

This is clearly shown in Fig. 21 where an experimental test set-up was photographed. To correct this, the Barber-Coleman Co. development engineer, W. W. Kennedy, introduced turning vanes in this aperture, with the results shown in Fig. 22.

This principle was used throughout the engine compartment. It was found that using turning vanes where necessary had the same effect as if the area of the passages had been increased. The combination of all such devices increased the total airflow 79% and heat abstracted by more than 65%.

Air was found to leave the fan in a great swirl – tests in these tank installations showed that it made as much as 2½ complete turns between the edge of the fan blade and the cylinder cooling fins. Then it struck them at an angle of about 30 deg to the plane of the cylinders.

These experiments also indicate how important it is to analyze the individual cooling of each cylinder, as this swirl can cause unpredictable inequalities. Some of the inequalities were indicated by the fact that when over 60% more heat was being taken from the engine the rear spark-plug gasket thermocouples only indicated 1 deg less cylinder-head temperature. This proves how inadequate conventional instrumentation may be and how misleading.

This led to more complete tests, which were run by the Ethyl Corp. with 13 thermocouples on each of Nos. 3, 5, and 7 cylinders. Analysis of these tests indicated the difference in individual cylinder temperatures. From five thermocouples equally spaced about the midpoint of a cylinder barrel, the importance of controlled airflow is even more clearly indicated. Other points as indicated by the legend at the bottom of the temperature curve sheets show the variation between different points on each cylinder.

Temperature comparisons made between three of the nine cylinders in an experimental tank mock-up installation are shown in Figs. 23 to 27. These data show how important it is to check *every* cylinder at many points on an engine.

The effect of imposing air deflectors at many points between the fan and the cylinders and an adequate quantity of cooling air are shown in Figs. 23, 24, and 25. They show how much had still to be done to equalize the temperature and how important this kind of research is to the successful operation of any aircooled engine.

The purpose of the turning vanes between the fan and the cylinders brings out one important factor which may have been overlooked in the past. When air is directed straight against the cylinder, there are two hot spots on each side, as can be seen in Fig. 28. These are in dead air spaces.

If the air impinges on the cooling fins at an angle, the dead air space is eliminated. With test thermocouples it can be shown when and how much the angular displacement must be to give the best results.

These fins, or "fix" as they were called, also take the 2½ twist out of the air as it leaves the fan and redirect it so as to obtain the most uniform cooling effect on the cylinder barrel.

Before a "fix" was used in tank-engine installations, the rocker boxes and cylinder heads were well cooled. Total pressure heads up to 13.5 in. of water were obtained across the rocker boxes while the cylinder barrels were surrounded by returning eddies of already heated air. Result was overheated barrels and stuck pistons.

Table 8 – Performance Figures

	Standard Engine	Special "Fix" Cooled Engine
Duration of Run, hr	50	200
Fuel, ASTM octane number	80	79.9
Average Hp observed	309	290
Fuel, lb per hr	241	238
Specific Fuel Consumption, lb per Hp per hr	0.78	0.82
Average Oil Consumption for test, lb per hr	3.46	1.2
Blowby Rate at Start, cfm	3.8	2.4
Blowby Rate at Finish, cfm	5.1	2.9
Rear Spark-Plug Gasket Average Temperature, F	446	428
Rear Cylinder Base Flange Average Temperature, F	350	326
Average Temperature at Midpoint of Barrel, F	334	299
Loss of Piston-Ring Weight, mg	602.3	148.0

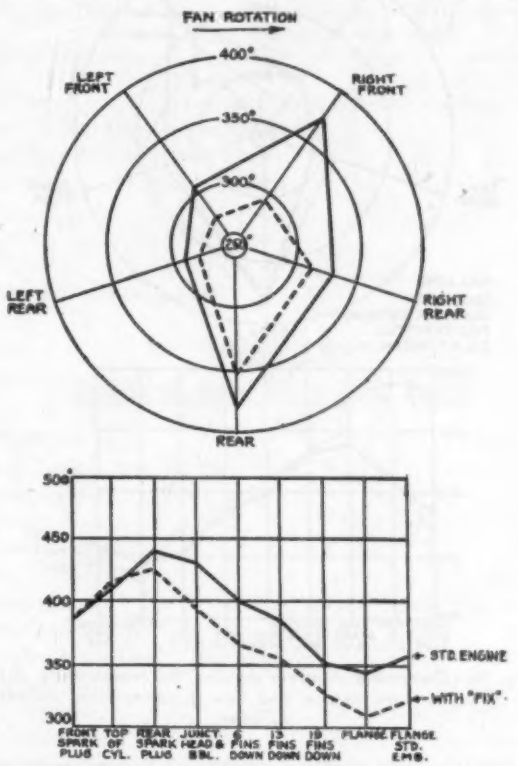
Before the "fix" was applied, the pressure head (in. of water) at the top of the cylinder barrel, at the base and at two points in between was as follows:

	Head			Base
Before "fix"	7.9	5.3	4.7	4.2
With "fix"	4.4	4.6	6.0	6.0

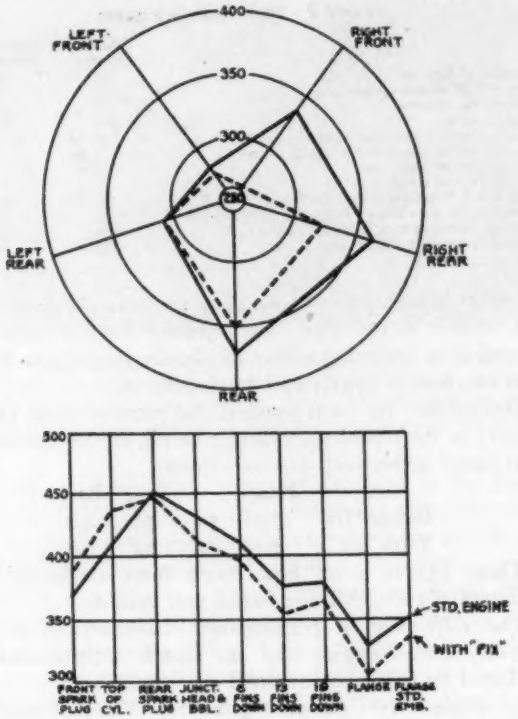
These figures show how much more uniformly and preferentially the cylinder barrel was cooled.

The difference in performance characteristics with a properly cooled engine and one insufficiently cooled are indicated by some experimental test results.

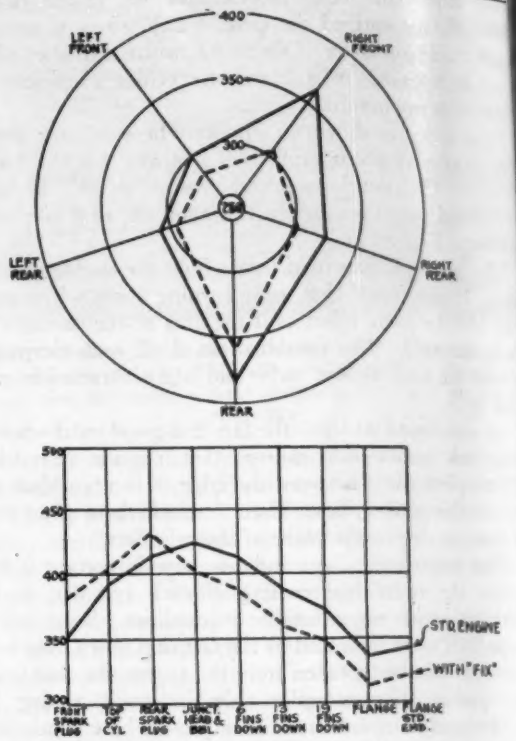
Six engines were put on a 50-hr test. The best standard installation, one which ran 50 hr, is compared with the "fix" cooled engine after the latter had run 200 hr. Both tests were run at 2200 rpm with full throttle and with air entering the engine compartment at 100 F. See Table 8.



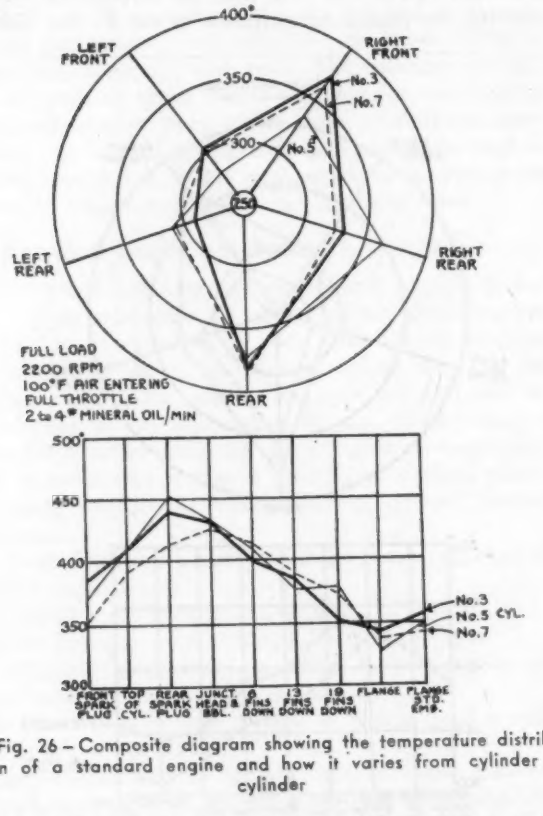
■ Fig. 23 – Distribution of temperature about No. 3 cylinder of an early tank engine running at 2200 rpm at full throttle. The dotted line shows the temperature range when the first Barber-Coleman "fix" was used between fan and cylinders



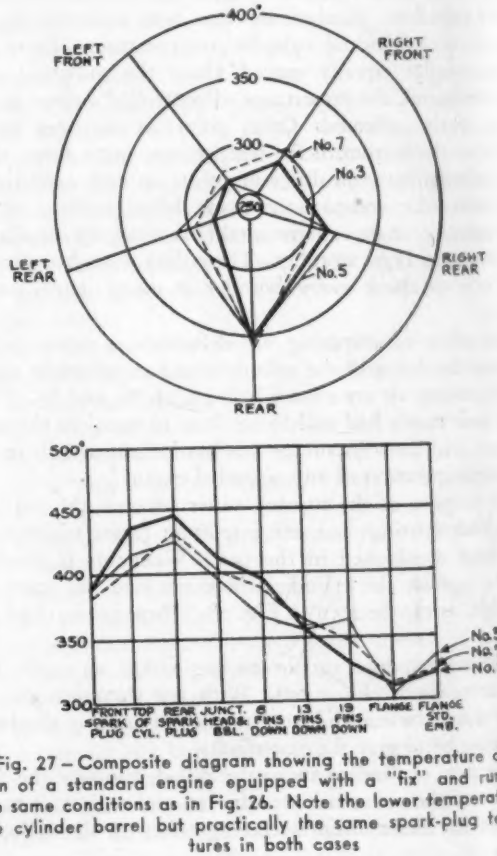
■ Fig. 24 – Temperature distribution on No. 5 cylinder made during the same test as that for Fig. 23



■ Fig. 25 – Temperature distribution on No. 7 cylinder made during the same test as that for Fig. 23



■ Fig. 26 – Composite diagram showing the temperature distribution of a standard engine and how it varies from cylinder to cylinder



■ Fig. 27 – Composite diagram showing the temperature distribution of a standard engine epuipped with a "fix" and run under the same conditions as in Fig. 26. Note the lower temperatures on the cylinder barrel but practically the same spark-plug temperatures in both cases

The differences in cylinder-head and flange temperatures as measured were only 18 F and 24 F, respectively, and are really no criterion of the difference in the overall cooling efficiency of the two engines. The total heat abstracted by the air was 60% more with the 200-hr engine. That the cylinder and piston-ring wear was greatly reduced is evident from a comparison of the loss of weight of all

piston rings in the two engines – actually on a per-hour basis there was a difference of 16 to 1 in favor of the "fix" cooled engine.

Conclusions

Engine cylinder temperatures need not be any higher than with water-cooled engines.

Aviation-engine aircooling has advanced far beyond the needs of automotive installations, so that it is only a case of building engines to meet them.

The road-load characteristic of passenger-car engines are so much less than those of aircraft that it may be possible to cool engines by exhaust-created suction instead of a fan.

Aircooled engines now have quiet valve gears since hydraulic lifters are used.

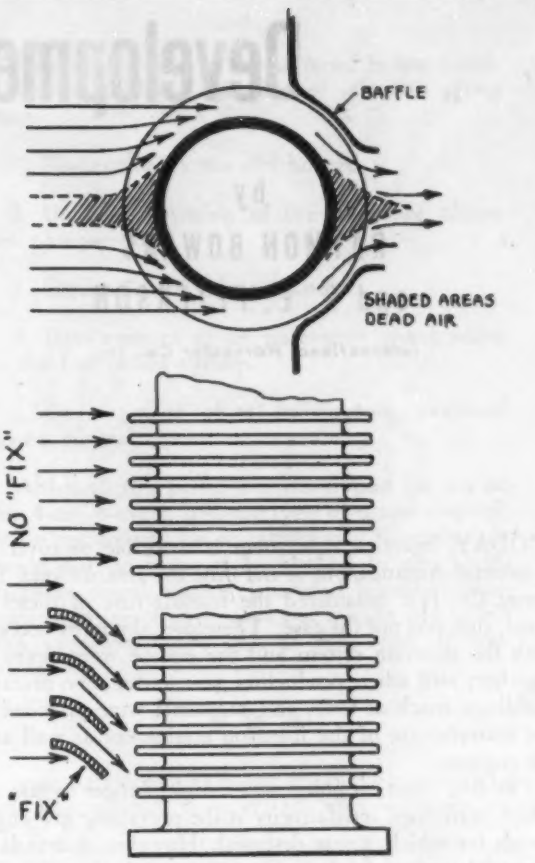
Aluminum cylinders, heads, and pistons must be used in any successful aircooled automotive engine, and the price of aluminum castings will determine whether aircooled engines can be built economically after the war.

The aircooled engine lends itself to rear-engine installations because it simplifies the installation due to the simplicity of the cooling system.

Aircooled aluminum engines are lighter than liquid-cooled engines of equal horsepower. This is an important contribution to reduced vehicle weight, better fuel economy, and performance.

Aircooled engines are desirable in climates where there are severe winters and a vehicle has to be kept in constant service.

Aircooled engines are even more desirable in the desert where water is scarce.



■ Fig. 28 – Diagram showing the advantage of directing the air at an angle on the cooling fins of a cylinder

APPENDIX I
Specifications of German Volkswagen[4, 5]

Wheelbase, in.	94.5
Tread – Front, in.	51.0
Rear, in.	49.3
Clearance, in.	8.6
Weight, lb	1430
Top Speed, mph	62
Fuel Consumption, mpg	27.7
Fuel Tank Capacity, gal	5.5
Rear Suspension	Swiveling axles Torsion rods Oscillating levers
Front Suspension	Dual torsion rods Dual oscillating levers
Battery Ignition	6 v, 75 amp
Solex Down-Draft Carburetor	
Tire Size, in.	4.50 x 16
Bhp	23.5 at 3000 rpm
Bore and Stroke, in.	2.76 x 2.52
Compression Ratio	5.6
Piston Displacement, cu in.	60
Overhead Valves	
Aluminum Cylinder Heads in Pairs	
Cast-Iron Individual Cylinders	
Aluminum-Alloy – 3-Ring Pistons	
Cooling by Belt-Driven Centrifugal Fan	
Copper-Lead Main Bearings	
Full-Pressure Oiling with Oil Cooler	
3 Main Bearing Crankshaft – No Counter Balances	
Clutch	Single dry plate
Gear Box	4 speeds and reverse – constant mesh
Two Helical Gear Noiseless Speeds	
Gear Box Ratios: 0.80 to 1 high	
1.25 to 1 third	
2.06 to 1 second	
3.60 to 1 low	
6.60 to 1 reverse	
Bevel Gear Ratio: 4.43 to 1	

APPENDIX II
Specifications of White Horse Truck[9]

Wheelbase, in.	104 or 121
Fuel Tank Capacity, gal	11
Brakes, Front and Rear, in.	14 x 2
Springs, Front	36 x 2 in. semi-elliptics
Rear: Coil	
Battery Ignition	Auto-lite
Battery	6 v, 17 plate, 120 amp-hr
Generator, amp	32
Single Down-Draft Carburetor	
Cooling Fan	12 in. Sirocco
Location	On front of crankshaft
Capacity	2400 cu ft at 2600 rpm
Tire Size, in.	7.00 x 16
Bhp	40 at 2600 rpm (governed)
Bore and Stroke, in.	3.62 x 3.50
Compression Ratio	5.7

concluded on page 302

Development of a SINGLE-PLUNGER INJ for Four-Cylinder Dies

by
RAYMON BOWERS
and R. E. PETERSON
International Harvester Co., Inc.

TODAY, injection equipment is available to cover any normal demands, but at the time the International Harvester Co. first considered the manufacture of diesel engines, that was not the case. Therefore, almost of necessity, both the injection system and the engine were developed together; and when production was started, the necessary buildings, machine tools, and personnel were provided for the manufacture of the injection equipment as well as of the engines.

The first injection pump was a multiplunger bypass unit which performed satisfactorily at the prevailing low engine speeds for which it was designed. However, demands for increased engine speeds and a wider range of engine sizes soon put injection requirements beyond the scope of this original pump, and it became necessary to consider a new design. The demands called for an injection pump relatively small in size, low in cost, and adaptable to a considerable spread of engine sizes and speeds. It was also necessary that the pump be serviceable and capable of handling a variety of diesel fuels.

A thorough study was made of various types of injection systems before establishing the basic design believed best suited to our requirements, and a decision was finally made to concentrate efforts on a single-plunger pump for 4-cyl engines. There were three basic reasons for selection of the single-plunger type of pump:

1. It simplified construction through elimination of three precision plungers and barrels.

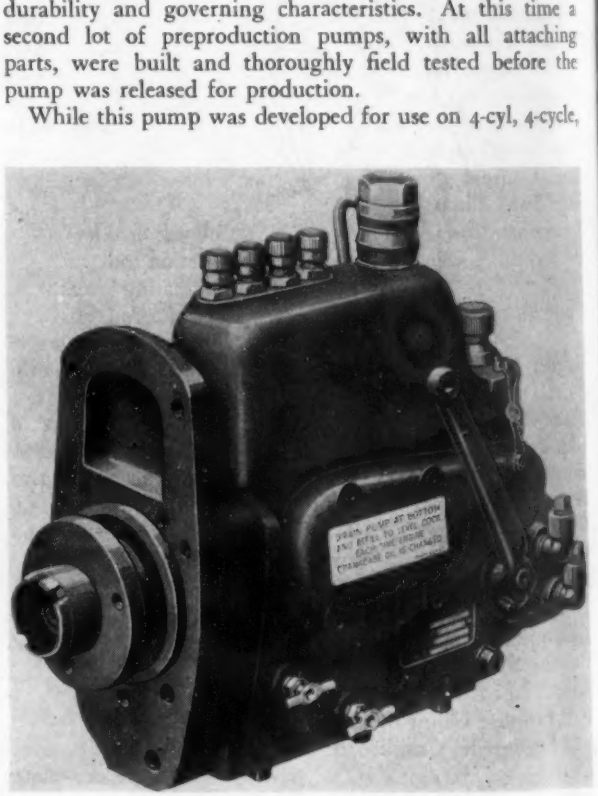
2. It eliminated the very difficult job of building and adjusting four plungers so that each would deliver exactly the same amount of fuel per stroke at all operating speeds and fuel settings.

3. It simplified the governor problem by reducing the amount of work imposed on it.

Two types of single-plunger applications were investigated. The first type used the full stroke of the single plunger to deliver fuel to a chamber kept under high pressure by a spring-loaded piston. This fuel, under pressure, was delivered to the nozzle when the distributor valve opened. The quantity delivered was controlled through a governor and linkage, which maintained the pressure in the chamber until the required amount of fuel had been delivered to the nozzle. This principle was abandoned for the type now in production, which uses a conventional plunger with a helix for varying the fuel delivery per stroke. In this case, the distributor valve opens several degrees before the start of injection, and the plunger meters and delivers fuel directly to the nozzle.

It required approximately four years of experimental work before the pump was sufficiently developed to warrant consideration for production. At that time 25 preproduction pumps were built and shipped to various locations throughout the country for field tests under all kinds of operating conditions. Results of these tests indicated a need for increased ruggedness in construction, improved fuel filter equipment, and an overload governor control which would give the lugging characteristics required from engines when overloaded and pulled under speed. It required approximately two more years to obtain acceptable durability and governing characteristics. At this time a second lot of preproduction pumps, with all attaching parts, were built and thoroughly field tested before the pump was released for production.

While this pump was developed for use on 4-cyl, 4-cycle,



[This paper was presented at the SAE War Engineering-Annual Meeting, Detroit, Mich., Jan. 13, 1944.]

NJECTION PUMP

Diesel Engines

THE major problems encountered in the development of the single-plunger injection pump were:

1. Elimination of the dirt hazard.

2. Uniform charging of the chamber above the plunger.

3. Elimination of exhaust smoke.

4. Development of an automatic check valve in the fuel return system.

5. Development of a satisfactory overload speed control.

Although this pump was developed for use on the 4-cyl, 4-cycle, pre-cup type of diesel engines, there appears to be no reason why this principle could not be successfully applied to any multicylinder diesel engine, provided injection pressures and injection strokes per minute are within the practical limitations of this type of construction.

■ ■ ■

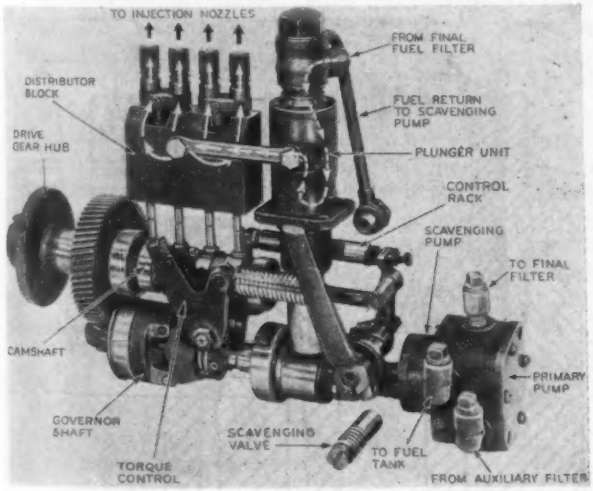
THE AUTHORS: RAYMON J. BOWERS (M '30) began his commercial career in 1928 when he became connected with International Harvester Co., for whom he has worked ever since. Specializing in research and development, he is now in charge of internal combustion engine development. He spent two years as an instructor at the Agricultural College, University of Minnesota, where he had graduated with a B. S. degree in mechanical engineering. RAYMOND E. PETERSON has been with the International Harvester Co. for the past 26 years, the last 10 of which he has been in charge of design and development of diesel injection pumps. He is also consultant manufacturing engineer at the Milwaukee Works on injection pumps.

pre-cup type engines having relatively small displacement and operating up to 1800 rpm, the single-plunger principle is not necessarily limited to 4-cyl engines, but could be used on any similar multicylinder engine, provided the maximum number of injection strokes per minute was not beyond the practical limitations of a single plunger.

Description

The single-plunger pump (Fig. 1) is flange mounted on the crankcase front cover. It is completely enclosed and is driven by a gear on the pump camshaft. A single plunger meters and delivers all the fuel to the engine through a distributing system consisting of four cam-operated distributor valves. The manner in which the fuel is metered and delivered is basically the same as in the conventional multiplunger pump except that the single plunger delivers all the fuel. The governing is controlled by a flyball governor through a simple linkage which rotates the plunger in its barrel. See Fig. 2.

For convenience in describing the pump, the main assemblies and parts which are mounted on, or installed within, the main pump housing will be discussed separately. The plunger unit, the distributor unit, the primary and scavenging pump assembly, and the mounting flange by which the pump is attached to the engine, and in which the camshaft and governor shaft gears are enclosed, are all mounted on the pump housing. The camshaft, governor shaft, governor fork shaft, distributor valve tappets, and other essential parts are installed within the pump housing.



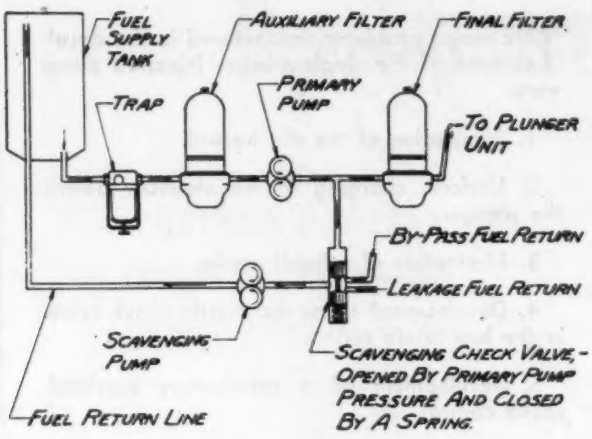
■ Fig. 2 – Skeleton view of pump mechanism

■ Fig. 1 – Single-plunger injection pump

The plunger unit includes a single conventional helical grooved plunger, operated in a two-port plunger barrel by an eccentric on the end of the governor shaft, a check and reverse check valve above the plunger, a bypass valve on the primary fuel side and other parts to make it a complete assembly. This unit is located at the top of the pump housing.

The distributor valve block assembly holds the four valves which distribute the fuel to the various cylinders. This unit is assembled on the top of the pump housing, in front of the plunger unit, and is connected to it by a high-pressure pipe.

A primary pump and a scavenging pump of conven-



■ Fig. 3 – Primary fuel supply system

tional gear type are assembled as a complete unit in the same housing at the rear of the main pump body, and are driven by a common shaft. The primary pump, by means of a pressure-regulating valve, maintains practically constant pressure on the supply fuel at all speeds and loads. The scavenging pump returns to the supply tank, all fuel bypassed at the metering assembly and the leakage fuel which escapes past the plunger and distributor valves. In connection with the scavenging pump, there is also a scavenging valve which is installed in the leakage passage in the main pump housing. This valve automatically opens when the pump is in operation, permitting leakage fuel to be returned to the fuel tank, and automatically closes when the pump is not in operation, preventing fuel from the supply tank backing up and flooding the pump sump.

The camshaft operates the distributor valves and drives the governor shaft. It is installed in the top of the main pump housing and extends through the pump flange, which is mounted on the front end of the pump housing.

The governor shaft carries the governor weight assembly, operates the plunger by means of an eccentric, and drives the primary and scavenging pumps. It is installed in the main pump housing beneath the camshaft and is driven at four times camshaft speed.

This briefly describes the single-plunger pump, and now an explanation of its operation will be given by tracing the fuel system.

■ Operation

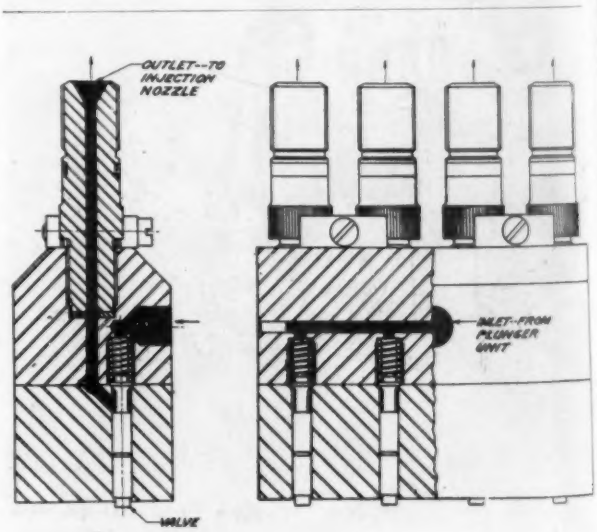
The fuel (Fig. 3), on leaving the supply tank, passes through a trap which removes water and the coarser particles of foreign matter. From the trap the fuel goes through an auxiliary filter element, where additional filtration takes place, and then through a filter screen at the inlet to the primary pump. It then passes through the primary pump where it is put under light pressure, and on through a final filter element and into the plunger unit assembly. Here it goes through a second filter screen before passing into the reservoir surrounding the ports in the plunger barrel. This amount of filter equipment may seem to be more than is necessary to those not familiar with the dirt conditions encountered in tractor operation, but experience has shown that the dirt hazard, even under what most operators consider good fuel handling conditions, is a serious problem. There is a pressure-regulating valve in the primary pump which maintains practically constant pressure on the supply fuel at all times. This is accomplished through bypassing all fuel not required by the engine or bypassed at the plunger unit, back into the inlet of the primary pump.

Primary pump pressure charges the chamber above the plunger when the ports are uncovered. In order to obtain uniform charging of this chamber at all speeds and fuel settings, it was necessary to introduce a bypass valve in the fuel supply system. This bypass valve is located at the top of the plunger unit, and the bypassed fuel is returned to the scavenging pump and thence to the supply tank. After the chamber above the plunger has been charged and the plunger is starting its upward travel, a cam on the camshaft raises a distributor valve which opens a direct passage between plunger unit and a nozzle in the cylinder head.

Fig. 4 shows a section through the distributor block assembly.

Injection starts when both ports of the plunger barrel have been covered by the plunger, the check valve above the plunger raised off its seat, and the fuel put under sufficient pressure to open the nozzle valve. Because of a ⅛-in. helix running halfway around the top of the plunger and on the side opposite from the helix which ends injection, one port starts injection, and the other ends injection. When the lower helix on the plunger uncovers its port, injection ends, and the check valve above the plunger returns to its seat. To prevent fuel dribbling from the nozzle after injection ends and causing exhaust smoke, a reverse check valve is built into the check valve assembly above the plunger. This reverse check valve opens when injection ends and drops the line pressure existing in the system between the check valve and nozzle to a pressure below the opening pressure of the nozzle, by discharging fuel back into the chamber above the plunger.

Here it would be well to explain that, although this plunger does the work of the four plungers in the conventional-type injection pump, it does not operate four times as fast. It so happens, that after determining the best

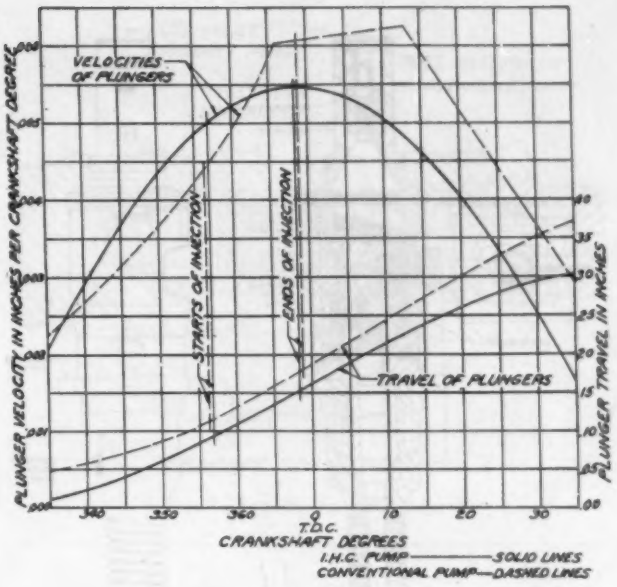


■ Fig. 4 – Distributor block assembly

plunger velocity for injection by actual test on any particular application, results showed this velocity to be essentially the same as that of a plunger of similar size in a conventional pump operated under the same conditions. The main difference between the plunger action in the single- and multiplunger types of pump is that the single plunger is going up and down continuously, whereas in the other case those plungers not delivering fuel remain motionless on the heels of their respective cams.

It should also be pointed out that the simple harmonic motion obtained with an eccentric is less severe on the plunger and plunger return spring than is the cam action of the conventional pump. See Fig. 5. Factual proof of this is substantiated by the fact that breakage of plunger return springs on the single-plunger pump is unknown, despite the fact that this spring does the work of four springs in a multiplunger pump.

Leakage fuel which escapes past the distributor valves and plungers gathers in a reservoir and is drawn off through the scavenging valve and returned along with the bypassed fuel to the supply tank. Because the scavenging pump must have excess capacity above that of the leakage and bypassed fuel, a small amount of air is taken from the clean side of the engine air cleaner, through an air filter into the injection pump, where it is returned with the leakage and bypassed fuel to the supply tank. The scavenging valve is opened by primary pump pressure when the pump is in operation and is closed by a spring when operation ceases. A filter screen is installed in the leakage passage ahead of the scavenging valve for its protection. In brief, this covers the operation of the single-plunger pump. The major problems encountered and their solution will now be discussed.



■ Fig. 5 – Plunger velocity and travel curves

Problems Encountered

■ Fuel Filter Equipment

One problem which became apparent from the very start of our field tests was excessive wear of some of the delicate parts of the injection pump. The answer was, obviously, better fuel filter equipment. So an engine was set up with a conventional fuel system, except for the introduction of a means of constantly agitating the fuel in the main tank to keep in suspension fine abrasive material which was added to the fuel.

With this set-up and through the close cooperation of filter manufacturers we were able to work out adequate protection. This test method proved itself most valuable in two other ways: first, accelerated wear tests could be made that would point out in a few hours of operation certain weaknesses which, otherwise, would have taken weeks or even months to discover; and, second, it proved exceptionally valuable in selecting materials least subject to abrasive wear. With this set-up, but without adequate filter equipment, we could definitely impair engine performance in 8 hr and in most cases practically wear out an injection pump in 25 hr.

■ Uniform Fuel Delivery

To obtain uniform fuel delivery from all four nozzles at all operating speeds and fuel settings was another problem requiring considerable development. All of this work was done on a test bench with the injection pump driven by a variable-speed electric motor. Delivery through nozzles into glass graduates, with facilities for instantly switching to or from these graduates, made it possible to check uniformity of fuel delivery between nozzles at all speeds and fuel settings. From the beginning, fairly uniform delivery between nozzles was obtained through the lower speed range at all operating fuel settings. However, this was not the case at higher speeds where wide variations would occur in fuel delivery to the four nozzles.

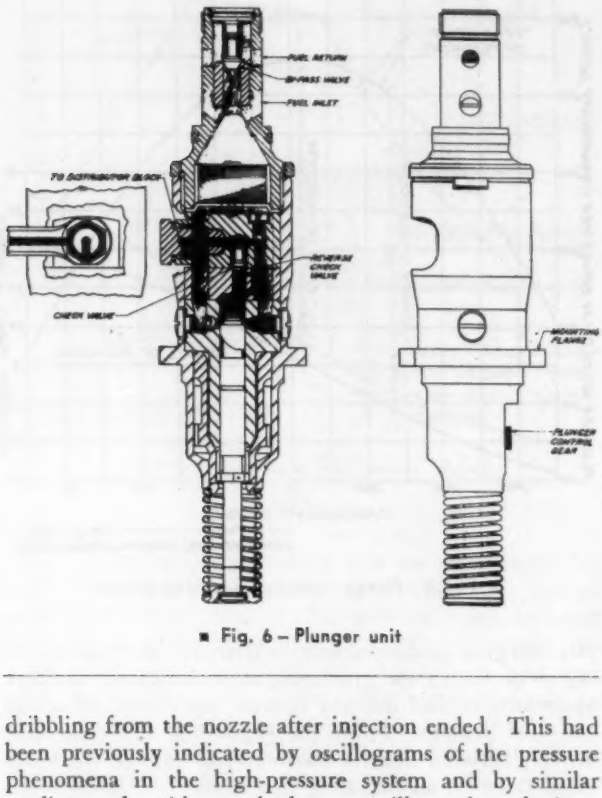
The reason for this variation was found to be the non-uniform charging of the chamber above the plunger. This condition, in turn, was caused by a pressure wave set up by the back surge of fuel from the injection line when the plunger uncovered the port at the end of injection. Pressures in the chamber above the plunger during injection range between 2000 and 4000 psi, depending upon operating conditions.

When port opening occurred, the primary fuel supply system carrying a pressure of approximately 35 psi was suddenly subjected to the high pressure existing above the plunger. This disturbed the charging through the higher speed range to such an extent that it was possible to start with uniform delivery between nozzles, and by gradually increasing the pump speed, two nozzles (every other one) would deliver more and more fuel per stroke, while the other two would deliver less and less, until all the fuel was delivered by two nozzles.

A synthetic rubber diaphragm installed on the supply pump side in the plunger unit assembly entirely corrected the trouble, and uniform delivery was obtained from all four nozzles throughout the speed and fuel delivery range. However, because of the questionable durability of synthetics at that time, plus the need for bypassing a certain amount of fuel to vent the injection system and to lubricate the scavenging pump, a bypass valve and spring were used, which served the same purpose.

■ Reverse Check Valve

Another important problem was the elimination of exhaust smoke. High-speed motion pictures of combustion in the pre-cup showed that the trouble was caused by fuel



■ Fig. 6 – Plunger unit

dribbling from the nozzle after injection ended. This had been previously indicated by oscillograms of the pressure phenomena in the high-pressure system and by similar studies made with a cathode ray oscillograph and piezo-electric crystal.

Since our injection line pressures range between 2000 and 4000 psi and our nozzles are set to open at 700 psi, the high pressure, which existed in this line after the opening of the port and the closing of the check valve, caused the nozzle to dribble until line pressure and nozzle opening pressure were equalized. Dribbling was corrected by the introduction of what we call a reverse check valve to relieve the pressure in this line. See Fig. 6. Various types of masked or displacer valves were tried instead of the check valve above the plunger, but exhaustive tests covering a wide range of speeds and fuel deliveries showed a distinct advantage in dropping the line pressure to a constant predetermined level by using a reverse check valve. The instant injection ends the reverse check valve drops this pressure, to approximately 600 psi (that is, 100 psi lower than the opening pressure of the nozzle), thus preventing any dribble. High-speed motion pictures of combustion, with and without the reverse check valve, clearly showed how the action of the reverse check valve eliminates dribble. Figs. 7A and 7B show enlargements taken from motion pictures with and without reverse check valve.

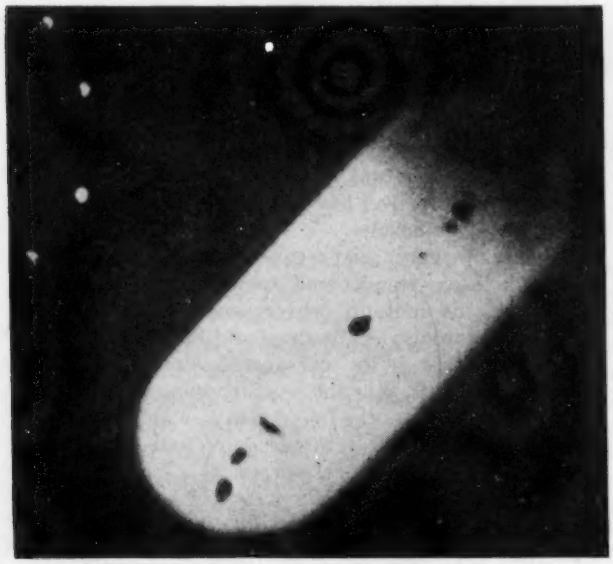
■ Scavenging Check Valve

Another requirement was a satisfactory check valve in the return fuel system. The scavenging pump was originally designed to return only the fuel that leaked past the distributor valves and plunger. However, this leakage was such a small amount that it did not adequately lubricate the scavenging pump. A method of lubricating the scavenging pump was provided by the necessity of using a bypass valve in the metering assembly to obtain proper charging of the chamber above the plunger. The problem then was to work out a check valve that would prevent fuel from backing up through the return fuel system and flooding the pump sump when the engine was not in operation and which would open when the pump was in operation to permit return of leakage fuel to the supply tank.

In the final construction a lapped piston and barrel were installed in the leakage passage. See Fig. 8. This piston, or scavenging valve, is opened by primary pump pressure when the pump is in operation and is closed by a spring whenever operation stops.

■ Governor and Torque Control

Probably the most difficult problem of all was to develop a satisfactory overload speed control, which we call a torque control. To work out a flyball governor to give



■ Fig. 7A – Enlargement of high-speed motion picture of combustion shows resulting dribble when no reverse check valve is used.

■ Fig. 7B – Enlargement of high-speed motion picture of combustion shows no dribble when reverse check valve is used

the right amount of control rack motion and to give acceptable speed control from full load to no load was relatively simple and easy. The conventional type of flyball governor was used, in which the governor energy, balanced by springs, is transferred to the plunger by means of a governor sleeve, fork, and rack. The governor fork, mounted on a cross-shaft, is held against the governor sleeve by the governor springs and is attached to the rack, which rotates the plunger whenever the governor weights move together or apart. However, to work out a torque control which would give the required increase in torque with a decrease in speed for satisfactory overload lugging characteristics, was not a simple matter. In general, the overload performance calls for a gradual increase in torque with a decrease in speed until at peak torque the engine is pulling 12 to 14% more torque than at full load, at a speed of from 40 to 50% that of the full-load governed speed.

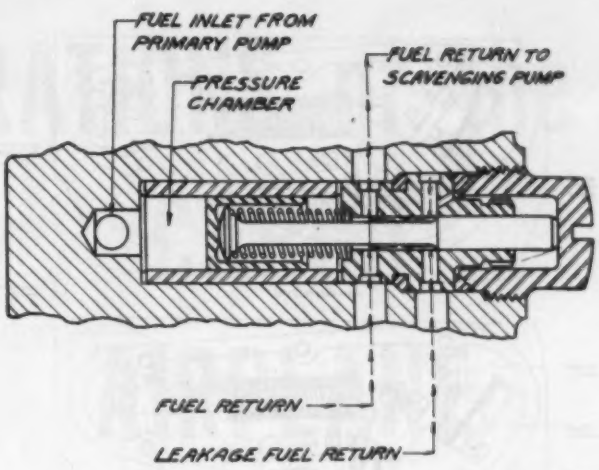
The fuel delivery characteristics of the pump, in terms of fuel per stroke for any fixed governor setting, is approximately constant throughout the speed range. So, it becomes obvious that to increase the torque of an engine with a decrease in engine speed would require some automatic control which would properly increase fuel delivery per stroke with a decrease in engine speed until maximum torque was reached.

The problem was clear cut as far as what had to be done. The torque control, beside governing the engine from full load to peak torque, had to be simple, reliable, and durable. It was also important that it be confined within the pump and flexible from an adjustment standpoint to meet special engine requirements, as well as variations found in the general run of production engines and injection pumps.

Our first problem, of course, was to work out a torque control regardless of size or kind that would automatically give the required engine performance between full load and peak torque. Many methods were tried, among them magnetic and hydraulic torque controls. However, it became apparent that as long as we used a flyball governor for part-load control, the most satisfactory plan would be to use this same governor for overload control.

An analysis of this problem from a governor standpoint showed that to do this with a flyball governor would require a mechanism in which the spring force would always be in balance with the energy of the governor weights throughout the speed range from full-load speed to peak torque, and also that the governor weights would move together, or apart, the right amount to increase or decrease fuel delivery per stroke to the engine with each change in speed. Since the energy of a flyball governor varies as the square of the speed and, since the engine speed at peak torque is from 40 to 50% of full-load speed, the governor energy at peak torque would be ¼ or less than that required at full load.

To repeat, specifically, this device in governing between full-load and peak-torque speeds would have to move the governor weights together, or apart, the right amount to increase or decrease fuel delivery per stroke for each change in speed. In order to do this it would have to increase or decrease the spring force which balances the force of the governor weights: first, to meet their rapidly changing power which follows each change in speed; and, second, to meet their change in power due to moving together or apart.

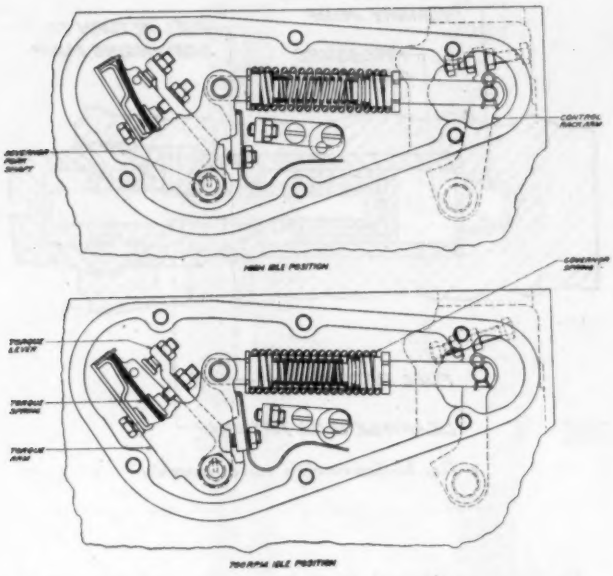


■ Fig. 8 – Scavenging valve assembly

To those uninitiated in the mysteries and vagaries of flyball governors, this may seem a simple problem; but much thought and effort were required before a torque control was developed which fulfilled all requirements. The complete governing problem for both part-load and overload was finally solved by making use of two sets of springs; one set of regular governor springs to control engine rpm from governed speed full load to high idle no load (and slow idle), and another set of torque springs to govern between full-load and peak-torque speeds.

The torque control as finally developed made use of a set of flat leaf springs and a specially shaped seat. The principle upon which it works is very simple. As the torque springs are pressed tighter against their seat their effective length decreases and consequently their rate increases. In this manner it was possible to vary the torque spring rate and thus balance the rapidly changing governor energy as the speed changes between full-load and peak-torque speeds, and at the same time move the governor weights the right amount to vary fuel delivery per stroke to meet engine requirements.

The torque springs and seat are installed in a torque arm which is mounted and free to oscillate within limits on the governor fork shaft. The regular governor springs are attached to this torque arm. The energy from the governor weights is transferred to the torque springs in the torque arm through a torque lever keyed to the governor fork shaft. Since the torque arm can oscillate within limits on the governor fork shaft the torque lever can oscillate in the torque arm within these same limits. At full-load governed speed, the governor springs hold the torque arm lightly against a stop, and the torque springs are compressed a maximum by the torque lever, which is also resting lightly against a stop in the torque arm. If the engine speed increases, the energy of the governor increases and pulls the torque arm away from its stop; and so the torque lever (already against its stop in the torque arm) and the torque arm act as a single lever. Thus the torque springs, although compressed to a maximum, are not in action whenever the engine is operating between full load and no load. Likewise when the engine is operating between full load and peak torque, the governor



■ Fig. 9 – Torque Control Diagrams

springs are not in action but the torque springs are. The reason for this is that at full load the torque arm is held lightly against a stop, and any decrease in speed does not permit any change in governor springs except to hold the torque arm tighter against this stop. However, since the torque springs compressed by the torque lever are balancing the governor energy at full-load speed, when the engine speed drops causing the governor to lose energy, the torque springs move the torque lever back until torque spring force and governor force balance one another. This increases fuel delivery per stroke to the engine and is possible because the torque lever is free to oscillate within the torque arm. Any further decrease in speed causes additional backward movement of the torque lever and a further increase in fuel delivery until peak torque is reached, at which point the torque lever strikes a stop and no further increase in fuel is possible. With an increase in speed, of course, the reverse action takes place until full-load speed is reached.

The torque control mechanism is simple and easily adjusted to meet varying engine requirements as Fig. 9 shows.

■ Conclusion

In conclusion it can be stated, that although the single-plunger pump was developed for use on 4-cyl, 4-cycle, pre-cup type diesel engines, there appears to be no reason why this principle could not be successfully applied to any multicylinder diesel engine or injection system, provided the maximum number of injection strokes per minute or the injection pressures were not beyond the practical limitations of the single-plunger construction. This pump is already supplied in production for a rather wide range of engine sizes for speeds up to 1800 rpm and, with minor changes, tests have shown that it meters satisfactorily up to 3000 rpm engine speed.

DISCUSSION

Outlines Specifications Of Ideal Injection System

—Ralph Miller

Worthington Pump & Machinery Corp.

A SINGLE pump and distributor injection system identical to that described by Mr. Bowers was used on the first high-speed diesel engine built for switching locomotives in this country. The fuel pump was eccentric driven and operated at 1800 strokes per min. A "Geneva motion" operated disc was used on the early engines but was later replaced with a poppet-valve distributor of a compact design in which all valves were operated by a single cam.

While this system may be satisfactory for engines of small dimensions such as the types built by International Harvester Co., it was found that when the fuel lines are increased in length pressure waves and resulting after-injections together with compressibility of the hydraulic system renders it unsuited for larger engines. When the engine referred to above was developed to operate at 750 rpm, the distributor system had to be abandoned in favor of a system using individual pumps with the shortest possible fuel line. Open-type nozzles were used in this engine, and my experience with the pump displacer valve precedes but parallels the work of Mr. Bowers. To avoid after-injections a return check valve such as described in a patent issued to me under No. 1,999,967 had to be used instead of the standard displacer valve.

I call attention to the fuel curve shown in Fig. 17 of E. C. Magdeburger's paper,[1] and express the belief that the altogether disproportional increase in fuel consumption with increase in speed shown on this curve is largely brought about by improper functioning of the injection system. None of the accepted injection systems used today possess the essential characteristics needed for high-speed operation. While the unit injector represents a step forward, the unavoidable shock loads and resulting deflection in the operating mechanism makes this arrangement unsuitable for installation directly in the cylinder head.

It is my belief that the ideal injection system, if and when it is developed, will be designed to generate injection pressures in the order of 15 to 20,000 psi, the injection pressure will be maintained constantly in the nozzle tip and the valve operated hydraulically. In other words, a common rail system in which the injection is operated hydraulically instead of mechanically.

[1] See SAE Transactions, Vol. 52, June, 1944, pp. 248-254: "Diesel-Engine Design Trends from War Experience of the U. S. Navy," by E. C. Magdeburger.

AIRCOOLED ENGINES for Automotive Vehicles

continued from page 295

Piston Displacement, cu in.	145
Overhead Valves	
Aluminum Cylinder Heads on Each Cylinder	
Cast-Iron Individual Cylinders	
Aluminum-Alloy, 4-Ring Pistons	
Full-Pressure Oiling with Oil Filter	
3 Main Bearing Crankshaft – Counterbalanced	
Clutch	10 in. single dry plate
Gear Box	3-speed and reverse
High and Second	Helical constant mesh
Gear Box Ratios: 1.00 to 1 high	
1.75 to 1 second	
3.50 to 1 low	
4.25 to 1 reverse	
Rear-Axle Ratios: 5.5 – 6.1 – 6.5	

DESIGN CONSIDERATIONS of the CARGO AIRPLANE

by CARLOS WOOD
Chief, Preliminary Design Section, Santa Monica Plant, Douglas Aircraft Co., Inc.

THIS paper deals primarily with the considerations affecting the airplane design, leaving discussion of operations to the operators.

The big probable field for air cargo lies in the transportation of perishable and relatively high valued classes of goods.

The structural design of floors, tie-down equipment, and the like, is shown to be basically dependent only on the maximum allowable cargo load and the maximum available cargo volume.

Analysis shows that the actual operations of the cargo airplane determine the relative importance of speed of loading and weight. Speed is essential in short-range operation, but minimum weight is essential in long-range operation.

In conclusion, the author suggests that reduction of rates can make air cargo a really big business, capable of affecting the prosperity of our country and the rest of the world. Rate reduction is not only possible, but necessary, as air cargo can be a powerful instrument of trade and commerce that may be used to promote and preserve world peace.

I. Introduction

There are signs of returning sanity in discussions of transportation of cargo by air. Even the most enthusiastic advocates of air cargo no longer believe that all surface transport agencies are useless and outmoded, and the most pessimistic observers are now inclining to the belief that some cargo will probably travel by air in the post-war period.

The aviation industry is naturally very interested in the possibilities for air cargo, and the influences that air cargo will have on the operations of the air transport industry and on the design of the planes to be used in air cargo service. This paper briefly covers some considerations affecting the airplane design, leaving discussion of operations to the operators.

II. General Considerations

A. Types of Cargo – The type of cargo to be carried by air will have a definite effect on the detail design of the cargo airplane. We will consider primarily any future commercial cargo, and not present military cargo.

First let us look at the types of cargo that normally will *not* be carried by air. In general these will be types of products that have no particular need for the main advantage of air transport – high speed – or that cannot afford the main disadvantage of air transport – high cost. These types of products can be classed as nonperishables, bulk goods, and goods of low dollar value per pound. These goods do not demand high speed of transport or special care in handling and cannot absorb high transportation costs. The non-air cargo types of products include:

1. Mine products.
2. Nonperishable agricultural products.
3. Nonperishable animal products.
4. Forest products.
5. Most heavy manufactured and low-cost miscellaneous products.

Conversely, the goods that may have cause to be carried by air may be classed as perishables, goods requiring spe-

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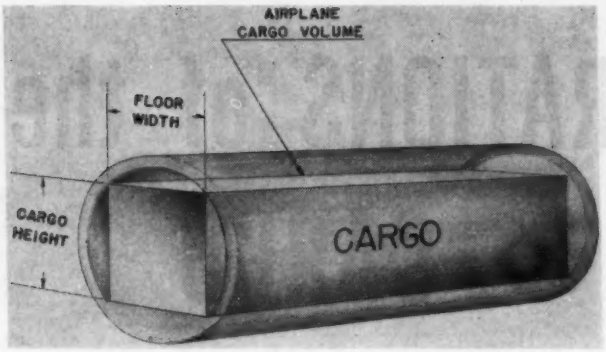


■ Fig. 1 – Structural loading from the cargo load

$$\text{Average maximum density} = \frac{\text{Maximum cargo load}}{\text{Airplane cargo volume}} \text{ (lb per cu ft)}$$

Average maximum floor loading = Average maximum density × Height (lb per sq ft)

Average maximum reaction load = Average maximum floor loading × Width (lb per ft)

cial handling, and goods of high dollar value per pound. By perishables we mean those goods that suffer physical spoilage, loss of value with time, or those that are seasonal in nature. These goods should have high speed of transport, and are normally capable of absorbing high transportation costs (at least by comparison with nonperishables, bulk, or low-cost products).

It has been said that the perfect air cargo is a valuable bank draft: it is perishable because of interest charges and it has a very high dollar value per pound. However, it is apparent that the field for air cargo is not limited to such types of goods alone, and it appears that the potential market for air cargo will probably include some portion of:

1. Perishable agricultural products.
2. Perishable animal products.
3. A few manufactured and miscellaneous products.
4. Items normally carried by l.c.l. freight.
5. Items normally carried by motor truck.
6. Express.
7. Nonlocal mail.

Immediately after the war a large part of air cargo probably will be composed of nonlocal mail together with a reasonable amount of express.

In the period following this immediate phase it may be expected that air cargo rates will be reduced as suitable cargo aircraft become available. When this occurs, then business may be developed in the transport of perishables and items normally carried by l.c.l. freight and motor truck. When air cargo is predominantly of this latter type of goods it can be said that it has come of age.

These considerations as to the type of cargo to be carried by air transport give a starting point in the determination of the other item for general consideration, that of structural loading.

B. Structural Loading – Structurally speaking, the airplane must be properly designed to withstand loads developed by the cargo in any structural design condition that must be met by the airplane.

The basic airplane structure is designed to withstand specified structural load factors; in flight due to maneuvers or gusts, and on the ground in landings or take-off. The structure supporting the cargo load should be capable of withstanding the loads developed by the weight of the cargo acting at the critical structural load factors for the airplane.

Cargo density in pounds per cubic foot is often discussed, and the range of densities from one type of goods to another generally leads to the conclusion that cargo floors and other structure should be of completely different design depending on whether the cargo is to be a load of feathers or a load of gold. Carried to the extreme there may be some basis for this conclusion, but it is felt that more general requirements have primary control over the structural design. (See Fig. 1.)

Any airplane has a maximum cargo load that it can carry, regardless of the cargo density. It also has a certain volume available for carrying that cargo. This results in an average maximum cargo density that applies to any airplane of a given design and may be expressed as:

$$\text{Average maximum density} = \frac{\text{Maximum cargo load}}{\text{Airplane cargo volume}} \text{ (lb per cu ft)}$$

This airplane may only be loaded to a given height because of its basic geometry. This will result in an:

Average maximum floor loading = Average maximum density × Height (lb per sq ft)

Also, the airplane may only be loaded across a certain width of floor because of its basic geometry. This results in an:

Average maximum reaction load = Average maximum floor loading × Width (lb per ft)

These average maximum values multiplied by the critical structural load factors for the specific airplane will then give average maximum structural loads for the design of the cargo supporting structure.

Naturally, balance or other considerations may force variations from these average loadings in practice. If this occurs, then the design conditions for the structure must be suitably modified. Modification is also necessary if certain specified pieces of equipment are to be loaded.

It is felt that the approach outlined above presents a rational basis for the structural design of cargo floors, tie-downs, retaining structure, and so forth, and that such an approach is much more reasonable than the arbitrary loads generally specified for the design of these items in cargo airplanes.

A check of present cargo airplanes indicates a value for average maximum density of about 13.5 lb per cu ft, if balance considerations are taken into account.

In any cargo airplane the following limits may be controlling:

1. If the average density of the cargo being carried is lower than the average maximum density as outlined above we run out of space before we run out of payload capacity.

2. If the average density of the cargo being carried is higher than the average maximum density we run out of payload capacity before we run out of space. As a result, it is possible to reduce the loading height and thus maintain a constant load on the cargo supporting structure.

Normally, it will be necessary for a suitable section of the cargo supporting structure to be designed for higher than average maximum loadings in order to accommodate

Table 1 – Average Cargo Density – By Stations
(Pre-War Domestic Air Cargo)

	Lb per cu ft
Los Angeles	20.2
San Francisco	19.8
Chicago	14.6
Dallas	12.3
New York	11.8
La Guardia Field	13.3
Atlanta	6.3

very heavy articles. However, it is felt that such articles generally should be handled as special loading problems, and suitable temporary load distributing members used in order that all possible structure weight may be saved. This procedure will allow the cargo airplane to retain the maximum inherent useful load at all times.

In addition, the cargo floor must be designed for impact loads from dropping of boxes and similar accidents. But in order that weight penalties from design for accidental damage may be minimized, care in handling cargo should remain the rule and not become the exception.

Before we leave the subject of structural loading it may be interesting to check on densities of typical air cargo shipments. The following data are pre-war and for a domestic airline. The average densities of cargo handled were as shown in Table 1.

It will be noted that these average cargo densities compare rather well with the average maximum density of 13.5 lb per cu ft as determined above for present cargo airplanes.

Average cargo densities of some possible post-war air cargo items are as shown in Table 2.

III. Specific Considerations

A. Compartments – It is readily agreed that any good cargo plane must be built around the cargo compartment. Undoubtedly the best and most flexible cargo compartment is one that is continuous in nature, and essentially of constant cross-section for its complete length and not interrupted by partitions, bulkheads, or structure that tend to break up the available volume into small and odd-shaped spaces.

We do not intend here to state any views as to whether the land plane or the flying boat is the cargo plane of the future, but, in passing, it should be pointed out that at present the necessary bulkheads and flotation provisions required in flying boats make a continuous cargo compartment difficult of attainment. In case flying boat type cargo airplanes are required, the requirements of the cargo compartment must be given suitable consideration for a successful design. This may entail the use of double bottoms as other suitable flotation provisions in order that a suitable cargo compartment will be available.

Table 2 – Average Cargo Density – By Products
(Possible Post-War Air Cargo Items)

	Lb per cu ft
Perishable agricultural	13-40
Perishable animal (meats)	36
Miscellaneous and manufactures	35
Store merchandise	10
L. c. l. freight	23
Express	16
Mail	17

■ Fig. 2 – Typical under-floor cargo compartment

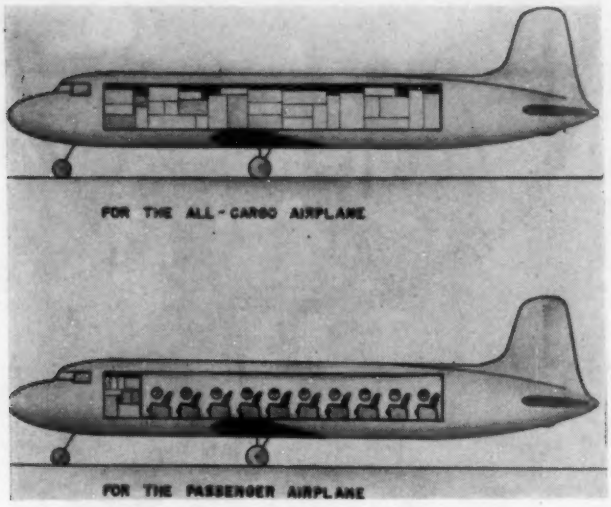
1. Under-Floor Compartments – In present-sized airplanes, under-floor cargo compartments tend to be hard to load and may be inefficient in use because of the shape of the compartment. In most cases, under-floor compartments are best used to carry passenger baggage or flexible packages such as mail sacks, as shown in Fig. 2.

Under-floor compartments may not be very satisfactory but their use is often times necessary, as it is sometimes found that the airplane is capable of handling more payload than is possible in the space available. This may result in a direct loss of revenue for the operator, and the only solution is to take advantage of every possible bit of available space for cargo even though the space may not be completely satisfactory.

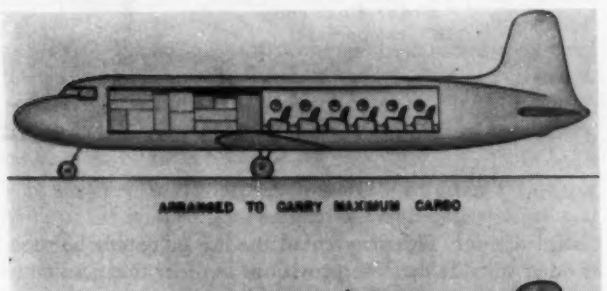
2. Above-Floor Compartments – Study indicates that the most satisfactory cargo compartment is one located above the floor. Here everything is normal, with a floor underfoot, the working space convenient and in a normal position. As a result, it is to be expected that most of the cargo in forthcoming cargo planes will be carried in the cabin.

a. Permanent Compartments (Fig. 3) – The true all-cargo plane will have a permanent cargo compartment, equipped with all of the necessary apparatus to load and retain cargo, suitably designed to provide the best facilities for the handling and transportation of air cargo.

But even in airplanes designed primarily for passenger use, permanent above-floor cargo compartments appear to be advisable and necessary, as mail, essential air-express, and passenger baggage must be carried in any case. The ease of handling and stowage of cargo above the floor



■ Fig. 3 – Typical above-floor cargo compartments – permanent compartments

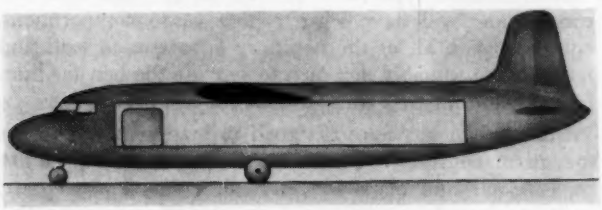


■ Fig. 4 – Typical above-floor cargo compartments – convertible cabin

appears to offer advantages substantial enough to require provisions for permanent above-floor compartments.

b. Convertible Compartments (Fig. 4) – As the use of air cargo grows there will be a growing demand for cargo space on passenger planes. For a time it may not be operationally feasible to run special all-cargo planes on some routes where the transportation of air cargo is essential.

As a logical extension of the above-floor cargo compartments in passenger planes it appeared essential to develop an arrangement allowing flexibility in the loading ratio between cargo and passengers, in order that either passengers or cargo could be carried when desired, and in order that the average payload factor could be kept as high as possible to promote profitable operation.



■ Fig. 5 – Proposed solution for access – side door

This development has been called the "convertible cabin," and in the several years that we have been working on it, it has had enough personal and press discussion to convince us that such an arrangement will be very useful in the immediate post-war future. It involved a considerable amount of careful design to assure real utility.

The convertible cabin consists of the following items:

1. A large cargo door opening into a permanent cargo compartment.

2. A movable bulkhead that may be moved to any one of several positions depending on the cargo space required.

3. Removable and foldable seats that preferably may be suitably stowed in the airplane.

4. Removable sections of carpet, and a compartment lining that may be rearranged to give the necessary protection to the interior of the airplane when cargo is carried.

5. Cargo-type floor complete with suitable tie-down fittings throughout the convertible portion.

Arrangements considered usually allow for from 80 to 90% of the payload to be carried in either cargo or passengers as desired.

It is felt that air cargo will receive a considerable stimulus from the flexibility of operation allowed by the convertible cabin.

B. Floors – It appears that cargo compartment floors should be approximately level when the airplane is being loaded as well as when it is in flight. This is most desirable for ease of loading and placement of heavy or bulky articles and for simplification of necessary activities of personnel during loading as well as in flight.

The floors and supporting structure must be designed for loads as covered earlier in this paper. The floor must be able to withstand abrasion and impact loads caused by accidents in handling. It should be made of material that may be simply repaired or quickly replaced at reasonable cost, and that will not deteriorate with age or use.

These are rigid requirements. Metal is the one material that we have found to be really satisfactory for floors. Much experimentation has been done with non-metallic flooring, but at present metal is the most satisfactory all-round material. We have found in practice that metal is as light as other materials considered suitable for comparable cargo floor use.

C. Access and Loading – There are as many solutions to the problem of access to cargo compartments as there are people thinking about the problem. Access and loading must be considered together due to their intimate correlation.

Among the solutions proposed for access are:

1. Through a large door in side of fuselage (Fig. 5).

2. Through an opening in the nose of the fuselage (Fig. 6).

3. Through the rear of the fuselage by (Fig. 7):

a. Swinging the tail out of the way.

b. Mounting the tail on twin booms.

c. Mounting the tail very high to allow a ramp to drop down for loading.

4. Through an opening in the bottom of the fuselage (similar to a bomb-bay) (Fig. 8).

We do not feel that there is any single best all-round solution for access. Method 1. may be used with either a high- or low-wing design; methods 2. or 3. with a high-wing design preferably; and, method 4. with a large low-winged design. The solution for each design should be determined by the expected operating conditions.

It is safe to say that airplanes now in use will continue in use for some time to come. They will not all be scrapped. Most of these airplanes have access through the side of the fuselage. Undoubtedly, the primary efforts on design of loading equipment for immediate use should be expended in increasing the utility of present types of airplanes.

The eventual cargo plane for domestic use will probably be a high-wing type with the floor close to the ground, but transoceanic types will probably continue to be low-winged because of superior water landing characteristics.

New and improved means of access and loading must be developed as new cargo types come into service. Each system must be designed for the specific service under consideration, and this in turn will force such decisions as the necessity for manual or power operation, self-contained hoist equipment, mono-rail systems, and other items.

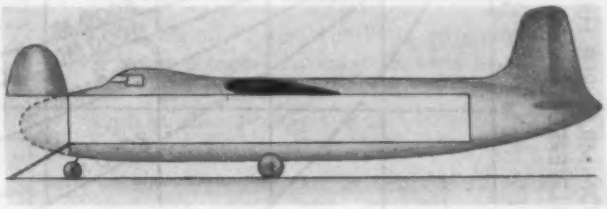
D. Cargo Tie-down and Retention – The tie-down and retention of cargo is worthy of considerable thought, as bad detail design here can cause an unprecedented amount of heartfelt profanity by the members of the loading crew, can delay schedules and cause delays en route, and if bad enough can actually endanger the safety of the aircraft because of the possibility of shifting of cargo in flight.

Tie-down and retention equipment should be so designed that there can be no shifting of cargo from time of loading to unloading. The equipment should be simple, easy and quick to use, but definitely lockable in a manner such that no slack exists in any of the cargo load. It must be designed to take all loads arising from handling of cargo and from loads arising from flight and ground accelerations and decelerations.

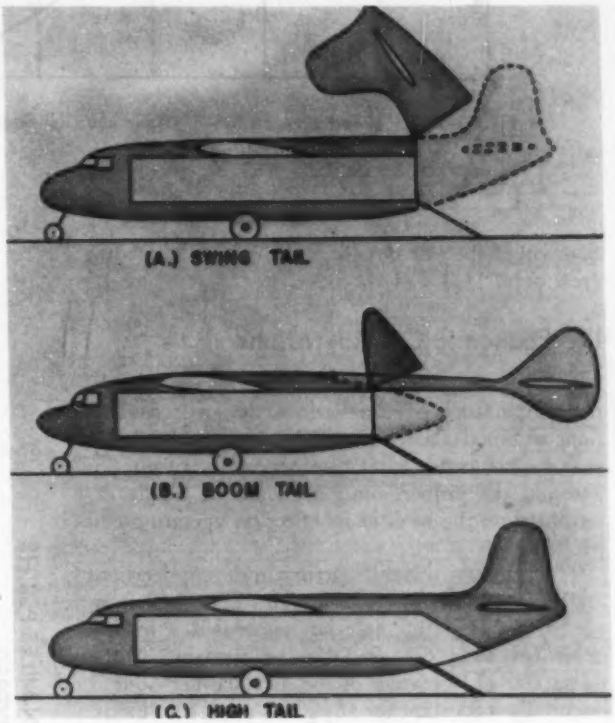
For light cargo, nets are sometimes used for tie-downs but tend to allow cargo shifting. Ropes and straps are often used, as well as other specialized types of tie-downs.

Tie-down fittings should be conveniently located to allow the tying down of any reasonable size and shape of cargo. The latest thought calls for permanently installed fittings folding flush into the floor when not in use. They should be designed to accommodate a variety of tie-down means, such as straps, ropes, cables, and tension rods. This is necessary in order to allow more flexibility in the use of various tie-down means.

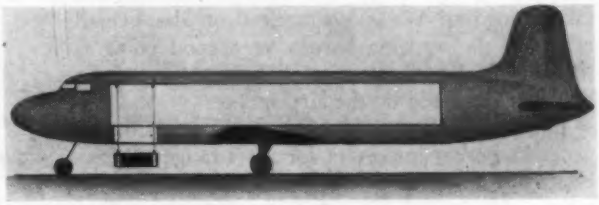
There is a growing thought that it would be very advisable to preload cargo on platforms or in containers before loading in the airplane. If properly handled, preloading may result not only in reduced loading time but in reduced packaging and equipment weights, thus promoting lower overall cargo rates. If all cargo were to be carried on platforms or in containers it would be necessary to provide tie-down fittings only at proper locations to coordinate with the platforms or containers, materially simplifying the tie-down problem. In addition, the floor design in the cargo compartment could change radically if rollers or casters were used on the platforms or containers, as it would then be necessary only to provide suitable tracks to accommodate the rollers. The remainder of the



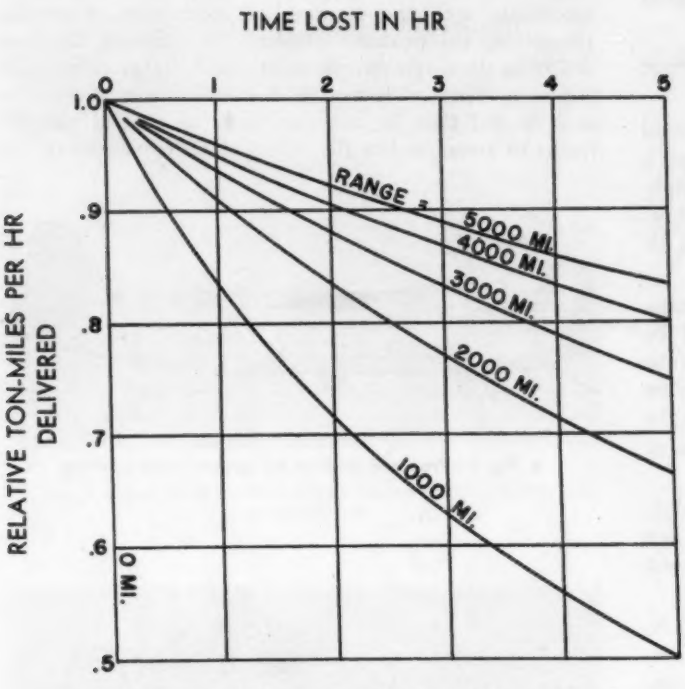
■ Fig. 6 – Proposed solution for access – nose opening



■ Fig. 7 – Proposed solution for access – rear opening



■ Fig. 8 – Proposed solution for access – belly opening



■ Fig. 9 – Effect of lost time on ton-miles per hour delivered

floor could then be designed to support the loading crew only.

IV. Economic Considerations

Before it is possible to evaluate properly the relative importance of the various design considerations outlined above, it will be well to remember that we are considering commercial air cargo. As a result, the proper solution should be finally determined on the basis of its effect on operating cost per ton-mile.

The primary influences acting in design decisions are those of lost time and weight. The best measure of the effect of these influences is that of ton-miles per hour delivered by the airplane, as the direct cost of operation of the airplane per hour is essentially a constant for any given set of conditions.

The following data based on a hypothetical cargo airplane design show the relative importance of time and weight for several different operating conditions:

A. Ton-Miles per Hour Plotted against Lost Time (Fig. 9) – Exponents of the simplest and fastest loading system regardless of the overall effect on the airplane have a very good point as time lost in loading reduces the effective ton-miles per hour delivered by the airplane.

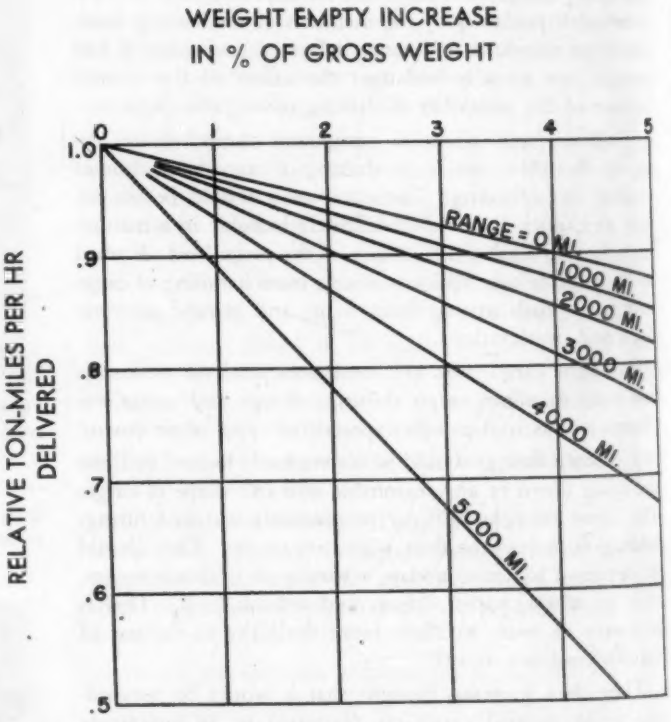
The time factor (in loading or in lost time in general) is very important for short ranges. However, this importance decreases with increase in range. As a result, it can be said that speed of loading, although very important for short-range operation, is comparatively unimportant for long-range operation.

B. Ton-Miles per Hour Plotted against Weight (Fig. 10) – The weight penalties that must be paid for easier loading, access, or more satisfactory cargo provisions must come out of the payload.

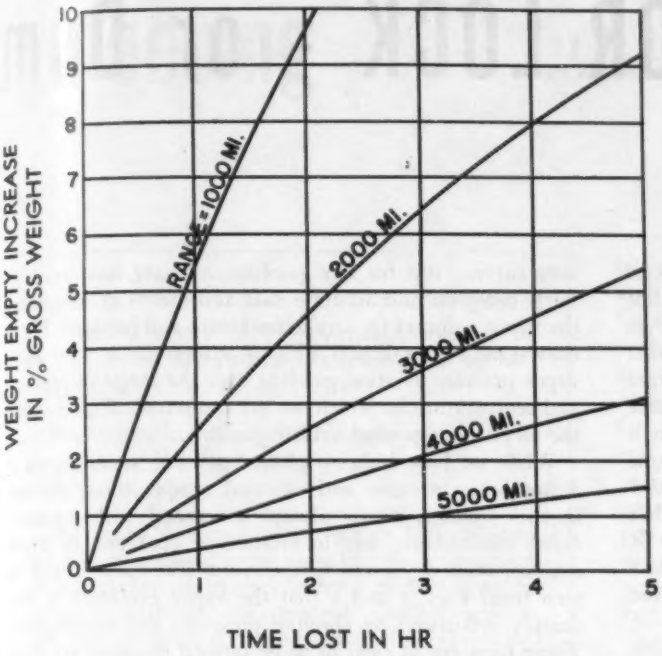
As a result, although at short ranges the payload is large and added weight has a comparatively small effect on the ton-miles per hour delivered, at long ranges the factor of weight becomes extremely important because of the correspondingly smaller payload. Thus it can be seen that large weight penalties for loading equipment may be warranted for short-range operation, but are not allowable for long-range operation.

C. Weight Plotted against Lost Time for Equal Ton-Miles per Hour (Fig. 11) – From the above is seen the necessity of obtaining a balance between the weight involved in loading equipment and access provisions and the time saved by these items.

From the figure it is seen that time is relatively very important at short ranges, calling for almost any reasonable sacrifice of weight in order to save time. But for long-range operation the picture is reversed, as it appears to be much more important to save weight to put into payload than to save time in loading. Each operation should be judged for its own conditions in determining the relative cost of weight and time penalties for different means of loading and access.



■ Fig. 10 – Effect of weight empty increase on ton-miles per hour delivered



■ Fig. 11 – Weight empty increase plotted against time lost for equal ton-miles per hour delivered

D. Cargo Volume Plotted against Rates (Fig. 12) – The question arises as to whether transportation costs are really important. It can be said that evidence indicates a tremendous increase in potential cargo available as air cargo rates are reduced.

It is not implied in the figure shown here that air transportation will carry all of the cargo at any rate indicated on the figure. All that is intended is to show the total amount of cargo that moves at or above the indicated rates, and to point out that air transportation can compete for the transport of any and all cargo that will move at or above the rates that must be charged for air cargo. Thus, lower air cargo rates allow us to compete in a much larger market. The amount of cargo that becomes air cargo depends on the advantages that our means of transport possesses in speed and convenience.

V. Conclusions

1. It has been shown that the big probable field for air cargo lies in the transportation of perishable and relatively high valued classes of goods.

2. The structural design of floors, tie-downs, and so on, has been shown to be basically dependent only on the cargo load that the airplane may carry and on the cargo volume available.

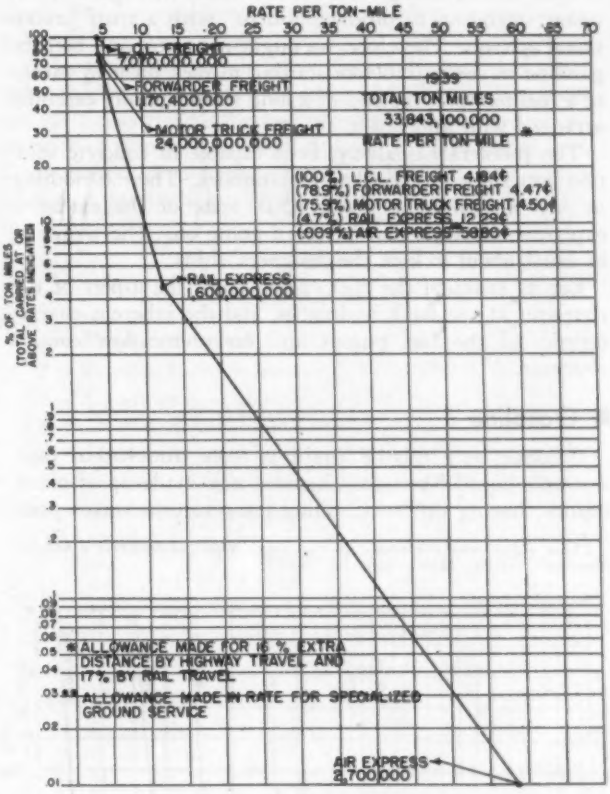
3. The relative disadvantages and advantages of underfloor and above-floor cargo compartments have been discussed, as well as some advantages of the convertible cabin and its application in the immediate post-war period.

4. Access, loading, tie-downs, and so on, are only discussed in general, as detailed decisions depend on the operational conditions.

5. Economic considerations show that the actual operations of the cargo airplane will determine the relative importance of speed of loading and the weights involved. Short-range operations demand speed of loading, but long-range operations demand minimum weight penalties.

6. Economic statistics indicate an enormous increase in potential air cargo as rates are reduced. The amount of air cargo actually developed depends on the advantages of air transport over other transport in speed and convenience.

If air cargo rates can be reduced to reasonable values, this business not only can be big business, but can profoundly affect the prosperity of our country and the rest of the world. We believe that this rate reduction not only is possible but necessary as air cargo can be a powerful instrument of trade and commerce that may be used to promote and preserve world peace.


RATE PER TON-MILE
L.C.L. FREIGHT 7,070,000,000
FORWARDER FREIGHT 170,400,000
MOTOR TRUCK FREIGHT 24,000,000,000
1939
TOTAL TON MILES 33,843,100,000
RATE PER TON-MILE *
(100%) L.C.L. FREIGHT 4.184¢
(78.9%) FORWARDER FREIGHT 4.47¢
(75.9%) MOTOR TRUCK FREIGHT 4.50¢
(4.7%) RAIL EXPRESS 12.29¢
(.005%) AIR EXPRESS 58.80¢
RAIL EXPRESS 1,600,000,000
% OF TON MILES (TOTAL CARRIED AT OR ABOVE RATES INDICATED)
* ALLOWANCE MADE FOR 15 % EXTRA DISTANCE BY HIGHWAY TRAVEL AND 17% BY RAIL TRAVEL
** ALLOWANCE MADE IN RATE FOR SPECIALIZED GROUND SERVICE
AIR EXPRESS 2,700,000

■ Fig. 12 – Cargo volume plotted against rates

From this figure, it can be seen that a reduction of air cargo rates will allow air transportation to compete in a much larger market on an equal basis as far as rates are concerned. The amount of this cargo that will become air cargo will depend on the advantages that air transport possesses over the other means of transportation, namely, speed and comfort

VAPOR LOCK - or Dumb Eng

VAPOR lock was with us for the first time some 12 years ago in a few cars. It was cured by taking the gas line off the hot parts of the engine, by putting a larger vent on the carburetor, or by putting in a fuel pump with "larger vapor-handling capacity." This last simply means a larger pump. In the early part of the present war, the trouble showed up again, only this time it was more serious, as it occurred in combat vehicles, and in planes as flying altitude went up. So everybody bustled around to figure out how we were to fix that damnable fluid gasoline so it wouldn't give off vapor. But inasmuch as we need a controlled 10% point for starting, a 50% point for good warm-up and distribution, and a 90% point sufficiently removed from furnace oil to keep from washing the lubricating oil off the cylinders, such an effort was bound to be sad. Aviation gasoline is excitable material, and almost any experienced drinker knows enough not to suck Vichy through a straw, unless intending to produce "burps" with a truly professional aplomb. Therefore, we ought not to try and boil the gasoline in the tank before starting to suck on it by means of a fuel pump at the end of a long straw, usually carefully arranged with knots in it.

The problem has always been capable of concrete solution ever since we have had automobiles. There is nothing in any way abstruse about it, in spite of the extensive experimental work that has been going on. The writer is, as usual, about to kick the designers' shins.

Let us examine the fuel system from the aspect of the character of the fluid, hydraulics, and the inherent characteristics of the fuel pumps and carburetor flow control elements.

Gasoline

Gasoline, as a volatile liquid, is more troublesome than a simple liquid like water, because it is made up of many liquids, having different boiling temperatures versus pressure curves. But for any gasoline, we have now available fairly complete and accurate data and means of computing the vapor volumes for any temperature and pressure. Fig. 1 shows the vapor/liquid (V/L) values for a 7-psi Reid vapor pressure aviation gasoline over the range of pressure and temperature in which we are interested. Fig. 2 shows the all-purpose ground vehicle gasoline.

While we have hitherto plotted gasoline vapor curves at atmospheric pressure and elevated temperatures, we are, in fuel systems, nearly always concerned with pressures *below* atmospheric, and in aviation or operation in Arctic regions, with depressed temperatures likewise. It will be seen from Figs. 1 and 2 that the vapor evolution is very sharply influenced by absolute pressure, and temperature. These facts are so clear as to be termed obvious: yet there has been relatively little intelligent attention given to decent hydraulic design of the fuel system to suit the characteristics of the gasoline. The gasoline V/L curves provide the basic conditions around which the fuel system must be designed in order to function properly. Three principles can be laid down:

1. The heat flow into the fuel system must be controlled, or eliminated, on the inlet side of the fuel pumps.

2. The resistance of the piping and pumps must be low enough so that the total pressure drop from tank to pump will not produce more vapor and liquid total volume than the delivery capacity of the fuel pump.

3. Flow-metering devices, such as carburetor orifices, can only meter a homogeneous fluid, either a liquid or a gas. They cannot measure a mixture of the two correctly, but may approximate correct measurement for a limited range of mixture.

One method of preparing gasoline V/L curves is fully given in the References at the end of the paper, items 1 to 6, and need not be further considered herein.

Air is soluble in gasoline to the extent of about 12% by volume, ntp. The closely approximate methods of calculating solubility have been given by Bridgeman and Aldrich.[1] Air comes out of solution by *weight* in proportion to the difference in pressure, according to

$$a = a_o \left(\frac{p_o - p_1}{p_o} \right) \qquad (1)$$

Where

a = Weight of air per unit volume of liquid

a_o = Weight in solution at p_o

p_1 = Depressed pressure at which release occurs

Solubility of air is not appreciably affected by temperature. Supersaturation has been put up by some as a bogie causing sudden release of air. Those advocating supersaturation ought to remember that water-deaerators work

[This paper was presented at the SAE War Engineering - Annual Meeting, Detroit, Mich., Jan. 14, 1944.]

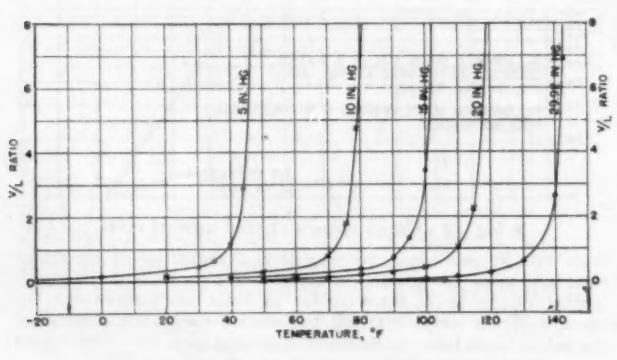


Fig. 1 – V/L values for 7-psi Reid vapor pressure aviation gasoline (sp gr 60/60 = 0.706)

[1] See SAE Transactions, Vol. 25, 1930, pp. 144-150, 230: "Effect of Weathering on the Vapor-Locking Tendency of Gasolines," by O. C. Bridgeman and E. W. Aldrich.

Engineering

by R. J. S. PIGOTT
Gulf Research & Development Co.

A QUICK survey of the hydraulic aspects of the vapor lock problem is given here by Mr. Pigott.

The hydraulic design of the supply system from the tank to the fuel pump is discussed by the author.

The cause for carburetor vapor lock beyond the fuel pump becomes very largely a question of the ability of control orifices of any type to measure mixtures. In general, control orifices can only measure a homogeneous fluid correctly.

The general methods of calculating a system for obtaining the limiting tank temperature for take-off and the limiting altitude for any system, are discussed, and the requirements summarized.

■ ■ ■

THE AUTHOR: R. J. S. PIGOTT (M '18), chief engineer of Gulf Research & Development Co., has been engaged in work on heat distribution in internal combustion engines and the behavior of lubricating and fuel pump systems. Mr. Pigott, who has about 30 patents covering equipment for power stations, oil burners, meters, safety valves, and so forth, is active in Coordinating Research Council work.

perfectly by gently boiling at any corresponding temperature and saturation pressure; all the air comes out with the vapor. It seems inconceivable that with the gasoline boiling to give off vapor, any air can remain in supersaturation; it is bound to come out.

Since the V/L curves as calculated (Figs. 1 and 2) are for gasoline saturated with air, the values include air evolved as well as vapor. In any case, the fuel pump cannot distinguish between air and vapor, only between compressible and incompressible fluids.

Piping

Much work has been done on the flow resistance of piping, pipe fittings, and bends during the last 50 years. The friction of fluids in all kinds of pipe was examined by Kemler,[2] and summarized, with a reduction to a single family of curves by the writer.[3] This paper of the writer has been widely republished and used in standard texts, and its general accuracy is now quite well confirmed. The family of friction factor curves is shown in Fig. 3.

In 1939 and later, a research specifically on gasoline and aviation tubing and fittings was carried out at the Bureau of Standards, under the auspices of the Vapor-Lock Group, Coordinating Research Council, Inc., and under the general direction of Dr. O. C. Bridgeman.[4] This work has been completed, but the full correlation has not yet been reported. However, examination of the 28 progress reports indicates that the friction loss for liquid aviation gasoline coincides with line 1 of Fig. 3. The Bureau experiments covered ⅜ to 1 in. O.D. tubing and fittings, and a range of Reynolds numbers from 2000 to 60,000. This is the full range likely to be found in any fuel system. Since line 1 in Fig. 3 applies to all smooth-drawn copper and aluminum tubing, we are only concerned with this single line, at values of Reynolds number not exceeding 100,000.

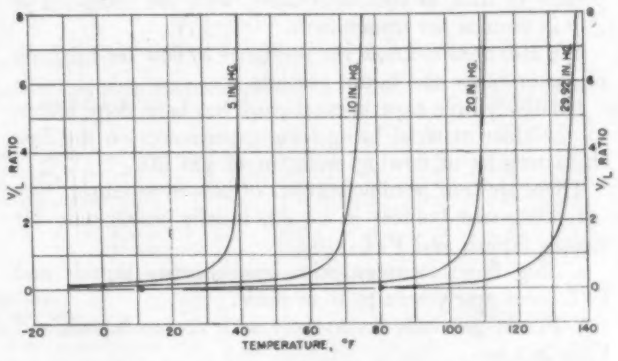
A formulation can, therefore, be established that eliminates the friction factor f as such, and likewise the necessity for a chart. This method is given in the writer's paper mentioned above, and is also used in the Vapor-Lock Group's work discussed above. They have adopted a value of −0.25 for the slope of the friction factor line 1. The usual formula is

$$\Delta p = \frac{f l \rho v^2}{144 \times 2gd} \qquad (2)$$

Where

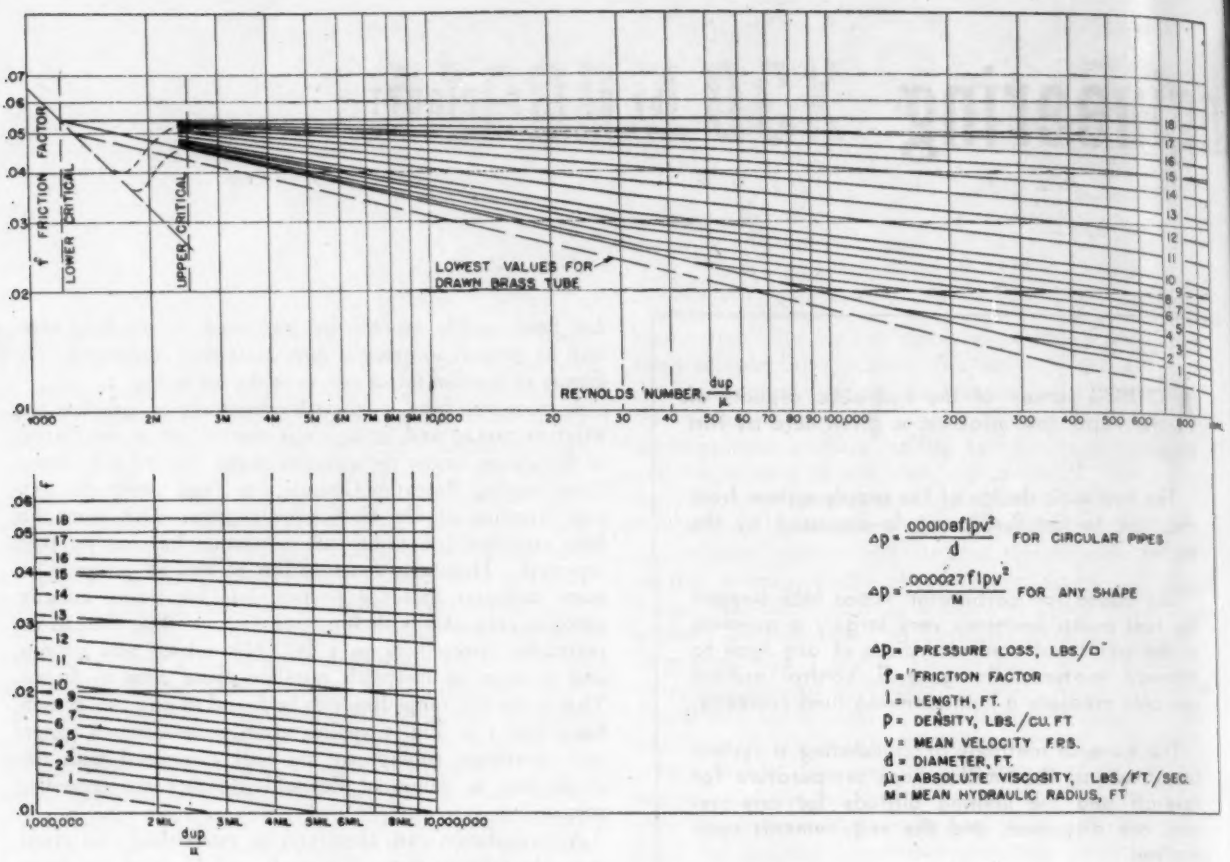
Δp = Pressure drop, psi
f = Friction factor, a number
l = Length of pipe, ft
ρ = Density of fluid, lb per cu ft
v = Mean velocity of fluid, fps
μ = Viscosity, English units (centipoises × 0.000672)
d = Inside diameter of pipe, ft

All consistent units.



■ Fig. 2 – V/L values for 8-psi Reid vapor pressure motor gasoline (sp gr 60/60 = 0.730)

[2] See ASME Transactions, Vol. 55 (Hydraulics), Aug. 31, 1933, pp. 7-32: "A Study of the Data on the Flow of Fluids in Pipes," by E. Kemler.
[3] See *Mechanical Engineering*, Vol. 55, August, 1933, pp. 497-501, 515: "The Flow of Fluids in Closed Conduits," by R. J. S. Pigott.


f FRICTION FACTOR
LOWER CRITICAL
UPPER CRITICAL
LOWEST VALUES FOR DRAWN BRASS TUBE
REYNOLDS NUMBER, dvρ/μ
Δp = .000108flρv²/d FOR CIRCULAR PIPES
Δp = .000027flρv²/M FOR ANY SHAPE
Δp = PRESSURE LOSS, LBS/□"
f = FRICTION FACTOR
l = LENGTH, FT.
ρ = DENSITY, LBS./CU. FT
v = MEAN VELOCITY F.P.S.
d = DIAMETER, FT.
μ = ABSOLUTE VISCOSITY, LBS/FT/SEC.
M = MEAN HYDRAULIC RADIUS, FT

■ Fig. 3 – Friction factor plotted against Reynolds number

But for the single smooth pipe, we can write

$$f = \frac{0.3164}{(Rn)^{0.25}} = 0.3164 \left(\frac{\mu}{dv\rho}\right)^{0.25}$$

This is the form used in the Coordinating Research Council, Inc., reports. Substituting

$$\Delta p = (0.000108\ \rho v^2)\ 0.3164 \left(\frac{\mu}{dv\rho}\right)^{0.25} \frac{l}{d}$$

$$\Delta p = \frac{3.417 \times 10^{-5}\ \mu^{0.25}\ l\rho^{0.75}\ v^{1.75}}{d^{1.25}} \qquad (3)$$

This formula can, of course, be readily rearranged in terms of any convenient quantities such as gallons per minute or per hour, or pounds per hour, diameter in inches, viscosity in centipoises, as required. The writer prefers to stick to consistent units, with the exception of Δp in pounds per square inch.

We also need to know the resistance to flow for mixtures of gas or vapor and liquid gasoline.

Relatively little experimental work has been done hitherto, the chief material being from experiments on the flow characteristics of flowing wells, or of gas lifts.

There are four possible regimes of flow in mixtures:

1. Dispersed bubbles in liquid, usually stable only for viscous liquids and $V/L < 0.5$.

2. Slug flow: common with low viscosity liquids and V/L from nearly zero to 10 or more.

3. Foam: generally occurs only with viscous liquids and $V/L > 1.0$.

4. Dispersed droplets: with any liquid at high velocity and $V/L > 50$.

With gasoline, (1) is not stable and can be disregarded; (3) is unstable and extremely difficult to maintain; (4) is far beyond the V/L range possible in any fuel system. We are, therefore, left with (2) as the probable regime.

Considering the pressure drop in terms of liquid flow only, regarding the vapor or gas resistance as negligible, we have:

$$v_m = v_l\ (1 + V/L)$$

Where

v_m = Average mixture velocity, fps

v_l = Average liquid velocity, no vapor present

Since we are considering only slug flow, and since the resistance is almost entirely skin friction, we can regard the gas or vapor as a reduction of skin area, or as a reduction in effective density ρ. We then have:

$$\rho_m = \frac{\rho_l}{1 + V/L}$$

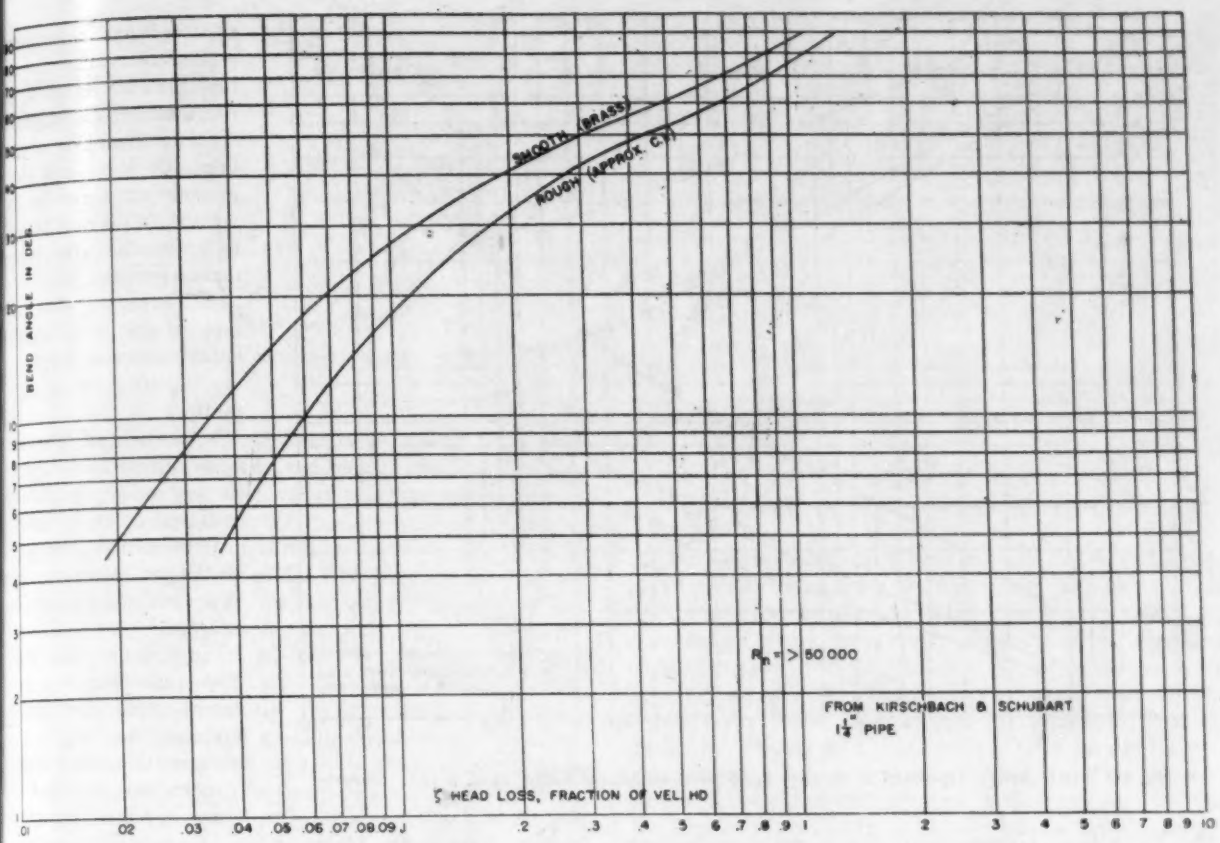
Substituting in equation (3):

$$\Delta p = \frac{3.417 \times 10^{-5}\ \mu^{0.25}\ l\rho^{0.75}\ v^{1.75}}{d^{1.25}} \cdot \frac{(1 + V/L)^{1.75}}{(1 + V/L)^{0.75}}$$

$$= \frac{3.417 \times 10^{-5}\ \mu^{0.25}\ l\rho^{0.75}\ v^{1.75}}{d^{1.25}} (1 + V/L) \qquad (4)$$

In short, we should expect that the pressure drop for mixtures of gasoline and air or vapor should be simply the ordinary liquid pressure loss times the ratio total volume/liquid volume, or $(1 + V/L)$. The work at the Bureau of Standards fully substantiates this formulation as correct, so it may be safely accepted.

In the case of fittings, the situation is not fully buttoned



■ Fig. 4 – Loss in miter bends, not including pipe friction of bend

up yet, but sufficiently close for all practical purposes. Experimental results reported by Gibson, Davis, Schroder, Brightmore, Hofmann, and Freeman, indicate that there is some tendency for the losses in elbows, bends, and other fittings to vary as an inverse function of Reynolds number, at values below 150,000, but the results are by no means consistent. On the other hand, most of the work except that of Hofmann was done on cast fittings, therefore rough. The Vapor-Lock Group's work has been done on airplane fittings which are either drilled or smooth cast. These results, although likewise none too consistent in some respects, do show a fairly clear variation with Reynolds number in the smaller sizes.

Fig. 4 shows a curve for sharp bends corresponding to those aviation ells that are drilled, not cast. These data were the result of very careful work by Kirschbach and Schubart. The values obtained in the Vapor-Lock investigation do not differ too much from these values.

Fig. 5 shows the variation of ζ, the bend loss; this chart was prepared from data given in a number of sources, and indicates a variation of loss with Reynolds number below 170,000; no effect is shown above that value, and Freeman's extensive work corroborates this fact.

Since designers need specific data on losses in various fittings, and the writer has used values such as Figs. 4 and 5 for years with quite satisfactory success, it might be of interest to present these data for immediate use, pending the issue of the Vapor-Lock Group's final report. The data in Table 1 have been extended in some of the special aviation fittings from the progress reports of the Vapor-Lock Group, but the data on unions, tees, ells, and bends are the same as the writer has been using, whether these agree with the latest dope or not.

The variations found in the work of the earlier investigation, as well as 2:1 variations in data in the latest investigation, show that we cannot be too fussy about frog-hair accuracy, as the data certainly do not justify more than two-place figures. But it is comforting to realize that the biggest variations occur in very small resistances and therefore are not serious anyway.

Most of these individual small values must be obtained as differences between much larger quantities, and therefore the variations will be exaggerated.

Consideration of the earlier work, together with that of

Table 1 – Friction Loss in Fittings

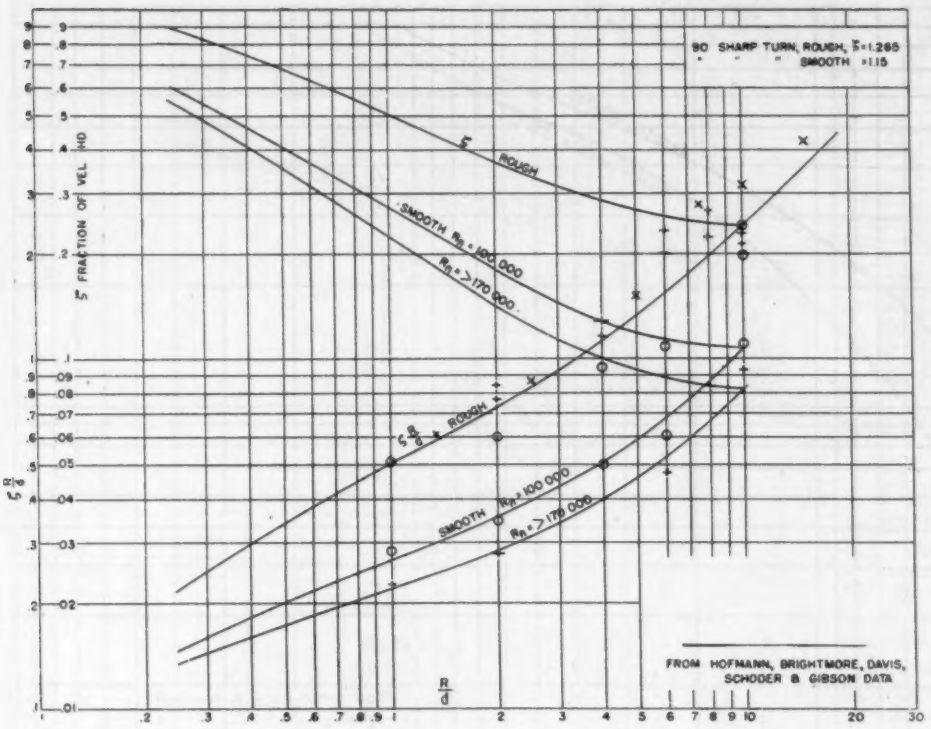
(In terms of loss per foot of pipe)

All values are bend losses only; length of fittings must be added to tubing length for total loss.

Connectors and unions, hose fittings unlined					0.20 all sizes
Tee, straight through flow					0.35 all sizes
	⅜	½	⅝	¾	1
Tee, angle flow and 90-deg sharp ell	1.50	2.15	2.73	3.42	4.95
90-deg ell, R/d = 0.5	1.00	1.44	1.82	2.28	3.30
45-deg ell	0.50	0.71	0.91	1.14	1.65
sharp bends R/d = 1	0.22	0.30	0.37	0.45	0.62
R/d = 2	0.15	0.21	0.27	0.32	0.44
R/d = 4	0.11	0.15	0.19	0.23	0.32
R/d = 6	0.09	0.12	0.15	0.18	0.25

Accessories, deduced from Vapor-Lock Group data

Strainers	4.8 all sizes
D hand pump	12.4 + 0.095 psi static loss
Aero poppet selector valve	8.4
Parker disc-type selector valve	6.3
AEL-1 unit	36.0 + 0.675 psi static loss
G2A selector	5.9
E2 selector	7.4


90 SHARP TURN, ROUGH, ζ=1.265
" " SMOOTH =1.15
ζ ROUGH
SMOOTH Re = 100 000
Re = > 170 000
ζ R/d ROUGH
SMOOTH Re = 100 000
Re = > 170 000
FROM HOFMANN, BRIGHTMORE, DAVIS, SCHODER & GIBSON DATA
ζ FRACTION OF VEL HD
R/d

■ Fig. 5 – Loss in bends, expressed in velocity head plotted against radius bend / pipe diameter

the Vapor-Lock Group, shows that a sharp 90-deg turn loss is cut down one-third by a mere R/d value of 0.5, down one-half at $R/d = 1$, and nothing appreciable is gained beyond $R/d = 6$.

The values on strainers, transfer valves, and similar items are large enough for reliable measurement, and the work of the Vapor-Lock Group will be valuable, because it is very nearly the only material on such structures.

The differences between tees and ells, one-end tubing connection and one-end pipe size, and the tubing to tubing type, seem to be due to the fact that when the liquid is flowing into the pipe end (larger than the tubing I.D.), there is a vena contracta formed by the edge at the point of reduction in section.

The classic hydraulics usually teaches that the loss at a sharp reduction is $0.000108\ \rho\ (v_2^2 - v_1^2)$. If the coefficient of a sharp edged orifice for the ratio of diameters is used, (varies from 0.60 at $d_2/d_1 = > 0.40$ up to 0.63 at $d_2/d_1 = 0.80$) this formula gives the velocity head to bring the liquid up to the speed at d_2, and accounts for the vena contracta. Not all of this head is lost. At emergence into a large diameter, the loss is represented very closely by $\left(\frac{v_1 - v_2}{v_1}\right) p_1$, where v_1 and v_2 are the higher and lower average velocities, and p_1 is the velocity head for v_1. These considerations apply generally to sharp-edged restrictions, such as bulkhead fittings or some of the dump valves.

■ Pump Losses

It seems that in spite of the immense amount of design effort expended on pumps, practically no critical attention has been given to the inherent pressure losses on the suction side of any kind of pump.

It ought to be quite clear that a vacuum gage at the suction flange of the pump is no criterion of the actual pressure subsisting in the interior of the pump.

Some discussion of suction side losses was given in another paper by the writer[4], but it may be well to outline the principal factors comprising this loss.

1. Turns and skin friction in the inlet passages between suction flange and moving mechanism of the pump.

2. Centrifugal force due to liquid in the impellers for any rotary pump taking in liquid at the periphery.

3. Inlet-valve loss in a diaphragm pump, port loss in a vane pump, tooth loss in a gear pump.

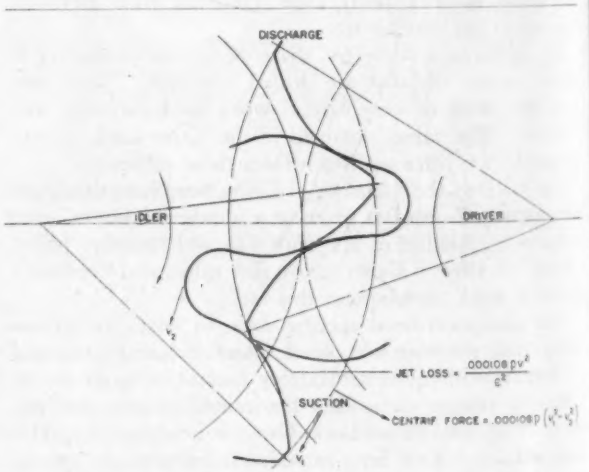
Fig. 6 shows where these losses take place in a gear pump: small loss item 1, moderate loss item 2, generally considerable loss item 3, tooth loss, provided the pump is not in cavitation.

Fig. 7 shows the points of loss in a vane-type pump as often constructed: item 1 again low, 2 moderate, 3 often quite large.

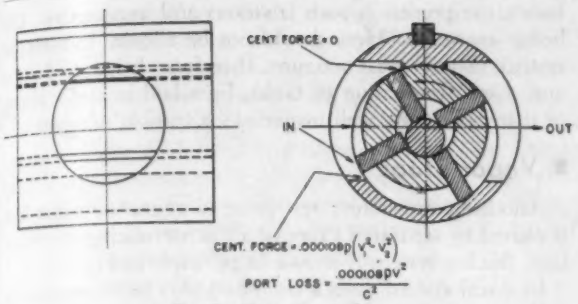
Fig. 8 shows a reciprocating type-diaphragm operated pump. In the suction cycle, flow quantity follows a sine wave closely. The important loss, aside from the pulsating flow, occurs usually in the suction valves, and is, in earlier designs, quite high.

Fig. 9 shows the cavitation test on a vane-type pump, performed as described in the above reference, and Fig. 10 shows the test equipment. The loss at normal full speed and flow is 6 to 7 in. of mercury on aviation gasoline. The reason is obvious. The intake chamber losses, item 1, are

[4] See SAE Transactions, Vol. 52, March, 1944, pp. 73-86: "Oil Aeration," by R. J. S. Pigott.



■ Fig. 6 – Spur-gear tooth loss



■ Fig. 7 – Vane pump losses

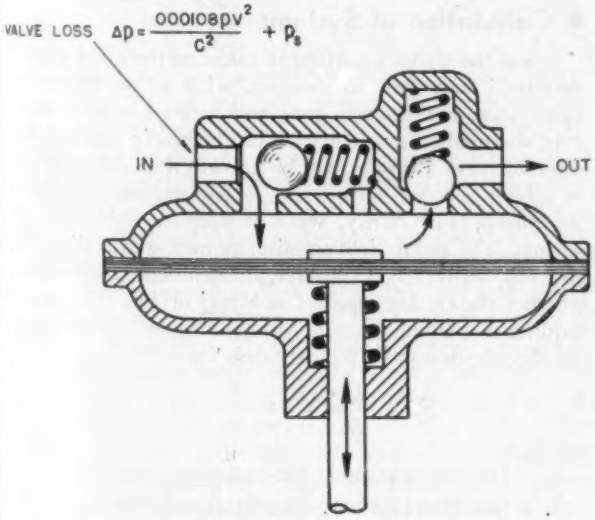
insignificant, the centrifugal force, item 2, bucking inflow is 0.71 in. at maximum, and the bulk of the loss is in the port sleeve. It is easy to perceive that the vane space requires a considerable flow of liquid to satisfy the volume change up to the point of cutoff at the lower abutment of the port sleeve, but the area for flow between the edge of the port, which is *round,* and the advancing edge of the next advancing blade, has been rapidly decreasing toward zero for a considerable arc, and the consequent velocity through the port theoretically should reach high values. Actually it does not, because the increased pressure drop causes the temperamental gasoline to "pooh out" a lot of vapor, and some cavitation occurs. A single simple change to a square port dropped the cavitation point 2.0 in. of mercury, and other changes no doubt can be made to reduce the loss still further.

Kick in the shins for the designers: they should have caught this fault. Orchids for the rest of the design – it is otherwise a nice pump.

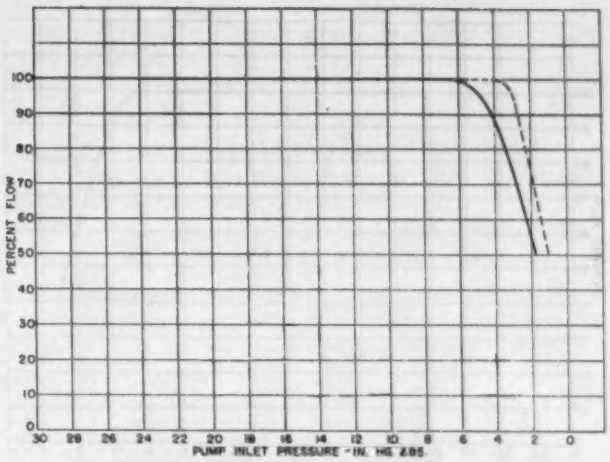
Fig. 11 shows the test of another make of vane pump, just to show that the neglect of intake design is not limited to one designer.

The writer has not had time to get data on the diaphragm pumps, but, from experience, the single-valve designs generally are bad bubble-jerkers. The recent development of a six-valve design showed a very marked improvement. Orchids: one designer woke up.

Summing up: for high-altitude work (airplanes) or hot

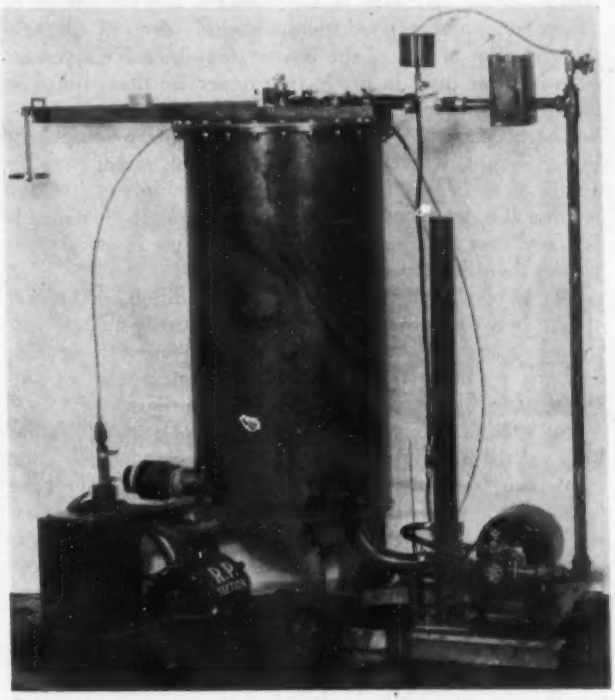


■ Fig. 8 – Diaphragm pump losses

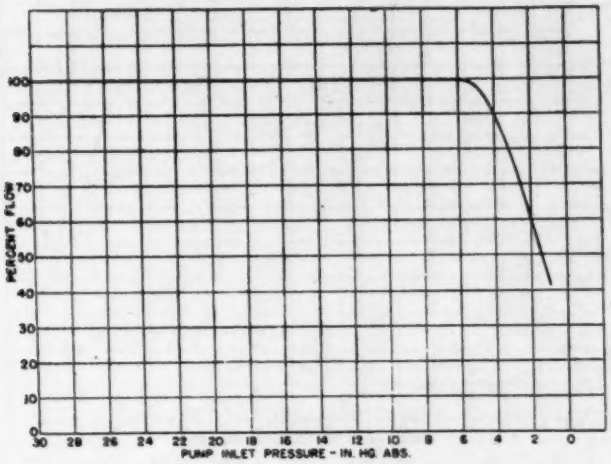


■ Fig. 9 – Vane-type fuel pump A – cavitation test – 2500 rpm

gasoline (tanks), we must come back to God and cut out all the idiotic resistances we now build into the suction side of the fuel system – no sharp elbows, large lines, *no strainers or filters,* reduce the pump losses by improved design, no needle or globe valves, all valves designed for low resistance. And when we have done all these things, it will still not be good enough for 40,000- to 50,000-ft altitudes. So we go a step further. The lowest suction-side pressure drop can be obtained when there is no suction pipe whatever, and the only loss is the pump loss. This theorem of course leads to the conclusion that the pump, or at least the first pump, should be at the tank – a booster. The pump must be selected for low inlet losses, and this generally points right at the centrifugal, or to a redesign of other pumps to get as low a loss as a really good centrifugal. In a centrifugal pump, it is apparent that centrifugal force has been eliminated by the entry at an "eye" concentric with the shaft, but it is also very impor-



■ Fig. 10 – Test set-up



■ Fig. 11 – Vane-type fuel pump B – cavitation test – 2500 rpm

tant that the pickup edge of the blades be designed to avoid shock loss, which can be quite heavy.

For example, the straight radial blade impeller is bad in this respect, as the designers of this type of booster are now aware, since the blade is now modified. Even with pickup blades aimed at best angle, it is sometimes difficult to get a small enough angle and still manufacture the thing. In this case, composite impellers, or even an auxiliary intake propeller, can readily be employed. We have lowered cavitation point by as much as 6 in. of mercury on high-speed centrifugals with an auxiliary pickup wheel.

Theoretically, a centrifugal booster should attain 0.5 to 0.75 in. of mercury cavitation point, but most of them so far are not better than 4 in. Naturally, the use of boosters on each of several tanks in a plane or tank gives rise to some awkward valving and control problems, but we had better face the situation.

So far as ground vehicles are concerned, it should ordinarily not be necessary to use boosters; a good design of the piping, and a good pump should serve in all new designs. But of course the booster may be a life-saver for fixes in those cases in production or service that cannot be conveniently rebuilt in the main fuel system.

The booster should be treated as a primary component in the design, where it is necessarily used, rather than an auxiliary to be cut in when needed for higher altitudes, because the design of bypasses of low enough resistance is far from easy and may cost more in pressure loss and complexity than the expedient may be worth.

For airplanes, the problem of heat addition to the fuel system between tank and fuel pump is usually absent. But the plane may be heated by standing in an African sun, and the resulting temperature of gasoline must be considered as a major factor in take-off conditions.

For ground vehicles, the larger part of vapor lock has been caused by improperly protected lines and tanks aside from poor hydraulics of the system and pump. We have had gasoline tanks reaching 160 F by recirculation of hot engine compartment air around them, and a few insane examples where a gasoline line has been laid near the exhaust lines, or even bracketed on them with no insulation.

It appears that most of these cases have been cured – the hard way by building them first, and thinking about them afterward. Although heat transmission into the fuel system may be reasonably closely calculated, given the ambient temperatures and approximate flow speeds (which we now have), the process is both laborious and unnecessary. The better answer is: Heat should not be allowed to enter the system in any great amount, therefore design to keep it out. Good ventilation of tanks, lines laid in cooler places, or if in hot zones, well insulated – a feasible solution.

■ Vapor Traps

Another contributory resistance to aggravate vapor lock is caused by separated vapor or air accumulating in a rising line. Such a system is shown in principle in Fig. 12.

In a real system, such a condition may be represented by a filter or strainer. The small-size openings of the filter or screen can produce a pronounced Jamin effect. Bubbles of released air or vapor accumulate on the surface until a critical pressure difference is built up that flushes out the bubbles, resistance drops, and the cycle begins over.

Strainers or filters should not be on the suction side of the fuel pump, but if a booster pump is used, can be placed between booster and main pump. It is wise to avoid all capacities followed by a restriction in any suction line.

■ Carburetor

Vapor lock, of course, can occur at the carburetor, and will certainly center at control orifices. An orifice can measure either liquids or gases correctly, but it cannot measure both together, particularly as the discharge for a liquid is widely different than that for a gas. The answer is that if the system is designed for low suction loss, and low heat addition, there will be little or no bubbles of evolved air, or practically no vapor on the discharge side of the pump, so that the float valve or carburetor orifice can function normally on a liquid; nor would it be necessary to raise pump pressure to help condense the vapor. But if the suction side of the system is sour, as pointed out, a booster pump assists; and a separator of some kind between pump and float valve to get rid of vapor before metering would be of considerable value.

For those cases where air is admixed as part of the metering operation, obviously the addition of free vapor not metered by the air control will upset the mixture, by changing resistance to flow and consequently altering the amount of liquid. Therefore, heat addition should be avoided until after the metering operation is completed.

■ Calculation of Systems

It was the writer's purpose to calculate the losses in some definite installations to compare with actual failures by vapor lock, but space is gone and time's awastin'. However, the method of calculation can be briefly outlined. All dimensions of a system will be known, size and lineal feet of tubing, elbows, valves, and so on, and the capacity of the pump. For clarity, we will keep to cubic feet per second. The pump delivers displacement minus slip (cfs), and this delivery is a constant, independent of altitude or whether the cfs are supplied as liquid or gas. The engine requires a definite amount of gasoline in cfs to supply full-throttle demand. We may then state:

$$\frac{Q_p}{Q_e} = 1 + \frac{V}{L} \qquad (5)$$

Where

Q_p = Delivery of pump at full-load speed, cfs

Q_e = Gasoline required to supply full load, cfs

V/L = Vapor/liquid ratio

This means that the pump (always larger in capacity than engine demand) can satisfy the engine up to a definite V/L value. Since this mixture of liquid gasoline, vapor, and air is a chosen constant quantity, the losses for the system can be figured once for all, and will be independent of altitude. Total the equivalent bend resistance in feet of pipe, for all fittings and accessories, with the lineal feet of pipe, including the length of bends, ells, and the like, and calculate the pressure loss by formula (4). Since the resistance increases as a linear function of V/L and V/L is zero at the tank, the arithmetic mean of the V/L value in formula (5) can be used, that is $(1 + V/2L)$, as the multiplier in formula (4). Since part of this vapor is not formed in the line, but in the pump, the calculated value is slightly on the high side, but safe. A cavitation curve similar to Figs. 9 or 11 for the fuel pump actually used must be available, and the loss should be taken at the liquid delivery value corresponding to the V/L condition in the pump. This will be $\frac{1}{1 + V/L} \times 100\%$ normal delivery. The sum of the line and pump losses is the total loss to be subtracted from ambient pressure in the gas tank at any altitude, to get the minimum pressure in the pump.

For take-off, a high tank temperature such as we have recorded from planes in the desert may now be selected. Since the drop in temperature for evaporation to $V/L = 5$ is less than 10 deg, it is probably not worth while to correct temperature. We may, for usual airplane conditions, assume heat addition to be zero for this part of the fuel system. With this pressure at the pump (ambient pressure in tank – total loss), the V/L which should have developed in the gas can be spotted on Fig. 1. If this value is higher than the V/L found for limiting capacity for the pump, there will be vapor lock; if less, definitely no vapor lock. By cut and try, the exact temperature corresponding to beginning of vapor lock can be found, so far as the pump and suction line are concerned. No doubt, later, we shall be able to work out methods for unique solution.

For altitude, the situation is slightly different; we may safely assume that supersaturation is not likely to be present at any time in the tank; gasoline will evaporate freely as the plane climbs, taking dissolved air with it, and provided the tank is adequately vented. For a lined or bulletproof tank, heat loss is very low, and there will be very little cooling of gasoline during a fast climb. As a result, we can assume that the gasoline will stabilize as it climbs, to the ambient pressure. The vapor evolved is freed in the tank and vented, therefore, does not appreciably affect the fuel line and pump. The vapor and air evolved in the fuel line and pump will be the difference between that evolved by the tank ambient pressure and that evolved by the final pressure in the pump. If we now make a plot – difference of V/L ratio at the two pressures (and temperatures) versus altitude, we can select the critical altitude where vapor lock should begin; this will occur at the difference in V/L values for tank and pump, which is equal to the limiting V/L value for the pump as figured by equation (5). If this altitude is not enough to satisfy the requirements, then: (1) Pipe losses must be further lowered; (2) Pump intake losses must be lowered; (3) A larger pump must be used.

It is impossible in the ordinary space of a single paper to do this subject proper justice. But the outline is given, and the more detailed consideration can be given in separate papers, bringing in more data, and some refinements looking toward higher accuracy. It is understood that practical experiences with troublesome cases are to be given by others.

■ Conclusions

1. The vaporization characteristics of any gasoline can be determined. The hydraulic system must be designed specifically to meet these characteristics.

2. The pipe and fitting resistances are quite fully known, including that for flow of mixtures of gas and liquid.

3. The ratio of pump volume rate of delivery to required maximum volume rate of liquid gasoline determines the limiting V/L for successful operation without vapor lock from the fuel system.

4. From the foregoing, the critical vapor-lock tank temperature and maximum altitude can be determined.

5. Vapor lock beyond the fuel pump is chiefly a matter of elimination of vapor or gas at measuring points, and no heat addition before the metering operation.

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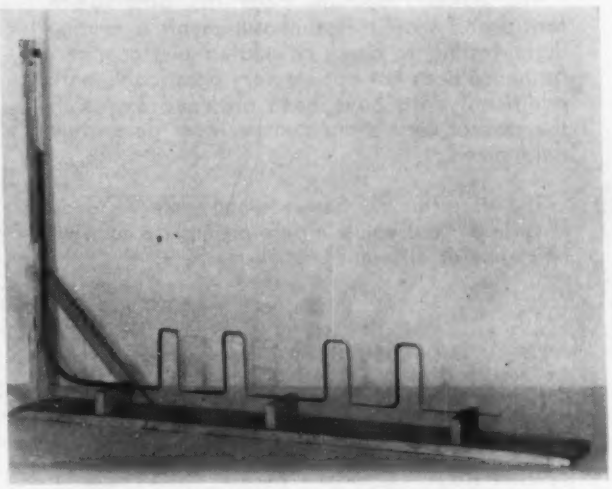

■ Fig. 12 – Airtrap

Apparatus showing gas separation vapor lock. Right end of apparatus is open to atmosphere, and, although there is considerable head at other end, no flow takes place. Summation of vertical risers equals head of fluid in left-hand riser

FLIGHT TESTING with a R

by GEORGE W. BRADY

Propeller Division, Curtiss-Wright Corp.

THE two items which must be determined in order to calculate the performance of an aircraft are horsepower required and thrust horsepower available. To date, no satisfactory method of measuring either of these quantities to check the computed performance has been developed. If either quantity could be measured to a reasonable accuracy then the other would also be known. Hence, the desirability of a method to measure either horsepower required or available is obvious.

This paper describes some of the work accomplished during the past several years by the Curtiss-Wright Propeller Division in endeavoring to make a measurement in flight of thrust horsepower available. To date, this work has been based on the measurement of propeller shaft thrust, which can then be used to calculate an apparent or "shaft thrust" efficiency. Although a method of this sort does not give a complete and final answer to the question of thrust horsepower available or propulsive efficiency, which in the average case will be different from "shaft thrust" efficiency, it does provide a means of adding considerably to our knowledge of propeller behavior and as such is a very useful experimental tool.

[This paper was presented at the SAE National Aircraft Engineering-Production Meeting, Los Angeles, Calif., Sept. 30, 1943.]

IN this paper, Mr. Brady discusses thrustmeter design, calibration, flight instrumentation and procedure, test results, body-propeller interaction; and summarizes the usefulness of the thrustmeter.

The author concludes with the observation that use of shaft thrust measurements in routine flight testing to check calculated airplane performance does not appear very practicable until additional data have been obtained by which the several corrections can be more accurately determined.

The thrustmeter, however, he asserts, does have important value in determining the relative efficiency of different propellers.

■

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To obtain apparent or "shaft thrust" efficiency, three principal items must be known: shaft thrust, shaft horsepower, and airplane velocity, which are then used in the following formula:

$$\eta_a = \frac{\text{Shaft Thrust} \times \text{Airplane Velocity}}{\text{Engine Power}} = \frac{T_s V}{P}$$

Fortunately, in attacking the problem of measuring the above three quantities, the direct measurement of engine power was possible, since the development of torquemeters for a majority of American high-horsepower aircraft engines had already been accomplished. Measurement of airplane velocity is, of course, straight forward except possibly at high forward speeds where compressibility effects on the air-speed indicator must be allowed for, so that the instrumentation problem was reduced to development of a suitable shaft thrustmeter.

In undertaking this development, a review was made as to the possible methods for measuring propeller shaft thrust. It was found that a few thrustmeters of various types had been built and some data published regarding them. The most obvious and possibly the most desirable way would be to incorporate the thrustmeter in the design of the propeller hub. For fixed-pitch propellers this might not be too difficult, but with controllable-pitch propellers it is not a simple matter, as a complete redesign of the propeller is required in order to provide space for the thrust indicator. Also, the necessity of having a torque connection sufficiently rigid to permit driving the propeller and at the same time sufficiently flexible in the thrust direction to permit an accurate measurement of the thrust, makes design of a satisfactory mounting very difficult.

An alternate method of measuring shaft thrust is by determining the tensile stress in the propeller shaft. Because of the close coupled design of most aircraft-engine nose sections and propeller shafts and hubs there is no space sufficiently long where an extensometer or other tension measuring gage can be installed. Also, the necessity of having transfer rings or similar equipment to transmit the indication from the rotating shaft to the fixed structure complicates the problem to the point where this method of attack is not satisfactory.

Engines with planetary spur-gear reduction gears usually have a propeller shaft separately mounted in bearings of which only one is fixed in the thrust direction. Hence, for this type of engine if an accurate method of measuring the fore-and-aft thrust on this bearing were available the

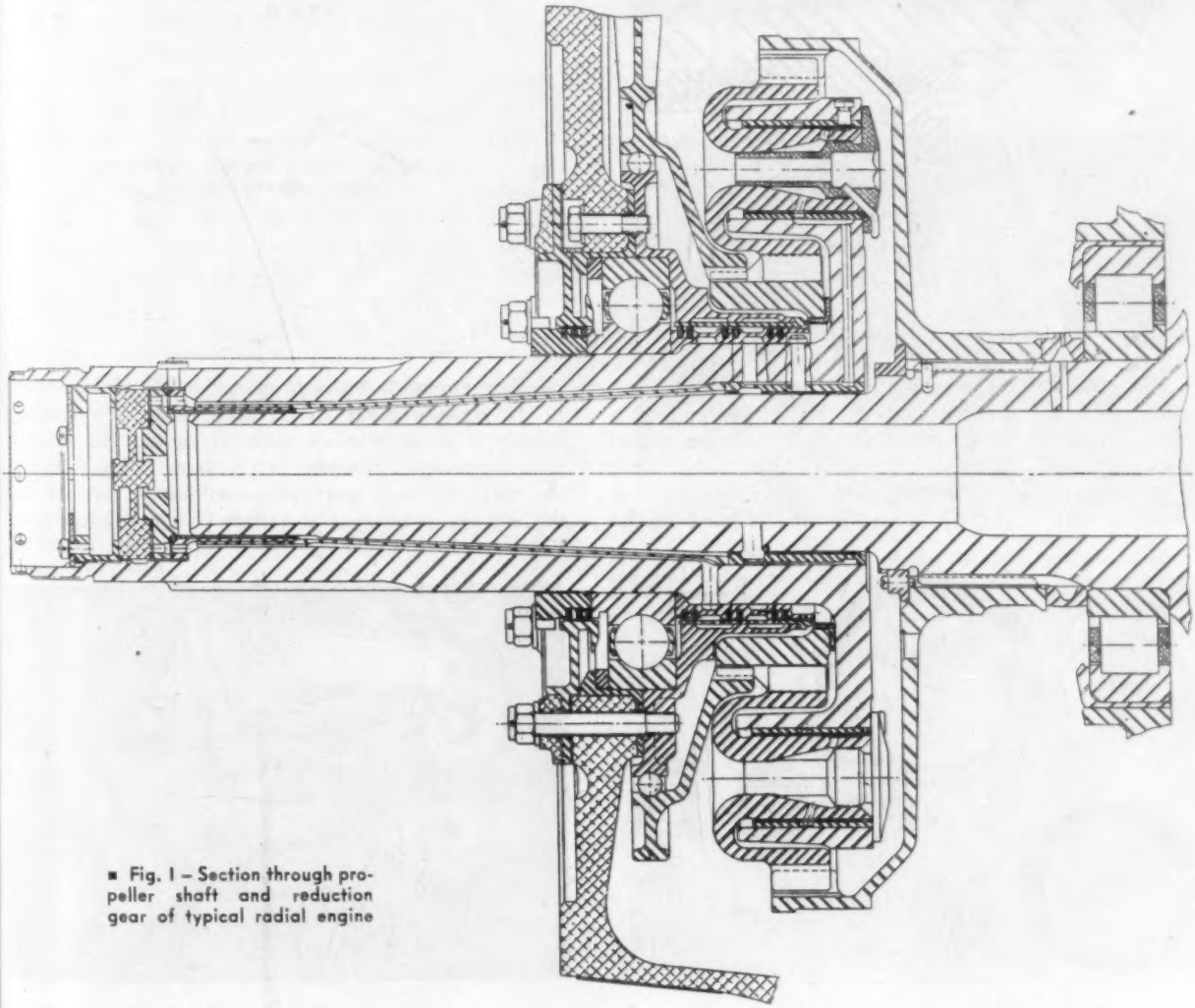
PROPELLER THRUSTMETER

shaft thrust instrumentation problem would be solved. This method of approach is the one which was adopted for the thrustmeter developed as part of the program described in this paper.

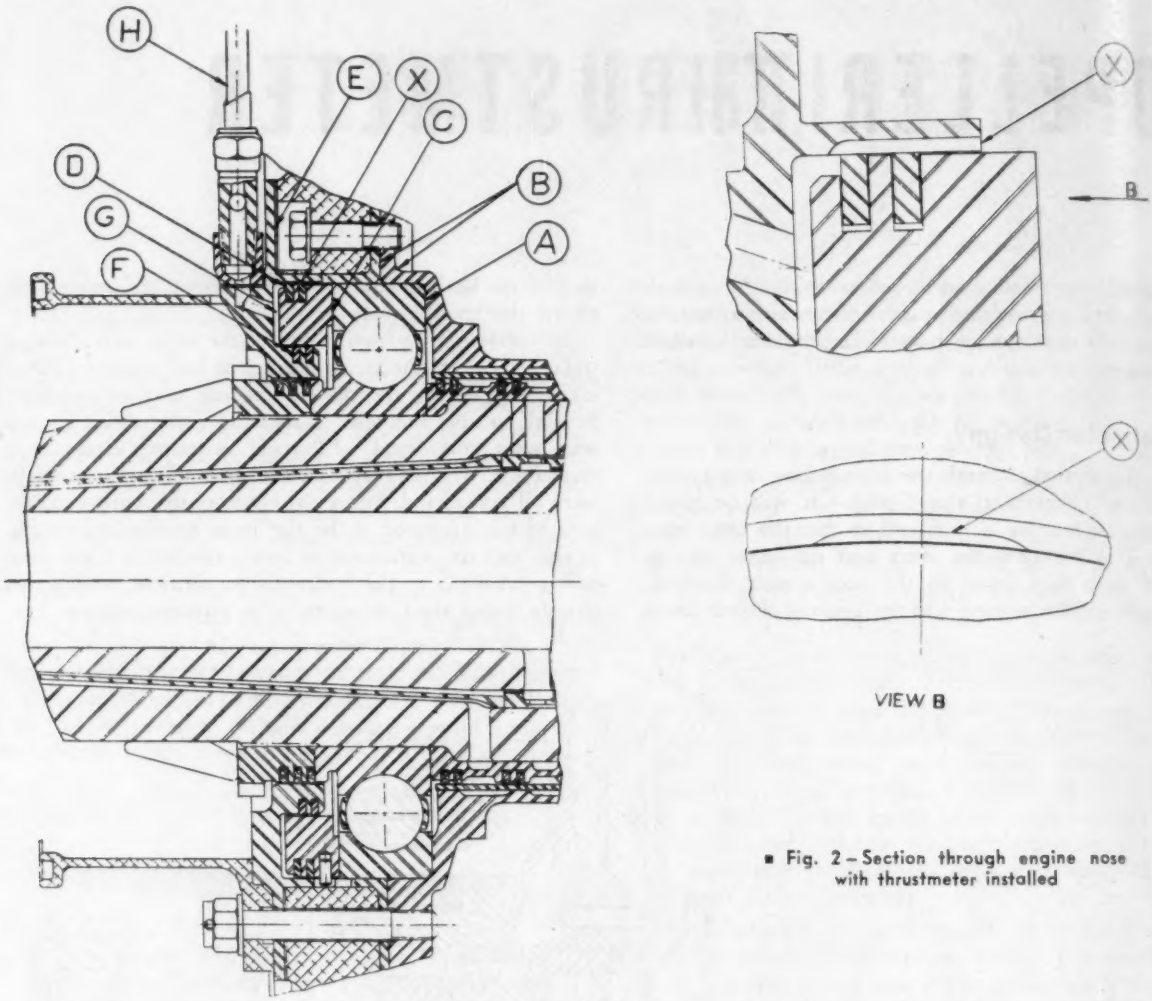
Thrustmeter Design

Fig. 1 is a section through the nose section of a typical radial engine (Wright R-1820-G200). It will be noted that the thrust bearing is mounted so that the inner race is clamped to the propeller shaft and the outer race is supported in a steel insert in the nose section housing. The fit between the bearing and the insert is slightly loose so that the bearing can move by a very slight amount between the retaining shoulders.

By removing the front flange of the insert and allowing the outer race of the thrust bearing to butt against a thrust measuring device, the desired reading may be obtained. Several possible methods of measuring the thrust in this way were considered. Electrical resistance elements, a mechanical weighing system, and hydraulic pressure cells were all considered, but a hydraulic piston with variable-area orifice appeared to be the most satisfactory system. It also had the advantage of being similar in force measuring principle to the hydraulic torquemeter which was already being used successfully in aircraft engines.



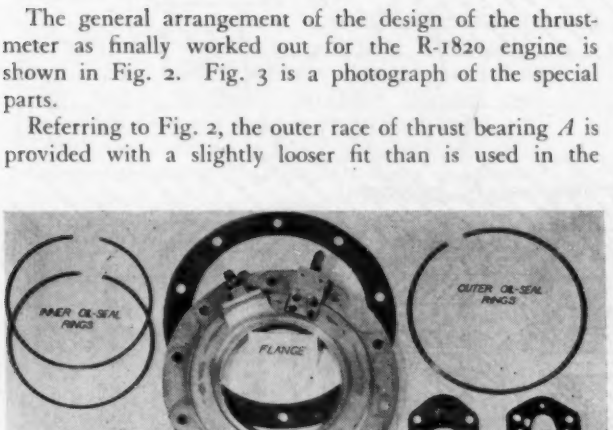
■ Fig. 1 – Section through propeller shaft and reduction gear of typical radial engine



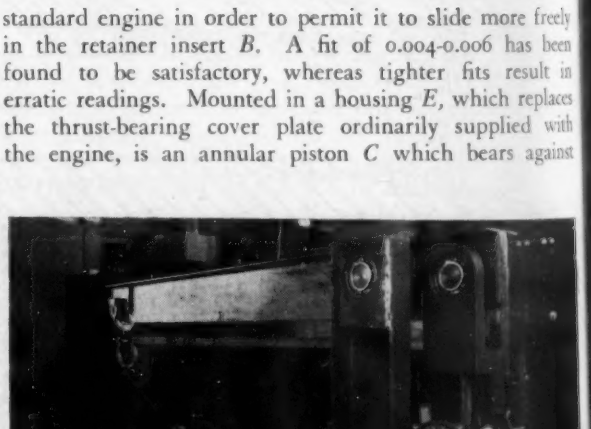
■ Fig. 2 – Section through engine nose with thrustmeter installed

The general arrangement of the design of the thrustmeter as finally worked out for the R-1820 engine is shown in Fig. 2. Fig. 3 is a photograph of the special parts.

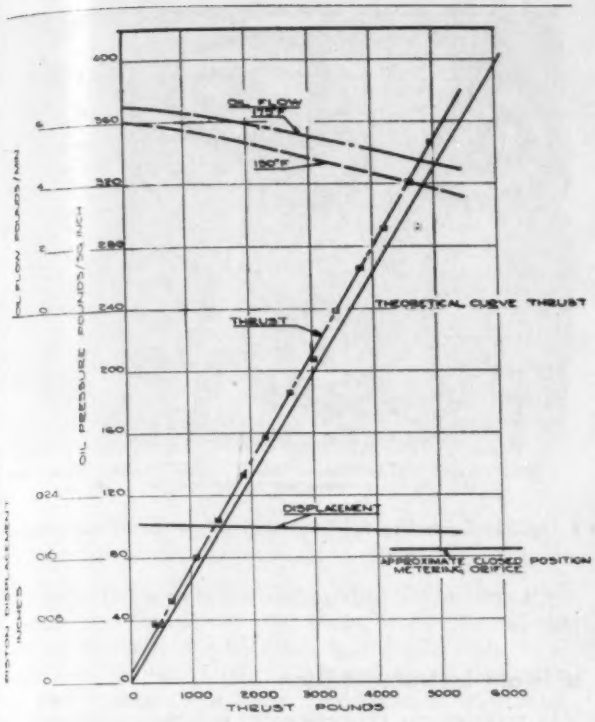
Referring to Fig. 2, the outer race of thrust bearing *A* is provided with a slightly looser fit than is used in the standard engine in order to permit it to slide more freely in the retainer insert *B*. A fit of 0.004-0.006 has been found to be satisfactory, whereas tighter fits result in erratic readings. Mounted in a housing *E*, which replaces the thrust-bearing cover plate ordinarily supplied with the engine, is an annular piston *C* which bears against



■ Fig. 3 – Detail components of the thrustmeter



■ Fig. 4 – Laboratory test rig for thrustmeter



■ Fig. 5 – Calibration of thrustmeter by laboratory test rig

the outer race of the thrust bearing. Two sets of piston rings *D* and *F* provide the seal for a cylindrical space *G* between the piston and the housing into which oil under pressure can be pumped. A series of six orifices *X* are provided in the housing to permit the piston to find a position where the pressure from a constant-displacement pump supplied through supply duct *H* will just balance the thrust force on the thrust bearing. Then, by measuring the pressure in the cylinder with a gage and knowing the area of the piston the thrust can be computed.

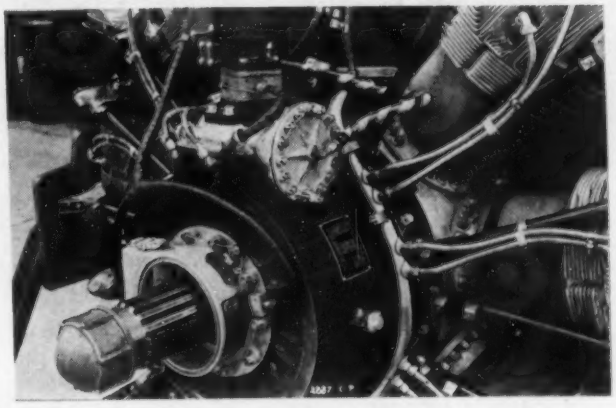
The oil supply for the thrustmeter is provided by a positive-displacement gear pump which, in the case of the present tests, is arranged as a "sandwich" between the governor and the governor pad. The pressure in the thrustmeter is read on a pressure gage which in the flight tests had a full-scale reading of 30-180 psi, the dial being graduated in 1-psi increments. This provided for thrust measurements up to about 2500 lb. In the laboratory tests where thrusts up to 6000 lb were measured, a gage with a range of 0-550 psi was used. Electrically heated oil lines were used to prevent a lag in thrustmeter reading due to congealing.

■ Thrustmeter Calibration

In practice it has been found that several corrections must be applied to the reading obtained from the thrust gage to obtain the propeller shaft thrust. These were determined in the course of about one year of airplane testing. Initially, a calibration rig for laboratory use was constructed. Fig. 4 shows the general arrangement of this rig, and Fig. 5, the results of a typical run. This rig was used in development of the initial design of the thrustmeter, which was done in connection with a development contract with the Army Air Forces.

In making the first flight installation, assistance was obtained from the Navy's Bureau of Aeronautics in the form of an F2A-2 airplane equipped with a Wright R-1820 engine with 0.667 gear ratio. Fig. 6 shows this airplane with the thrustmeter and a test propeller installed. Fig. 7 shows a close-up of the engine nose with thrust- and torquemeters installed. In the initial operation of this airplane, rather erratic and nonrepeatable results were obtained with the thrustmeter. A careful review was therefore made of all factors which might affect the results. Also a special ground calibration rig was made to permit application of a known thrust to the propeller shaft while the engine was running. The arrangement of this rig is shown in Fig. 8. The tension bar and lever loading system by which a known thrust was applied to the rotating shaft is evident in this figure.

■ Fig. 6 – Airplane used in flight tests



■ Fig. 7 – Close-up view of thrust- and torquemeters installed on engine

■ Fig. 8 – Test rig used for calibrating thrustmeter on airplane

Results obtained with this rig are shown in Fig. 9, which is the calibration with which all flight-test readings were corrected. The various points plotted on this curve are for rpm's varying from 1800 to 2700. In addition, periodic checks of the thrustmeter were made to be sure the original calibration still held. It was found in the course of this testing that in addition to making the corrections for individual instrument errors it was necessary to apply a further correction for the particular engine used because of the construction of the propeller shaft. It will be noted in Fig. 1, that engine lubricating oil under pressure travels through the crankshaft to a bearing in the propeller shaft and from there into a conical annular space between the two shafts. This oil pressure acts on the internally tapered propeller shaft as a piston causing a force in the forward direction. Also due to the rotation of the shaft there is a centrifugal pressure built up in the oil in passing through the exit hole in the shaft to the point where the pressure is measured. This pressure, which is calculated as the difference in centrifugal pressure between the outside and inside diameters of the shaft, is deducted from the measured pressure. The value of this correction for the test installation is given in Fig. 10.

One factor which was checked very carefully during the calibration of the thrustmeter on the engine was the possible effect of deflection in the engine reduction gear to give a fore-and-aft load. However, no effect of this sort was observed in the type of engine used in these tests.

A typical procedure in correcting the various readings to obtain the correct shaft thrust is as follows:

1. Correct observed thrustmeter pressure for gage calibration.

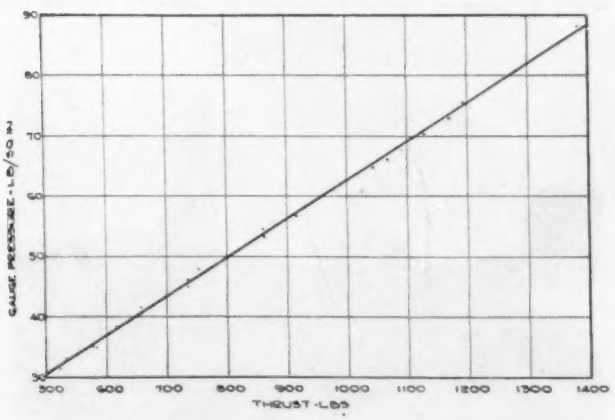
2. Using the corrected thrustmeter pressure, obtain the indicated thrust from the thrustmeter calibration, Fig. 9.

3. Correct observed intershaft pressure for gage calibration.

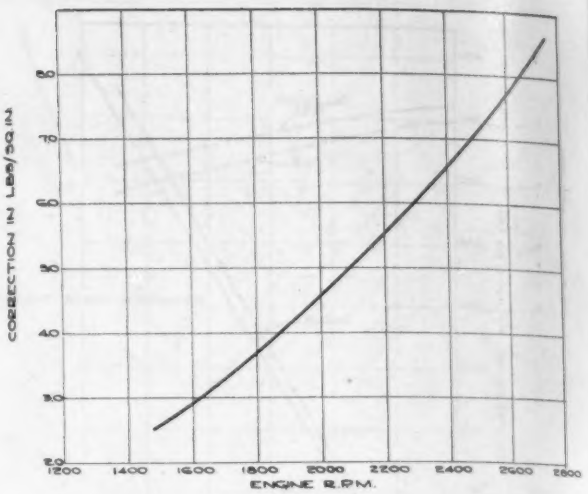
4. Read correction to intershaft pressure for centrifugal boost for the observed engine speed, Fig. 10.

5. Calculate thrust on propeller shaft due to oil pressure effect by multiplying the effective area (4.2 sq in. for the test installation) by the difference between (3) and (4).

6. Calculate the component along the propeller shaft axis of the propeller and propeller shaft weight using the observed inclination of the thrust line from the horizontal.



■ Fig. 9 – Calibration of thrustmeter on airplane with running engine



■ Fig. 10 – Correction to observed thrust for centrifugal pressure

7. Calculate the corrected shaft thrust as (2) minus (5) plus (6).

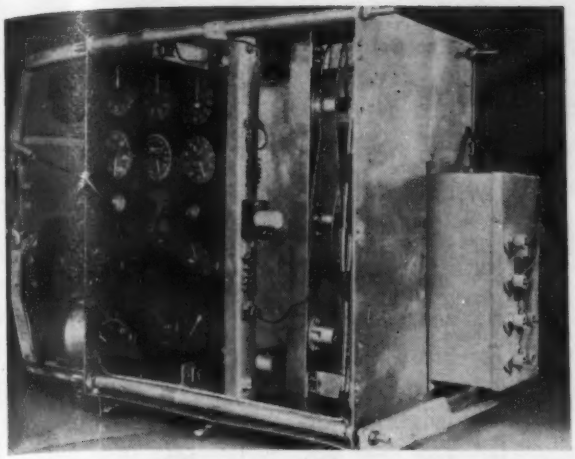
■ Flight Instrumentation

In applying the thrustmeter to a flight-test program, it was desired to determine shaft thrust and shaft thrust efficiencies from climbing air speeds of about 150 mph to flight speeds of about 400 mph. Since the airplane available, which was a Navy Brewster F2A-2 with a two-speed R-1820 engine, had a level flight speed at critical altitude of approximately 310 mph at the power to be used in these tests, to achieve the higher end of the speed range required diving the airplane. This meant that simultaneous readings were required of all instruments as the desired speed and altitude would only be encountered momentarily. A small photo-observer was therefore constructed to meet this need. This piece of equipment is shown in Fig. 11. The following instruments were mounted on the photo-observer panel:

Air-speed indicator
Tachometer
Altimeter
Oat
Bmep gage
Manifold pressure gage
Thrustmeter pressure gage
Intershaft pressure gage
Gyro inclinometer
Pitch indicator
Clock
Counter
Tel-light
Attention light

The accuracy of the various readings is believed to be as follows:

1. Shaft thrust in pounds accurate to ±20 lb.
2. Horsepower accurate to ±1%.
3. Engine rpm accurate to ±½%.
4. Indicated air speed to ±1%.
5. Pressure altitude accurate to ±25 ft.
6. Outside air temperature at 100 mph ias accurate to ±3 F.



■ Fig. 11 – Photo-observer panel used in tests

7. Propshaft inclination accurate to ±2 deg.

8. Blade angles at 42-in. station accurate to ±0.25 deg.

Besides the thrustmeter the only instruments in the above list which were sufficiently nonstandard to be mentioned were the pitch indicator and the gyro inclinometer. The pitch indicator was a specially developed instrument to give a reading of blade angle. The general arrangement of the system, which consists of a transmitter on the propeller and the indicating instrument on the photo-observer panel is shown in Fig. 12. The transmitter gives an impulse for each fraction of a degree of pitch change and the indicator which has an escapement mechanism will count the number of pitch-change increments in either direction. By determining the amount of pitch change from a predetermined blade angle, usually low pitch, the actual blade angle can be determined.

The gyro inclinometer used in these tests was a modification of a standard Sperry directional gyro mounted on its side so as to give the inclination from the horizontal of the propeller shaft axis. With this reading, the component of the propeller and propeller shaft weight along the axis can be calculated.

■ Flight Procedure

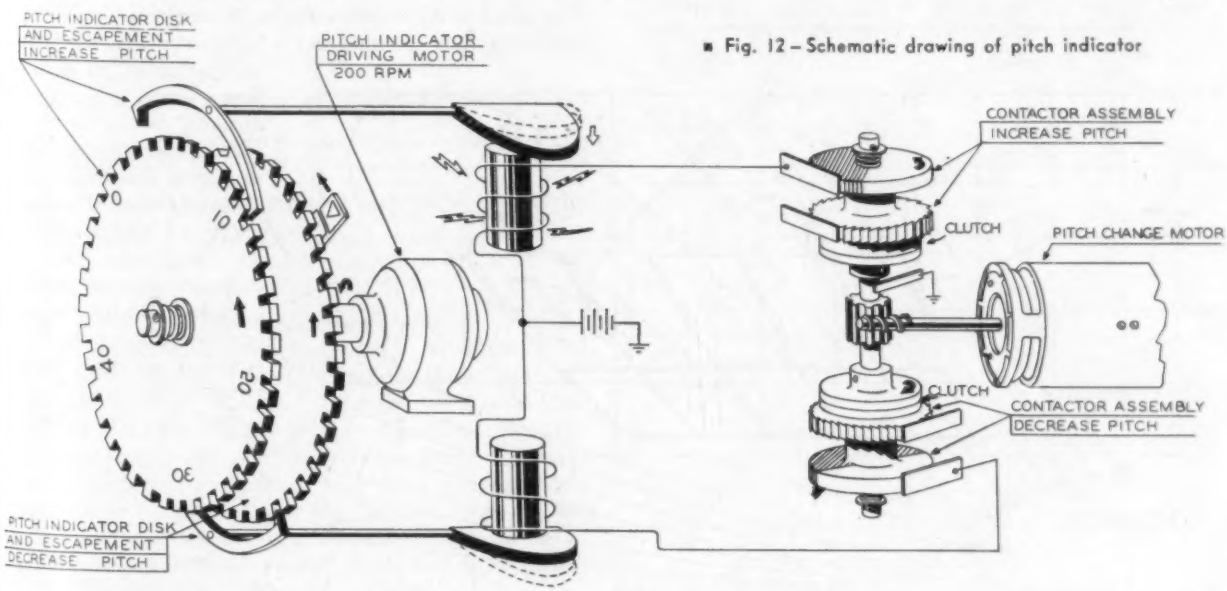
The procedure of the pilot in a typical level flight run was to adjust power and rpm to the specified conditions after locating the desired density altitude. As soon as a steady state was reached, the photo-observer was started and allowed to operate automatically for at least 3 min. The frequency of operation of the photo-observer camera was four frames each 5 sec.

Climb tests were conducted so that the specified power and rpm conditions were reached at the specified altitude and air speed. This was accomplished by starting the climb about 2000 ft lower than the desired altitude, setting power, rpm, and velocity at the specified values and adjusting manifold pressure so that the desired bmep would be obtained at the specified altitude. Holding these conditions, the airplane was climbed through this altitude and for another 1500 ft above, the photo-observer operating continuously.

Tests above maximum level flight speeds were made by diving the airplane. These tests were made in a manner similar to the climb tests except that the tests were started above the altitude at which it was desired to obtain the specified air speed. The approximate air speed was obtained as rapidly as possible in the run by diving quite steeply. Several thousand feet above the specified altitude, when the desired air speed had been obtained, the dive was flattened out somewhat, retaining sufficient angle so that constant indicated air speed was maintained.

■ Test Results

During the flight tests conducted with the thrustmeter to date, no attempt has been made to cover the complete range of propeller operation even for a single design. Rather a single power, the maximum that the engine could develop at the lowest of the test rpm's, was selected and a series of engine rpm's at constant power was then checked



■ Fig. 12 – Schematic drawing of pitch indicator

over the speed range from 160 to more than 400 mph. The power used was 850 hp at 14,000 ft and the engine rpm's were 2100, 2400, and 2700. Table 1 gives values for the various propeller coefficients. Their location on a propeller power coefficient chart is shown in Fig. 13.

The level flight and dive tests were conducted at approximately 14,000-ft pressure altitude. The climb tests, because of the slightly lower critical altitude raising of the engine in high blower, were conducted at 12,000-ft altitude. However, all results were corrected by cross plotting to 14,000-ft density altitude. Tests results obtained with one typical propeller having blades of conventional design are included in this paper. This propeller is Curtiss model C532D with 89306-22S blades.

The characteristics of this propeller are given in the following tabulation while the blade data are given in Fig. 14.

Propeller model	C532D
Blade design	89306-22S
Diameter	10 ft 0 in.
Number of blades	Three
Activity factor	89.9
Thickness ratio at 0.75R	0.080

Shaft thrust for this propeller is given in Fig. 15. Apparent or shaft thrust efficiency is given in Fig. 16. In both curves the data are plotted against true air speed for the three test rpm's of 2100, 2400, and 2700.

In Fig. 16, it will be noted that at 160 mph, the lowest air speed used in the tests, 2100 rpm, gives the lowest efficiency and 2700 the highest. At 250 mph the efficiency for all three engine speeds is about the same. Above this air speed, the 2100- and 2400-rpm curves continue closer together but that for 2700 rpm drops appreciably.

■ Body-Propeller Interaction

It will be noted in Fig. 16 that the apparent or shaft thrust efficiencies reach a maximum of about 96% at a speed of slightly over 300 mph. This is as compared to a maximum calculated propulsive efficiency of the order of 86% and a measured value in low-speed-propeller tunnel tests of seldom over 86-89% for conventional types of installations. The difference between these two efficiencies is due to interaction between the body and the propeller.

This propeller-body interaction, which is discussed in the references,[1, 2, 3] is primarily due so far as the propeller thrust is concerned to the pressure gradient in the slipstream acting on the body to cause a mutual reaction. According to Glauert,[3] which contains a detailed discussion of this effect, the increases in thrust and in body drag are equal and "may be presumed to be proportional to the apparent thrust T_a of the propeller and to the maximum cross-sectional area S of the body," thus leading to:

$$\Delta T_B = \Delta D_B = f_2 \frac{4ST_a}{\pi D^2}$$

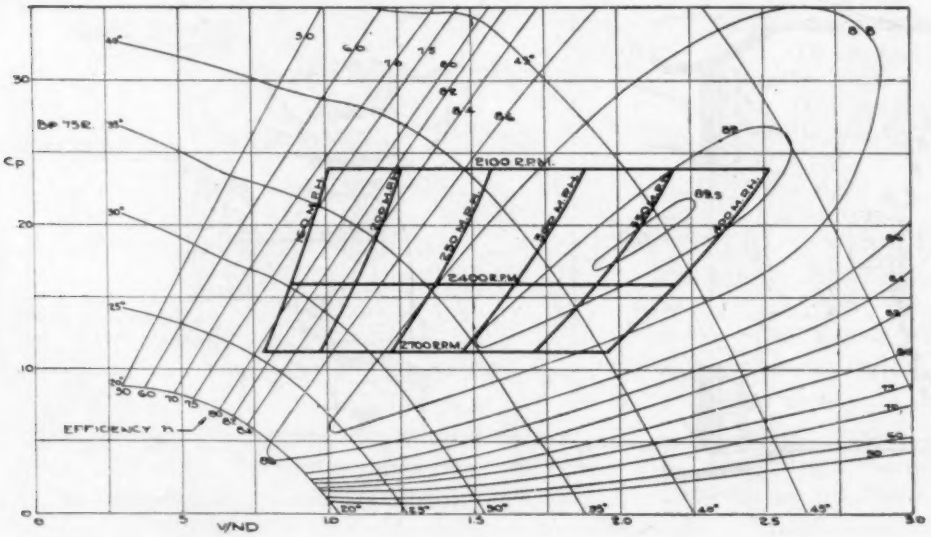
where T_a is the apparent or shaft thrust and f_2 is a factor depending mainly on the shape of the nose of the body. The effect of the body on torque characteristics of propeller should be kept in mind but it is believed that the magnitude is small. It is therefore omitted in the present analysis.

A second effect on thrust is due to the drag of the hub or spinner. For example on a liquid-cooled tractor installation having a propeller spinner, the hub is enclosed within the body but as the spinner is mounted on the propeller a part of the body drag is carried by the propeller.

Table 1 – Range of V/nD and Ehts[a] Covered by Flight Tests

Rpm	2,100	2,400	2,700
Hp	850	850	850
Altitude	14,000	14,000	14,000
Bmep	176	154	137
C_p	0.239	0.159	0.112
V/nD for			
V = 160	1.007	0.880	0.782
V = 200	1.26	1.10	0.978
V = 250	1.57	1.374	1.22
V = 300	1.89	1.65	1.465
V = 350	2.20	1.924	1.710
V = 400	2.52	2.20	1.955
πnD	732	839	941
Ehts[a] for			
V = 160	810	918	1,024
V = 200	831	937	1,039
V = 250	862	963	1,060
V = 300	902	996	1,091
V = 350	945	1,035	1,129
V = 400	991	1,080	1,166

[a] Effective helical tip speed.

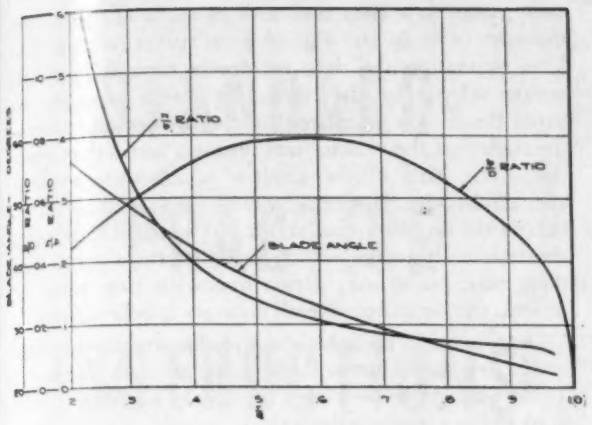


■ Fig. 13 – Cp versus V/nD chart with test points superimposed

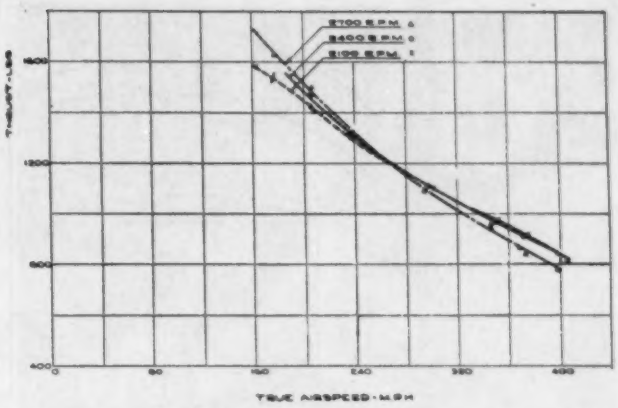
[1] See NACA Technical Report No. 219, 1925, "Some Aspects of the Comparison of Model and Full-Scale Tests," by D. W. Taylor.

[2] See "Aircraft Propeller Design," by F. E. Weick, McGraw-Hill Book Co., Inc., New York, 1930.

[3] See "Aerodynamic Theory," W. F. Durand, editor, Vol. IV, Division I, "Airplane Propellers," by H. Glauert, Julius Springer, Berlin, 1935.



■ Fig. 14 – Characteristics of blade used in tests



■ Fig. 15 – Flight-test results – variation of apparent or "shaft" thrust with air speed

This would cause a reduction in apparent thrust. For an aircooled installation, however, the result is very likely opposite, as the hub and inner portion of the propeller are working in a region of considerably reduced velocity as compared to the free airstream. Hence, the hub drag would be lower and the apparent thrust higher.

A third interaction is important if the propeller is working in a region where the velocity is appreciably reduced as compared to free air. Such a condition might occur in the case of a propeller operating ahead of a large body or wing. However, in the majority of modern installations, the working part of the propeller is operating in air which is substantially undisturbed and hence this effect is small.

In addition to the effect on the propeller there is the added drag effect due to the increased velocity over the fuselage or nacelle and such part of the wing and tail as lies behind the propeller. This drag increment does not cause any increase in propeller thrust but must be deducted from this thrust in obtaining propulsive efficiency.

The above considerations may be represented as follows:

$$\begin{aligned} T_a &= T_e \pm \Delta T_H + \Delta T_B \\ D_a &= D_e \pm \Delta D_H + \Delta D_B + \Delta D_S \\ T_p &= T_a - (D_a - D_e) \\ &= T_a \pm \Delta T_H - \Delta T_B - \Delta D_S \end{aligned}$$

where:

T_a = Apparent or shaft thrust
T_e = Free air thrust
$\Delta T_H = \Delta D_H$ = Thrust and drag increments respectively due to hub and spinner drag
$\Delta T_B = \Delta D_B$ = Thrust and drag increments respectively due to pressure gradient in the slipstream
T_p = Propulsive thrust
D_a = Apparent drag or airplane drag in the presence of the propeller
D_e = Airplane drag without the propeller
ΔD_S = Increment due to increase of velocity in slipstream acting on parts of airplane behind the propeller

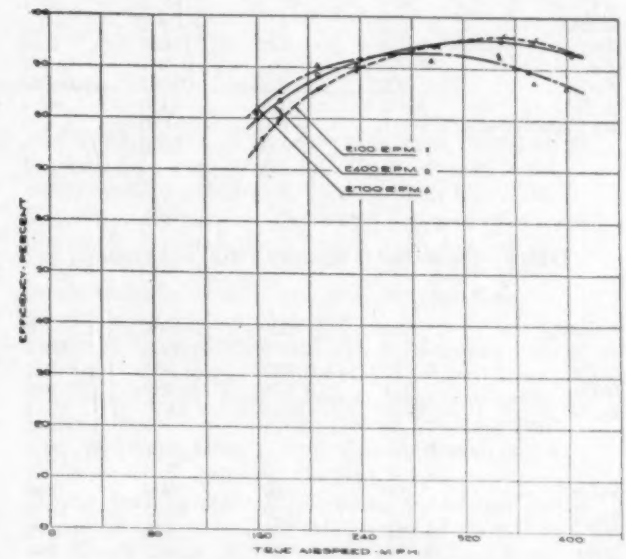
Evaluation of the above effects cannot be made at the present time with very much assurance, as experimental data to permit calculations of the various thrust and drag increments are very meager. A considerable amount of testing to determine apparent and propulsive efficiencies was accomplished about 10 to 15 years ago,[2, 4, 5] but the types of propellers and bodies used differ so much from those used in the present tests that the data do not appear applicable. Additional experimental work in the wind tunnel in which both propulsive thrust and shaft thrusts are measured is very desirable.

In recent wind-tunnel testing, the propeller-body question has been neatly sidestepped by taking net readings on the body with and without propeller. This has been done because the propeller efficiency with which the airplane designer preferred to work was propulsive efficiency. While this has been very convenient it does not require a full knowledge of the effect of the propeller and hence it is quite possible that something less than the most efficient design has in some cases resulted.

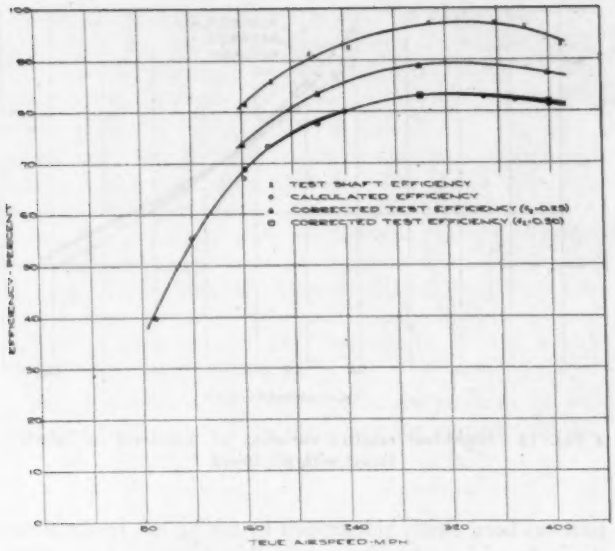
Even though experimental data are not available, from a theoretical standpoint, it is possible to make an approximate calculation of the thrust increments which make up the difference between apparent and propulsive thrusts and efficiencies. This has been done for one of the test rpm's, and the results are plotted in Fig. 17, for comparison with the test results and also with a calculated efficiency. This latter curve is determined by the standard Curtiss method for propeller efficiency calculation.

[4] See NACA Technical Report No. 235, 1926, "Interaction between Air Propellers and Airplane Structures," by W. F. Durand.

[5] See British Aeronautical Research Committee Reports & Memoranda No. 1445, 1931, "Analysis of Experiments on the Interference between Bodies and Tractor and Pusher Airscrews;" and No. 1528, 1932, "Interference between Bodies and Airscrews, Part II," by C. N. H. Loch and H. Bateman.



■ Fig. 16 – Flight-test results – variation of apparent or "shaft" thrust efficiency with air speed



■ Fig. 17 – Comparison of propulsive efficiencies determined by test and calculated for 2400 rpm

The value ΔT_H in the expression for propulsive thrust is assumed in this analysis to have a value of zero. This is believed justified, as the hub is of conventional size and is located in front of a fairly large body.

The pressure gradient increment ΔT_B is calculated as a constant percentage of T_a in accordance with the Glauert formula given above. An estimate of the magnitude of f_2 indicates that it should be between 0.25 and 0.5. Both of these values have been used with an assumed effective body diameter of 60 in. and Fig. 17 gives curves for both values.

In estimating the drag increment term ΔD_S the slipstream velocity for the various air speeds using the measured thrust was calculated by the momentum theory on the basis that the velocity was uniform across the propeller disc. The parts of the airplane which were considered affected by the slipstream were taken as those directly behind the propeller disc, and it was assumed that ultimate theoretical slipstream velocity acted on these parts. While these assumptions may depart somewhat from actual conditions, their effect on overall accuracy is believed small.

Detail steps in the calculation of the propulsive efficiency by the standard Curtiss method are given in Table 2. In Table 3 are given the values for the ΔT_B and ΔD_S corrections to the apparent efficiency.

It will be noted in Fig. 17 that agreement between the propulsive efficiencies determined by the two methods is very good for the f_2 value of 0.5. For f_2 equals 0.25 the agreement is not so good although the curves are parallel to the calculated propulsive efficiency curve.

It should be considered that a modified method of determining the propulsive efficiency from low-speed-propeller tunnel tests might raise the calculated curve to be in better agreement with the 0.25 f_2 curve. Hence, on the basis of analyzing only one set of test data, it is not possible to say what the exact values of the corrections should be. However, the fact that the two methods give curves which are substantially parallel is evidence that our methods of correcting for Mach number and Reynolds number effects are relatively accurate within the limits of this analysis.

Table 2 – Calculated Efficiency – Curtiss Standard Method

Velocity, tas	160	220	300	400
Altitude, ft	14,000	14,000	14,000	14,000
Hp	850	850	850	850
Rpm—Engine	2,400	2,400	2,400	2,400
Rpm—Propeller	1,600	1,600	1,600	1,600
C_p	0.1594	0.1594	0.1594	0.1594
V/nD	0.880	1.208	1.650	2.200
η Chart, %	78.5	86.0	88.5	89.0
α, deg	9.2	5.5	2.5	−0.2
$F\alpha$	1.255	1.132	1.051	1.00
F_B	0.961	0.961	0.961	0.961
F_T	0.902	0.969	0.987	0.964
η, %	68.1	80.1	84.0	82.4
$\Delta\eta$ (Interference)	−1.0	−1.0	−1.0	−1.0
η_p, %	67.1	79.1	83.0	81.4

Table 3 – Corrections to Apparent Thrust and Efficiency

$f_2 = 0.25$

Velocity, tas	160	220	300	400
η_a, %	81.0	91.0	96.5	94.8
$f\Delta T_B$	0.935	0.935	0.935	0.935
$f\Delta D_S$	0.971	0.978	0.982	0.986
η_p, %	73.6	83.3	88.6	87.4

$f_2 = 0.50$

Velocity, tas	160	220	300	400
η_a, %	81.0	91.0	96.5	94.8
$f\Delta T_B$	0.875	0.875	0.875	0.875
$f\Delta D_S$	0.971	0.978	0.982	0.986
η_p, %	68.9	77.9	82.9	81.9

■ Usefulness of Thrustmeter

Use of shaft thrust measurements in routine flight testing to check calculated airplane performance does not appear very practicable until additional data by which the several corrections can be more accurately determined have been obtained. However, it is believed that the more precise knowledge of the actual behavior of the propeller which can be obtained by thrustmeter tests should lead the way to some improvements in airplane performance.

The above discussion has been concerned primarily with the use of the thrustmeter to determine horsepower available. An instrument of this type has other uses, however. Of these, perhaps the most important is in determining the relative efficiency of different propellers. Because of the ΔT_B term, it is doubtful if propellers which have widely differing planforms and thickness ratios on the inboard sections can be compared too accurately by the thrustmeter method but major variations in outboard planform, thickness ratio, pitch distribution, airfoil section, and tip shape can be successfully studied.

Another possible use is in airplane performance tests in which it is desired to glide at or near zero thrust to determine the L/D characteristics of the airplane. Because the type of thrustmeter described in this paper is not suitable for reverse thrust measurements, it would be desirable to make tests of this type with a small but known positive thrust on the propeller.

Other uses for an accurate thrustmeter will undoubtedly be developed as experience is gained with this instrument. It is, therefore, believed that while it is a rather specialized type of instrument the propeller shaft thrustmeter will find a useful and valuable place in flight-test procedures.

STANDARD-PRACTICE INSTRUCTIONS

Report of Subcommittee, Maintenance Methods Coordinating Committee, SAE Transportation and Maintenance Activity, Submitted to the Vehicle Maintenance Section, Division of Motor Transport, Office of Defense Transportation

[Presented as a paper by J. Willard Lord at the War Transportation & Maintenance Meeting, New York City, Oct. 8, 1942.]

■ Application of Standard-Practice Instructions as developed by White Motor Co., including a steel file box for proper filing of Instructions

THIS war has been defined as a "War of Production" – a war of being able to produce and put into the field more and better equipment than that of our opponents. Also, it is a struggle of maintenance, for all this equipment we produce is of small use if we fail to maintain it, but permit it to stall on the repair line for lack of thorough lubrication, "stitch in time" adjustments, and accurate major repairs.

The war has created a tremendous demand, not only for production personnel, but for maintenance personnel as well. It has greatly accelerated the whole activity of adult training, and has given us a new and glowing viewpoint of the possibilities of organized training as an essential element of efficient production, operation, and maintenance. It's an element brought to the front under the pressure of the war, but industrial training also has tremendous possibilities in the post-war period, and the need for the very best instruction will be as great then as it is now.

Now more than ever there is the job to be done of teaching, not halfway repairs, but that quality of maintenance which will be in line with the manufacturer's production practices, and which will build back into the equipment, operating miles more nearly equivalent to the original mileage – and not a third or fifth of it.

We are being made to realize that the degree of success to which any preventive maintenance program is carried out is not only how well and how completely you have

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SAE Maintenance Methods Coordinating Committee

W. J. Cumming, chairman, chief, Vehicle Maintenance Section, Division of Motor Transport, Office of Defense Transportation; E. P. Gohn, test engineer, The Atlantic Refining Co.; M. E. Nuttila, superintendent, Motor Vehicles, Cities Service Oil Co.; G. W. Laurie, manager, Automotive Transportation Department, The Atlantic Refining Co.; J. Y. Ray, supervisor, Automotive Equipment, Virginia Electric & Power Co.; S. B. Shaw, automotive engineer, Pacific Gas & Electric Co.; W. A. Taussig, automotive engineer, Burlington Transportation Co.; E. W. Templin, automotive engineer, Los Angeles Department of Water & Power; D. K. Wilson, superintendent, Automotive Equipment, N. Y. Power & Light Corp.; A. M. Wolf, automotive consultant.

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Subcommittee on "Standard-Practice Instructions"

J. Willard Lord, chairman, The Atlantic Refining Co.; Victor L. Alsberge, foreman, Bus Division, New York City Board of Transportation; Frank R. Archibald, technical service supervisor, National Carbon Co., Inc.; Harold L. Carr, general manager, motor truck service, International Harvester Co., Inc.; Robert E. Cummings, valve engineer, Thompson Products, Inc.; Paul E. Friend, chief engineer, Wilkening Mfg. Co.; J. R. Glazebrook, Johns-Manville Sales Corp.; Ervin N. Hatch, senior mechanical engineer, automotive, New York City Board of Transportation; G. D. McCormick, mechanical engineer, Continental Motors Corp.; Charles Mitchell, field engineer, Bendix-Westinghouse Automotive Air Brake Co.; D. Webster Pedrick, Wilkening Mfg. Co.; J. Donald Roach, sales engineer, U. S. Asbestos Division, Raybestos-Manhattan, Inc.; Dayton H. Robinson, production service manager, Bendix-Westinghouse Automotive Air Brake Co.; W. E. Thill, design engineer, Federal-Mogul Corp.; S. B. Shaw, project chairman.

planned the program, but also, "Have you the trained men to do the work planned?" If not, "What are you doing about teaching them?" "How are you going to teach them?" *"What are you going to teach them?"*

Suppose one decides to start a training program. Are mechanics going to be taught this man's ideas of maintenance or that man's ideas of maintenance and repair methods, or are we going to do a comprehensive job of instruction based fundamentally on the knowledge and methods which the automotive industry has found essential in originally building the cars and trucks now in the hands of the Army and the public?

Maintenance is definitely a phase of the automotive industry, and improvement of the general caliber of maintenance and maintenance methods is very much an obligation of the industry rather than a problem for operators to struggle with individually. Operators – whether they be of the Army or of civilian status – are much aware of maintenance and how it affects the life of their equipment, but when we consider maintenance methods and how *quality of maintenance* is to be secured, then we should look to the manufacturer to take the initiative. Who can better define sound rebuilding practices than those who originally built the product and have closely followed it in service?

Educational Job to be Done in Which Automotive Industry Should Take the Lead – There is a huge gap between manufacturer and maintenance mechanic – an enormous difference between the little maintenance information to which the fleet mechanic is exposed, and all the engineering, technical, and practical "know how" information to be found at manufacturers' headquarters.

There is an educational job to be done to reduce this gap between manufacturer and maintenance mechanic. It's a job in which the automotive industry should take the lead to reach maintenance men and make available to them the best of instruction material. Perhaps an assertion such as this raises the question: *"What's the trouble with present maintenance literature?"*

This question has often been asked – sometimes rather defensively, sometimes in true fact-finding attitude. As a general observation, manufacturers of mechanical devices have done a far better job with their sales literature designed to sell the product than they have done with their maintenance literature, which should help the customer get the greatest value from his purchase.

Just stop for a moment and think of the various mechanical devices you have purchased and recall the splendid advertising and sales literature, and then think of the caliber of operating and maintenance instructions – often not more than a slip of paper with a few sketchy instructions referring to parts which cannot be identified for the lack of drawings or illustrations.

In broad aspect the situation reveals that consideration could well be given by manufacturers to some redistribution of the advertising dollar so as to put more effort into helping customers take better care of the machines they purchase.

■ Automotive Maintenance Literature

The automotive industry has been lavish in its gorgeous sales literature and its widespread advertising campaigns, and it has done an excellent job in the preparation of parts catalogs and in keeping these up-to-date, but the caliber of typical maintenance instructions does not exhibit a comparable effort. Simple disassembly and assembly instructions provided by vehicle manufacturers often are very fair, but maintenance data, that is, instructions and test data for diagnosing defective operation and data covering permissible tolerances for refitting or adjusting of worn parts, are seriously lacking.

Part of the difficulty, we suspect, lies in assigning a technical engineering task to those whose chief job and interests are in the distribution of parts and accessories and in sales and advertising, and who are rated by management from this viewpoint.

A technical maintenance instruction filtered through "a parts sales and advertising screen" is likely to come out shorn of its maintenance engineering features, and while its title may have survived, in reality it has become a piece of parts sales literature of disappointing and discouraging maintenance value to the fleet operator's mechanic.

It gives him only minor instructions often expressed in terms of "proper adjustment," and almost certain to include something such as: "If the part is excessively worn, send it to your dealer for repair," or "Replace it with a new genuine or factory-rebuilt unit," and so forth – in brief, a piece of parts sales literature given to describing how the unit functions, and emphasizing the purchase of none but "genuine parts," but totally lacking in how to measure for excessive wear, how to determine good condition, how to rebuild the unit, and how to make specific adjustments.

There is a lack of balance when you consider that, on the one hand, the bus or truck operator spends thousands of dollars for repair shop facilities, tools, and other equipment, and spends thousands more maintaining his fleet purchased for maybe $50,000 or $100,000 from some manufacturer who, on the other hand, has devoted a mere fifty cents or dollar or so per copy to maintenance instructions vital to the proper care of the operator's investment – instructions which should guide the operator wisely and in detail, and which should draw on the wide experience of the manufacturer on how to spend the maintenance dollar to the best advantage.

In advocating that more thought and time and money be put into maintenance instructions, it does not necessarily mean issuing instructions at no cost. Possibly the present general policy of issuing maintenance literature at no cost is one of the fundamental difficulties hampering the development of technical instructions. In all probability, commercial vehicle operators would look on first-class maintenance instructions as a real service to be purchased at a price which would justify and encourage the preparation of the best, and it would help place these data in the hands of those who really need them.

As it now stands, one manufacturer may issue repair instructions in booklet form 4 x 8 in.; another issues his data in a manual 8½ x 11 in.; a third issues his data in a manual 11½ x 18; a fourth issues a good manual covering every vehicle on the market but it is so bulky and heavy it does not lend itself to ready reference for a mechanic after he has been assigned some simple job. A fifth prepares instructions on a wall chart 22 x 38 in.; someone else gets out a wall chart 18 x 24 in., but none has had a real plan of getting the information in front of the mechanic.

It is questionable whether those who prepared charts gave consideration to whether there would be wall area to

accommodate the charts in the shop, and if so, would they be anywhere near the mechanic assigned a job to which the chart pertained? Why spend so much time, thought, and money in the preparation of a great variety of charts and leave to pure chance that there will be wall space for them after they are prepared and distributed? Finally, what thought has been given to the appearance of a repair shop with a hodge-podge of wall charts on every available square foot of wall area?

There is a mass of material – good, bad, and indifferent; much of it merely describing how various units function but lacking in essential instructions on the repair and rebuilding of these units, and how can any fleet operator possibly bring this material together, cull out what is worthwhile and really make it available to his men doing the work?

Maintenance Tool Literature – Here again there is excellent sales literature picturing various repair tools and often featuring their earning capacity; but instructions on exactly how to use the tools or how they should be checked and maintained so as to do satisfactory work, is often all too brief or entirely lacking.

Repair Manuals – The industry has prepared some fair repair manuals. You will find them used in the better dealer organizations where they may be in the hands of each mechanic. However, in fleet operator shops you find single copies in the hands of the shop foreman, but rarely in the hands of individual mechanics. When manuals are held by the shop foreman, mechanics hesitate to ask to see them.

Manuals are usually nicely bound books, printed on highly finished paper, size 8½ x 11 – such nice books that often they fail to reach the man on the job, nor do they lend themselves to "on the job" shop reference and greasy hands. A manual is very much a book for shop office reference, and not especially designed to be put right out on the job and be used as a working medium.

Again, many a manual bears evidence of having come under the jurisdiction of the advertising department. The material has been given a decided sales slant, with leading statements that the "product is one which will rarely require adjustment or repair," and with references to "purchase only genuine parts, use special lubricants, or buy factory-rebuilt units rather than attempt to make repairs." The instructions stop right there, failing to tell and to show specifically how to repair or adjust the unit. Imagine the feelings of some Army mechanic in Australia or Africa who would read in a manual to send some offending part back to the factory and secure a rebuilt unit!

Letters and Bulletins – These go mostly to dealer organizations. They may or may not reach all dealer mechanics. Bulletins again often have a decided sales slant, and distribution of this type of literature to truck operator mechanics is limited.

■ Problem of the Truck Operator

The fleet operator interested in training his mechanics is faced with the problem of "What is he going to teach his mechanics?" and "How is he going to teach them?" We cannot help but feel that lack of suitable material has stifled many attempts to inaugurate a training program. Here and there some operator, on his own initiative, has prepared his own material and he has done an excellent job of training. The work of W. J. Cumming, when he was with the Surface Transportation Co. in New York City, is outstanding in this respect. Mr. Cumming outlined his training program in a paper presented at Penn State College, September, 1941, in connection with the Annual Short Course for Fleet Supervisors and in subsequent papers, since his affiliation with the Office of Defense Transportation.

■ Why Quality of Maintenance?

The automobile industry has progressed to where cars and trucks are built which, when normally maintained and lubricated, will give 50,000 – 70,000 – 100,000 miles of service before major rebuilding. But what mileage is secured from the rebuilt jobs? 5000, 20,000, 40,000, 70,000? Here is where the quality of maintenance comes in and determines whether it is going to be a 5000-mile quality job or a 70,000-mile quality job.

■ Need for Better Maintenance Job

We need to secure quality maintenance as well as quantity. This calls for quality-trained mechanics working in accord with quality maintenance practices.

Right now, with new cars and trucks practically unavailable to civilian life, what better move can the automotive industry make for both present and future than render a vital service by helping their civilian customers really take care of their present equipment? This project has every promise of contributing to the advance of both Army and civilian maintenance methods and of being the foundation for a mechanic training program.

The value of better maintenance instructions within the industry's own organization should not be overlooked. Development of maintenance instructions will surely help the industry develop its own field personnel.

What better way to sell "genuine" quality repair parts than to foster quality maintenance methods and a general belief and pride in excellence of repair and rebuilding? Cheap, inadequate repairs and cheap inferior parts go hand in hand.

■ Standard-Practice Instructions

SAE-ODT Project No. 17 – Standard-Practice Instructions – was set up with the definite thought of accomplishing needed improvement and standardization of maintenance instructions as issued by manufacturers of automobiles, automotive parts, and maintenance tools.

The committee assigned to Project No. 17 was charged with the following scope of activity:

1. Determine the most desirable method, form, and so on, of presenting Standard-Practice Instructions for maintenance.

2. Promote acceptance among parts, vehicle, and shop equipment manufacturers, and induce said manufacturers to produce sample instructions.

3. Promote the use of Standard-Practice Instructions among all maintenance establishments as a medium for training mechanics and for reference by mechanics in actually performing maintenance operations.

Consideration of the merits of various forms of maintenance instructions and of what features would be most useful, led to proposing that instructions should meet the following three essential requirements:

1. *Serve as a training aid,* that is, the instructions wherever practical should be laid out in step-by-step procedure with plenty of clear pictures showing the tools required and exactly how the job should be done, and emphasizing the key points of correct fitting, accurate alignment, permissible wear tolerances, cleanliness and lubrication necessary in the course of assembly, and so forth.

2. *Serve as an "on the job" reference,* that is, the instructions would be made so available that mechanics would readily refer to them very much as a matter of routine in connection with their work. They would be used in much the same manner as Chek-Chart lubrication instructions serve in securing an A-1 job of lubrication.

3. *Be standardized as to size and filing system* to afford the widest latitude in bringing together the instructions of various manufacturers so as to meet individual operating requirements and include only those instructions which would apply to the particular fleet.

The first requirement calls for an organized job study of maintenance work. The material not only should serve a foreman or instructor to teach a mechanic or apprentice the job fundamentals and how to do the job correctly and in accordance with the knowledge and experience of the manufacturer who built the unit or the product, but also such studied instructions should be a sound base on which to develop time studies and flat rates.

The second requirement (to serve as an "on the job" reference) calls for consideration of what is the best medium or form for "on the job" reference. Should the data be put on large wall charts, or slide films, or in leaflets, or big manuals, or small pocket-size booklets? What form lends itself best to putting instructions into the mechanic's hands right on the job? That's the time he's most interested. He wants to do the job right.

It should be of secondary importance as to what is the easiest and cheapest way to print and issue maintenance instructions. Up to the present time it is felt that with few exceptions this latter consideration has taken too much precedence, and that insufficient consideration has been given to the final objective of really reaching the man on the job. The extent of failure to reach him is, in a way, a measure of the waste of the time, money, and effort put into inadequate instructions.

The third requirement calling for standard size and filing system is essential if we are going to secure industry-wide cooperation where each manufacturer makes a maintenance study of his own product and prepares the instruction so that it will tie in with the instructions of other manufacturers. This means, of course, that the manufacturer of the complete vehicle would call on his various parts manufacturers to supply the instructions on their particular assemblies. Thus Tru-Stop brake would prepare instruction on the adjustment and repair of their brake—Spicer would go into detail on the care of Spicer Universals, and so on.

■ Single-Card Instruction Approved

As a starting point the project committee approved a single-sheet maintenance instruction size 14 x 18 in. This size is large enough to lend itself to large, clear illustrations and the use of large type, yet it is not so large as to be clumsy to handle, or refer to, or file, and it cuts well from light card stock.

In proposing single-sheet maintenance instructions there is no new thought. Look at the outstanding job which has been done by Chek-Chart, with their single grease-proof sheets designed to be handled by mechanics or the man who is doing the job. Chek-Chart has made a tremendous contribution to instructing the maintenance world on the correct and thorough lubrication of cars and trucks.

Chek-Chart developed a standard-size instruction card (9 x 12) designed to be put in the hands of the man doing the work. They presented an educational and training medium, and it's one which has been recognized; for the Army has taken it up and Chek-Chart has prepared single-sheet lubrication instructions for all Army ordnance and automotive equipment. With this successful example in front of us, it would seem logical to profit by the experience and develop a slightly larger, single greaseproof sheet system for maintenance instructions.

■ Repetition of Effort

At present, most vehicle manufacturers prepare complete maintenance manuals covering both the units they build and those they buy. This means considerable repetition of effort on the part of vehicle manufacturers in the preparation of instructions pertaining to those units and assemblies which all purchase.

It is the thought of the committee that a far better job will be done if the various unit instructions are developed by the manufacturers of the units. They designed and built the units originally. They have followed their products in service, and they should know more than any one else what clearances, fittings, and adjustments must be observed, what wear tolerances are permissible, and what special tools and shop tricks will facilitate repairs.

Once a parts manufacturer has prepared a complete instruction, then it would be available to every manufacturer using the item. All would use the identical instruction, and none would be obliged to prepare their own version. If a general application of this principle can be put into effect, it should avoid much confusion and duplication of effort, and only leave to the vehicle manufacturer the preparation of instructions such as "engine tune-up," or "vehicle lubrication."

Manufacturers of maintenance tools would issue instructions on the correct use of their products and how to maintain them so that they will do first-class work whenever used.

All manufacturers would use the same size sheet and thus these sheets could be brought together and put into a standard box to be kept in the repair shop tool room where they would be available to mechanics for reference and would be drawn by mechanics as various maintenance jobs are assigned to them. This would more or less simulate production methods where production instruction sheets give each mechanic definite instructions of what he is to do, and what tools will be required to do it properly.

■ Training Application

Such instruction sheets would readily lend themselves to a training program which could be carried on not only by any vocational or trade school, but by any fleet operator in his repair shop.

1. The foreman or working instructor would discuss the job with the mechanic, following the instruction sheet step by step and bringing out the key points.

2. The man would do the job under the instructor's

supervision, following the instruction sheet as he has just been shown how to do.

If there were four or five mechanics to be instructed, the initial instruction might be given in the shop office, followed by an hour of shop instruction and practice with the unit being studied.

3. The instruction sheet would be *placed in the shop toolroom* and would be available to the mechanic any time thereafter should he again be assigned the same job.

Ready Reference-Essential

Emphasis should be placed on the matter of ready reference to specific data. When we think of the assortment of vehicles in almost any fleet purchased over a series of years – think how the original building and assembly of each one of the jobs was spread out over many men, each concentrating on his own part of the job – we begin to realize how impossible it is, how unfair it is, to expect the typical mechanic to keep all of this in his head and at his fingertips. Likewise, it is just as unfair to expect foremen to remember it all. Again it should be realized that the beautiful idea of mechanics taking home maintenance manuals and studying them from cover to cover is contrary to good training technique and will not produce satisfactory results. Mechanics will take the manuals home, and that is where many of them will be found, but it is doubtful if the typical mechanic will sit down at home and intensively study a manual and learn it from cover to cover.

Single-Sheet Instructions versus Bound Loose-Leaf or Bound Instructions – The petroleum industry has had considerable experience in dealing with both single-sheet and bound lubrication instructions at their numerous service stations. For years The Atlantic Refining Co. has been using single-sheet instructions, having found them far superior to any form of bound instructions. The crux of the whole matter is that men will use single instruction sheets but balk at taking a whole volume of bound instructions to the job.

Briefly, it looks to us as though bound books afford a convenient way to present a program of instructions, but single greaseproof sheets afford the best method of having mechanics use the instructions.

Specifications of Cards

It can be appreciated that for the success of this project it is essential that a single standard size of card be used. Only in this way can instructions prepared by various manufacturers be brought together as desired in a common file.

The committee went ahead and requested that the 14 x 18 card be made an SAE Standard for Maintenance Instructions, and submitted the following specifications:

1. Size: 14 x 18 in.

2. Material: a high-grade white tag stock at least equal to "WITAG" made by Linton Brothers, Fitchburg, Mass., which product runs 111 lb per 500 sheets 22½ x 28½ in.

3. Minimum size of type: for use in any text to be not less than 12 point.

4. Greaseproof treatment: instruction cards are to be treated with lacquer or other suitable finish so as to withstand and accommodate themselves to handling by mechanics' greasy hands, and so that grease can be readily wiped off without impairment of the cards.

Indexing and Filing of Instructions

The committee reviewed and approved a plan for indexing and filing Standard-Practice Instructions. This is presented in Fig. 1 (see p. 332), and is outlined with two types of applications in mind:

1. A file suitable for the typical fleet operator.

2. A file suitable for any vehicle manufacturer with branches or large service departments.

In the case of the operator, his fleet is usually composed of a few makes of vehicles, and the instructions he would need would only be those applying to these particular vehicles.

In the case of a large manufacturer of a complete line of light and heavy trucks, he would need a complete file on the various models of vehicle which he manufactures, and therefore he might elect to introduce supplementary "model guide cards" to meet his own particular problem. However, in both cases the primary elements of the proposed indexing and filing system would apply, and these are:

1. *Vehicle Guide and Index Cards* – (Item 1 in Fig. 1.) Vehicle Guide and Index Cards will carry the manufacturer's name on the tab and will carry the following numbered index headings across the top of the card. These headings conform with good practice as found in maintenance manuals.

1. Axle Front
2. Axle Rear
3. Body and Cab
4. Brakes
5. Clutch
6. Cooling System
7. Electrical System
8. Engine
9. Frame Springs
10. Fuel System
11. Lubrication
12. Steering
13. Trans.-Power Take Off
14. Propeller Shafts
15. Wheels, Hubs, Bearings
16. Winch – General

It is our thought that the vehicle manufacturer would print his own guide cards and place the guide in position according to the following allocation of the alphabet: (Item 3 in Fig. 1.)

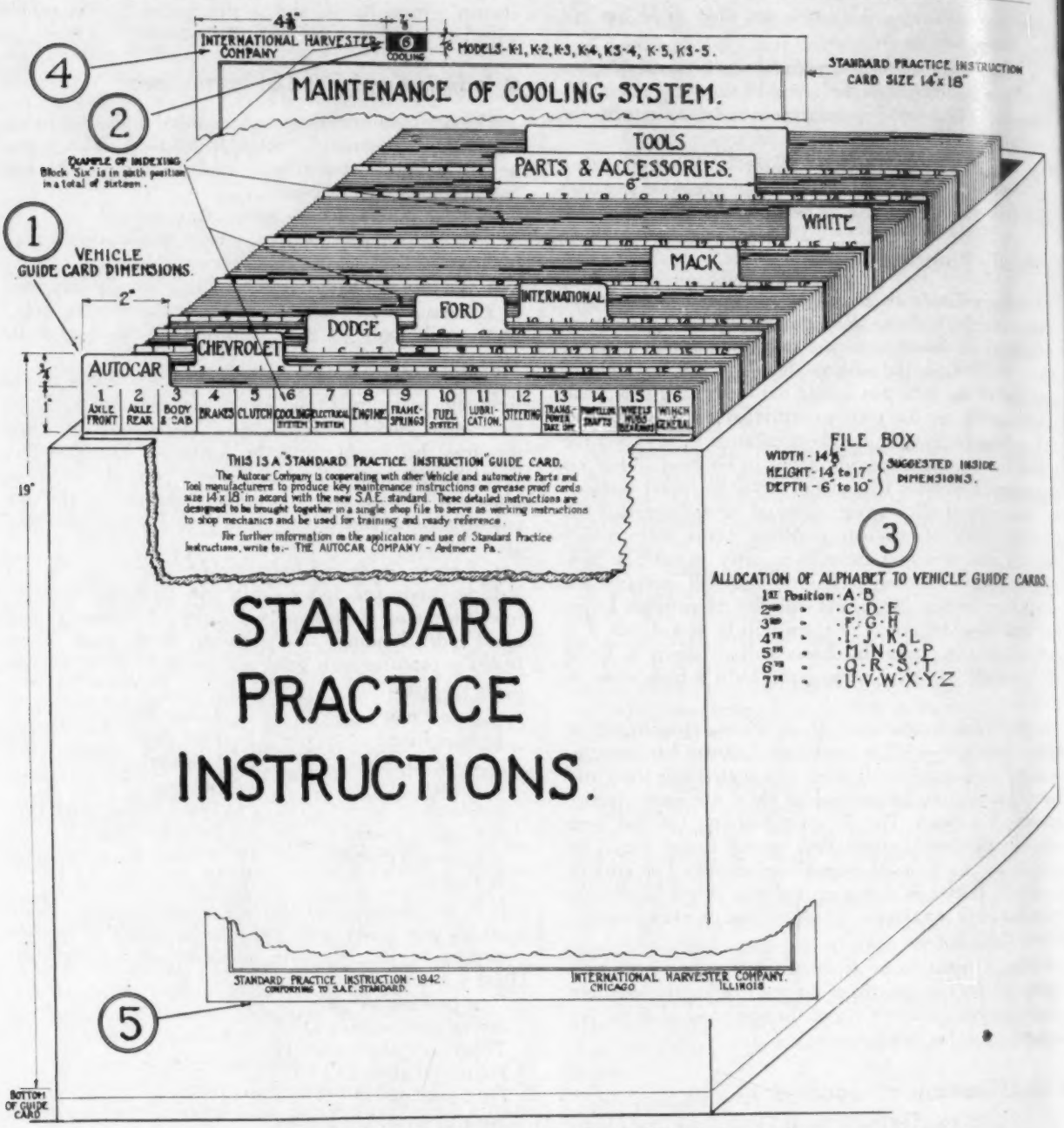
First position: A – B
Second position: C – D – E
Third position: F – G – H
Fourth position: I – J – K – L
Fifth position: M – N – O – P
Sixth position: Q – R – S – T
Seventh position: U – V – W – X – Y – Z

2. *Index Block at Top of Each Standard-Practice Instruction* – (Item 2 in Fig. 1.) Each Standard-Practice Instruction Card will carry an index block at top of the card, located so that it will register with the index item on the guide card to which the instruction applies. Thus a card devoted to maintenance of the cooling system (Item 6 on the index card) would have an index block located in the sixth position and similarly numbered.

3. *Company and Model Identification at Top of Card* – (Item 4 in Fig. 1.) It may be desirable to show company identification at the top of the card as well as models to which the card applies.

4. *Identification at Bottom of Card* – (Item 5 in Fig. 1.) At the bottom of the card it is planned to show:

1. The card is a Standard-Practice Instruction conforming to the SAE standard.


INTERNATIONAL HARVESTER COMPANY
COOLING
MODELS-K-1, K-2, K-3, K-4, K3-4, K-5, K3-5.
STANDARD PRACTICE INSTRUCTION CARD SIZE 14"x 18"
MAINTENANCE OF COOLING SYSTEM.
EXAMPLE OF INDEXING. Block "Six" is in sixth position in a total of sixteen.
TOOLS
PARTS & ACCESSORIES.
WHITE
MACK
INTERNATIONAL
FORD
DODGE
CHEVROLET
AUTOCAR
1 AXLE FRONT
2 AXLE REAR
3 BODY & CAB
4 BRAKES
5 CLUTCH
6 COOLING SYSTEM
7 ELECTRICAL SYSTEM
8 ENGINE
9 FRAME-SPRINGS
10 FUEL SYSTEM
11 LUBRICATION
12 STEERING
13 TRANS-POWER TAKE OFF
14 PROPELLER SHAFTS
15 WHEELS HUBS BEARINGS
16 WINCH GENERAL
1 VEHICLE GUIDE CARD DIMENSIONS.
2"
19"
THIS IS A "STANDARD PRACTICE INSTRUCTION" GUIDE CARD.
The Autocar Company is cooperating with other Vehicle and automotive Parts and Tool manufacturers to produce key maintenance instructions on grease proof cards size 14"x 18" in accord with the new S.A.E. standard. These detailed instructions are designed to be brought together in a single shop file to serve as working instructions to shop mechanics and be used for training and ready reference.
For further information on the application and use of Standard Practice Instructions, write to:- THE AUTOCAR COMPANY - Ardmore Pa.
STANDARD PRACTICE INSTRUCTIONS.
FILE BOX
WIDTH - 14"
HEIGHT - 14" to 17"
DEPTH - 6" to 10"
SUGGESTED INSIDE DIMENSIONS.
3
ALLOCATION OF ALPHABET TO VEHICLE GUIDE CARDS.
1st Position - A-B
2nd " - C-D-E
3rd " - F-G-H
4th " - I-J-K-L
5th " - M-N-O-P
6th " - Q-R-S-T
7th " - U-V-W-X-Y-Z
STANDARD PRACTICE INSTRUCTION - 1942. CONFORMING TO S.A.E. STANDARD.
INTERNATIONAL HARVESTER COMPANY. CHICAGO ILLINOIS
5
BOTTOM OF GUIDE CARD

■ Fig. 1 – Proposed index system for Standard-Practice Instructions

2. Name and address of the manufacturer who prepared the instruction.

3. Copyright notice if desired.

■ Data on Vehicle Guide Card

The vehicle guide card lends itself to tabulating data on various models; likewise it can be used to tell briefly about Standard-Practice Instructions.

The suggested note on the guide card (See Fig. 1) is purely optional, but is proposed with the idea of getting the interested operator to write the manufacturer and ask how he can use the instruction cards to fullest advantage.

This offers the manufacturer the opportunity of encouraging the operator to set up a definite mechanic training program, and to bring to the operator's attention work such as is now being carried on by the SAE on this same topic – training of mechanics.

■ Parts and Accessories Guide Card

The Parts and Accessories Guide Card will carry the same 16 index headings as used on the Vehicle Guide Card.

Standard-Practice Instructions prepared for direct distribution by parts and accessory manufacturers would be filed according to index number back of the main guide card "Parts and Accessories." However, if the instruction applied to some particular vehicle or group of trucks, it could be filed back of the vehicle guide card.

If the instruction is distributed indirectly through some vehicle manufacturer and applies specifically to the latter's product, then it should be filed in its index block position back of the vehicle manufacturer's guide card.

Tools Guide Card

The Tools Guide Card will carry the same 16 index headings as used on the Vehicle Guide Card.

Standard-Practice Instructions on the use, adjustment, and care of maintenance tools will carry an index block number at the top of the instruction which will tie in with the purpose of the tool, and will be filed back of the General Guide Card "Tools."

Thus a valve refacing machine is an engine tool and would be indexed with a No. 8 block. Supplementary identification at the top of the card could include the name of the article and the manufacturer. Thus: Hall Valve Refacing Machine.

SAE Standards

The material was submitted to the General Standards Committee of the Society and this Committee has voted these specifications be accepted so that now we have an approved SAE standard size card and a standard filing system as a sound base on which we can proceed. (See 1943 SAE Handbook pp. 726-727.)

Second Phase of the Project

Establishing standards completes the first and easiest phase of the project. To secure general acceptance of the project is quite a different and far more difficult matter.

Several members of the committee went to work on examples of instruction cards. These are in various stages of completion including cards on the following items:

1. Cooling system – National Carbon Co.
2. Engine check – preliminary to fitting rings and elements of ring fitting – Wilkening Mfg. Co.
3. Diagnosing the air brake system – Bendix-Westinghouse Automotive Air Brake Co.
4. Repair and rebuilding the relay valve – Bendix-Westinghouse Automotive Air Brake Co.
5. Bendix mechanical brakes as applied to a special group of trucks – The Atlantic Refining Co.
6. Removing and replacing truck tires on Firestone type "R" rims – The Atlantic Refining Co.
7. Engine bearing replacements – Federal-Mogul Corp.
8. Engine tune-up – International Harvester Co. (See Fig. 2, p. 334.)

Test Program

The National Carbon card on "Cooling," printed in four colors, was put through an organized test program to find out what operators and others thought of the idea of Standard-Practice Instructions. The results of that survey were spread on study sheets and summarized as follows:

Digest of information given in 220 questionnaires concerning the S-PI project:

85% of all who were questioned are in favor of the project.

80% gave as their reason – card system is simpler and more accessible.

85% are in favor of the project for use on service jobs by present shop mechanics. 10% think it would not be used by mechanics and 5% think manuals are better.

85% think the project would be very helpful in training new mechanics. 10% think manuals are better.

80% of manufacturers called on favored the S-PI project, while 20% thought manuals were more useful.

Many varied comments were made on the questionnaires, among them:

The card is too small.

The card is too large.

The card should be printed on one side only.

The card should be fixed for hanging up.

The card should be kept in a locked file box.

Many organizations have indicated their interest in the project but have not been free to go ahead due to their war work. Several large fleet operators and public utility companies have recognized the possibilities of the project and have taken on themselves to write to many of their vehicle and parts contacts, urging them to give favorable consideration to the plan.

The American Transit Association has shown in its comprehensive report "Wartime Selection and Training, Vol. 1, Transit Maintenance Personnel" its interest in Standard-Practice Instructions and has included in the report an extract from the original presentation.

The American Gas Association, through the chairman of the Motor Vehicle Committee, likewise has shown keen interest in the project and wishes to see it develop.

The Brake Lining Manufacturers' Association requested that a subcommittee be formed, and this group has developed material covering basic instructions on the application of brake lining.

The White Motor Co. has taken up the project wholeheartedly in connection with its new service program.

The Research and Marketing Committee of the National Standard Parts Association has given favorable consideration to the project.

Thus, some progress has been made in the second phase of the project.

From time to time various questions have come up regarding the project, and we would like to discuss these briefly.

1. Many see the advantages of the single greaseproof card of specific instructions but do not see how it will replace the repair manual, and therefore feel the cards become just an added responsibility. This is one way of looking at the matter, but ordinarily when we shift from one medium to another we carry on with the old until the new is ready to function.

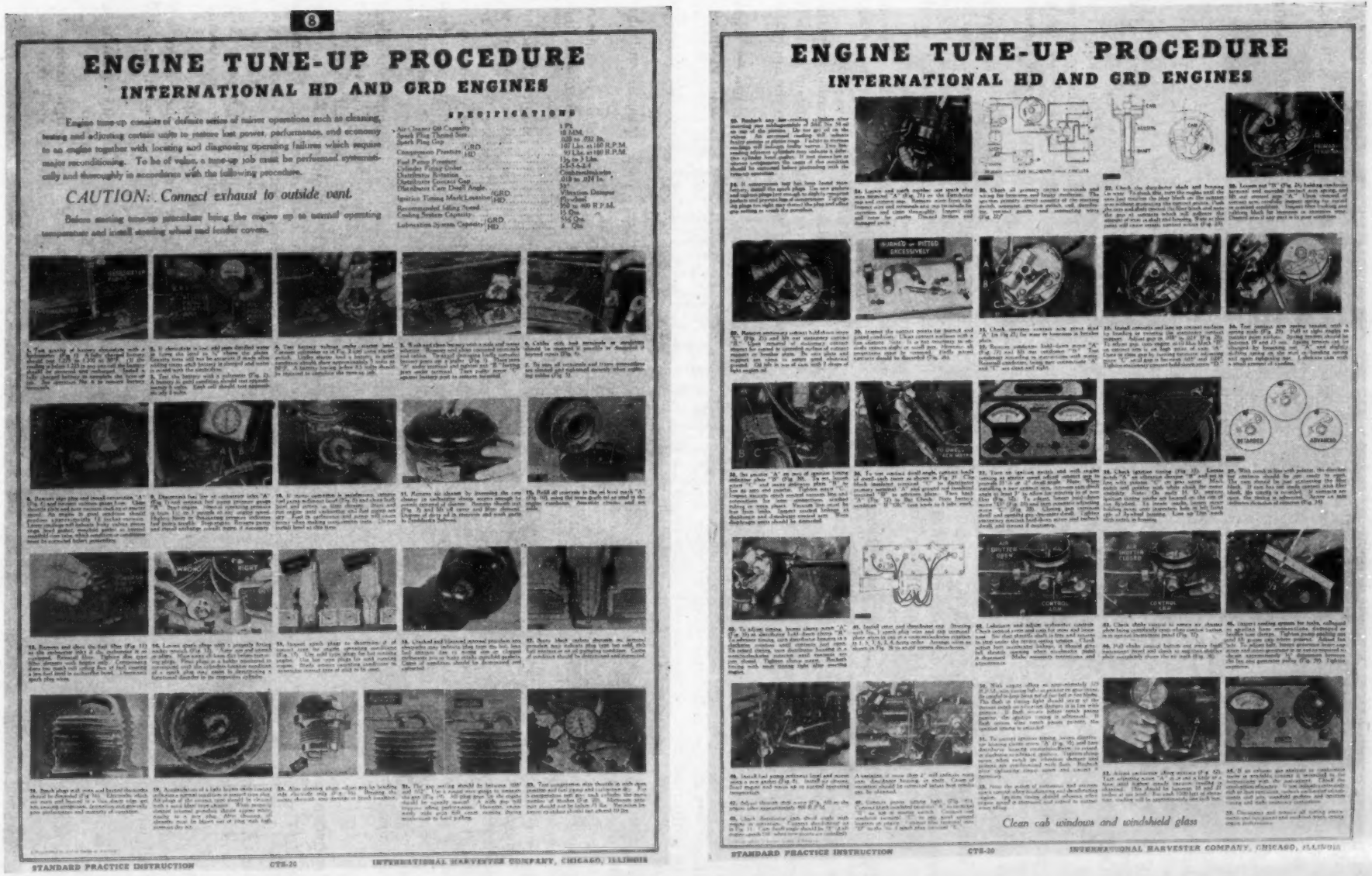
One successful truck manufacturer has found the advantages of the single-sheet instructions and develops his maintenance manuals by binding these loose-leaf sheets. For this manufacturer it should be simple, as S-PI cards are prepared and made available to discontinue his manual sheets which correspond and then insert a notice sheet calling attention to the new S-PI instruction. It is well to take time to do a good job, and all we can hope to do at first is to begin with key items such as brakes, steering, and the principal items of running maintenance.

2. Question is brought up of fear that the project will lead to tremendous and unnecessary duplication, and will fall of its own cumbersome weight.

In considering this it seems to us that the products of parts manufacturers fall in one of two groups:

1. Those products of very individual design having distinct operating and maintenance characteristics, that is:

Bendix-Westinghouse air brakes



8

ENGINE TUNE-UP PROCEDURE

INTERNATIONAL HD AND GRD ENGINES

Engine tune-up consists of definite series of minor operations such as cleaning, testing and adjusting certain units to restore lost power, performance, and economy to an engine together with locating and diagnosing operating failures which require major reconditioning. To be of value, a tune-up job must be performed systematically and thoroughly in accordance with the following procedure.

CAUTION: Connect exhaust to outside vent.

Before starting tune-up procedure bring the engine up to normal operating temperature and install steering wheel and fender covers.

SPECIFICATIONS

Air Cleaner Oil Capacity		1 Pt.
Spark Plug Thread Size		18 MM.
Spark Plug Gap		.025 to .032 In.
Battery Capacity	GRD	107 Lbs. at 160 R.P.M.
	HD	93 Lbs. at 160 R.P.M.
Fuel Pump Pressure		1½ to 3 Lbs.
Cylinder Firing Order		1-3-4-2
Distributor Rotation		Counterclockwise
Distributor Contact Gap		.018 to .024 In.
Distributor Cam Dwell Angle		30°
Ignition Timing Mark Location	GRD	Vibration Damper
	HD	Flywheel
Recommended Idling Speed		350 to 400 R.P.M.
Cooling System Capacity		15 Qts.
Lubrication System Capacity	GRD	5½ Qts.
	HD	8 Qts.

STANDARD PRACTICE INSTRUCTION CTS-20 INTERNATIONAL HARVESTER COMPANY, CHICAGO, ILLINOIS

ENGINE TUNE-UP PROCEDURE

INTERNATIONAL HD AND GRD ENGINES

Clean cab windows and windshield glass

STANDARD PRACTICE INSTRUCTION CTS-20 INTERNATIONAL HARVESTER COMPANY, CHICAGO, ILLINOIS

■ Fig. 2 – Sample instruction cards have been worked out by a number of members of the SAE-ODT Subcommittee on "Standard-Practice Instructions." Illustrated here is the front (left) and rear (right) of a typical card as produced by International Harvester Co.

Wagner Electric air brakes
Bendix vacuum brakes
Steering columns – clutches
A-C fuel pumps.

2. Those products more closely of common design such as the various makes of brake lining, engine bearings, piston rings, valves, ball and roller bearings, and batteries.

As to the first group, their very individuality of design will ensure against duplication. Right now Bendix-Westinghouse have two instructions in hand:

(a) A card devoted to systematic diagnosis of the Bendix-Westinghouse air brake system.

(b) A card devoted to the repair and rebuilding on the relay valve, which is an example of the manner in which we hope the other principal units will be covered.

These two Westinghouse instructions are typical of common thinking on the problem as a whole, that is:

1. *Tell and Show How to Diagnose and Adjust* – Running maintenance includes the work of testing, diagnosing, and adjusting. It does not ordinarily require the use of new parts, with the possible exception of gaskets, cotter pins, and the like.

2. *Tell and Show How to Repair and Rebuild the Various Units* – Normal wear ultimately will require repair, overhaul, and replacement of parts. In doing this work, it is essential that clearances, fittings, and permissible wear tolerances be considered, as well as disassembly, reassembly, adjustment, and testing procedures.

It is planned that these instructions would be made available as wanted through the vehicle manufacturer when he sells the vehicle to some operator, and that they would also be available through the service organization of Bendix-Westinghouse.

Diagnosing vacuum brake difficulties is quite a different job and the maintenance of the units is just as different, so there should be no duplication of effort between air and vacuum brake manufacturers. The same applies to steering columns, clutches, and the like. The individuality of design calls for specific instructions and precludes an instruction common to all.

Regarding the second group (those products much alike), it is going to call for closer cooperation, and it looks as though the various parts associations could take a most helpful part in working toward satisfactory solutions.

Thus, we have high hopes that the Brake Lining Manufacturers' Association will work out a plan of issuing a S-PI card where the data will be devoted to the most approved methods of checking conditions of brake shoes, attaching brake linings, call for resurfacing brake drums in accordance with standard oversizes of lining, and so on.

This would free the manufacturers of brake lining machines, drum lathes, and other special machines to devote their instructions entirely to the correct use of their respective machines, and to the lubrication adjustment and maintenance of these machines so that they will deliver first-class work.

When we consider piston rings we find much the same situation, and reflected in the sample instruction developed by Paul Friend and D. W. Pedrick of the Wilkening Mfg. Co.

A. One side of the instruction card is devoted to diagnosis, and the preliminary and final steps experience has shown desirable and essential toward securing a satisfactory ring job, that is, thoroughly flushing out the cooling system so that cooling will not be impaired by rust drying and hardening on the water-jacketed surfaces of the cylinders when the system is drained to install rings, checking bearings for excessive flow of oil, and so forth.

B. Reverse side of the card devoted to the fundamentals of fitting piston rings.

The instruction might be developed and printed by the ring people getting together, or it might be printed with variations by each ring manufacturer, in quantities to meet individual requirements.

■ Sound Slide Films and S-PI

Though some manufacturers question the place of Standard-Practice Instructions and whether they are needed where manuals and bulletins are issued, still we find these same people using sound slide films in a training effort to supplement their printed literature.

Slide films can perform an excellent and useful service. They have an emotional appeal and can do a fine job of awakening interest, of helping to visualize the possibilities and the many phases of good maintenance. They can stimulate the desire to know more about the job and to do better work; but when this stimulation has been secured, then a medium such as Standard-Practice Instructions will do a better job of carrying on, for slide films have some definite and obvious limitations.

With slide films it is difficult to control the time of showing a film to mechanics covering some job and the time the mechanic has that job to do. Often films are shown to shop mechanics prior to real acquaintance with the parts shown by the pictures. Thus, there is lacking all connection between the pictures and the actual parts.

After being shown, films are usually taken away by the service representative who showed the pictures, and it is almost an impossibility to refer to a specific instruction when really wanted, which is about five seconds after the mechanic is assigned the particular job to which the instruction refers. *Educators say that no one really learns until he is faced with a problem he wants to solve.*

Slide film scripts or the presentation texts which are prepared on some slide films lend themselves less to shop use than do manuals. The illustrations are small and the reading material is primarily prepared to help present the film. They are not engineered for "on the job" use.

Federal-Mogul, in order to secure the advantages of slide films and avoid their disadvantages, have developed a plan of using the slide film to tell a story and are supporting the presentation with a permanent record for training and shop reference in the form of Standard-Practice Instructions. The plan looks most sound.

■ Is This a Proposition to Raise Funds?

Sometimes when discussing this program we have been asked the question "Is this a proposition to raise funds?" Very definitely this is not a fund-raising program, nor is SAE planning to spend the money and prepare the instructions. Nothing could be further from our thoughts.

It is a program which will look to each manufacturer to see what each can do to prepare better, more specific data regarding the maintenance of his product, and to prepare these data on standard-size cards rather than in miscellaneous manuals or film strips or large wall charts. The same funds or a part of them that he would devote to printing manuals would instead go into printing Standard-Practice Instructions. Nor is it our thought that the manufacturers' schemes of distribution of maintenance instruc-

tions be disturbed. Quite a few organizations today place a price on maintenance instruction manuals, and we are inclined to believe that if valuable instructions are issued in card form, they will command a price and fleet operators will be glad to buy them.

■ Maintenance Engineering

An excellent paper presented at the 1942 SAE Annual Meeting brought out the relations between design and research engineering and production engineering.[1]

Should not consideration be given to the matter of maintenance engineering and its relation to the engineering and sales organizations? There would seem to be the very definite engineering job of developing maintenance instructions which will lend to raising the standards of maintenance; of getting into these instructions clearances, fittings, permissible wear tolerances, and items of precise adjustment so essential to good maintenance. We believe it's a job to be done by the engineering department.

It is interesting to note that one of those who has worked with us on this project and contributed a sample instruction, has the interesting title of "production service manager."

Some years ago, Buick cars were equipped with a vacuum-operated clutch. The engineering department issued a chart about 17 x 22, showing exactly how to adjust this clutch for smooth operation, and listing the adjustments step by step. It was distinctly an engineering department instruction to the maintenance organization, and it was put out in form to go into the shop for instant reference. It supplied the data to adjust that tricky clutch correctly and easily.

■ Industrial Applications of Instructions

While we have given primary consideration to better instructions for automotive maintenance, the general plan has definite application in other fields. There exists as wide a gulf between the typical maintenance crew in an industrial establishment and the manufacturers of the plant's machine equipment as exists in the automotive industry between typical maintenance mechanics and the manufacturers of the vehicles.

■ Conclusions

It is appreciated that in making comments such as in this paper, undoubtedly there are exceptions, and we hope those who have done an outstanding job in the preparation of maintenance data will appreciate that this paper is aimed at raising the level, particularly of truck maintenance instructions, to aid in a vital Army and civilian program of maintaining and rebuilding trucks with all the care and accuracy that is put into their original production.

To sum up, we believe it vital that far better maintenance instructions be prepared, which might be called Standard-Practice Instructions.

1. *We need Instructions which will Facilitate Training* – In preparing Standard-Practice Instructions we should consider not merely a reference manual, but consider a coordinated training program of which the instructions are a part.

Instruction should include not only how to do the job but list the tools required, show how they are used if there is any question regarding their use; also should give clearances and fittings, and stress cleanliness and lubrication in connection with reassembly.

2. *We need "On the Job" Maintenance Instructions to Reach Mechanics* – Instructions should be so prepared as to be readily available for reference so that they can be taken right to the job – not a whole book of them, but just the one instruction necessary for the job.

3. *We Need Instructions of Standard Size and Filing System* – Then instructions can be brought together in any desirable combination and kept in the repair shop toolroom where they will be available to mechanics for reference and be drawn by mechanics as various maintenance jobs are assigned to them.

4. *We Need Standard-Practice Instructions for Maintenance Machine Tools* and other special maintenance equipment, such as wheel aligning fixtures; and these instructions should include:

(a) How to use the equipment.

(b) How to maintain the equipment and keep it in accurate adjustment and repair.

5. The very character of the material needed would indicate that possibly it should be prepared by maintenance engineers associated with the engineering departments. The instructions from start to finish should be devoted to accurate rebuilding, specific adjustments, essential tools, and their application.

The parts of an assembly being repaired should be given consideration from the viewpoint of permissible wear, beyond which the parts should be rebuilt, refinished, or replaced.

6. A high degree of cooperative effort will be necessary so that there will be agreement by manufacturers such as Timken, Bendix, Wagner-Electric and Westinghouse, to prepare Standard-Practice Instructions on their products and make these instructions available to all manufacturers using their products; that A-C, Delco, and Auto-Lite would prepare the instructions on their products; that Continental, Waukesha, or Cummins would do likewise on their products. Then the manufacturers of complete vehicles would not assume the job of trying to prepare or rewrite these Standard-Practice Instructions, but would bring these Standard-Practice Instructions together and make them available to purchasers of the complete vehicle.

7. Many have indicated that the war's heavy demands on materials and on men qualified to prepare instructions have held up going ahead with the project.

However, manufacturers of complete trucks and buses can indicate to the manufacturers of parts, who are their sources of supply, that they are definitely interested in the project and would like to have the parts manufacturers look into the possibilities of preparing Standard-Practice Instructions, each covering his own particular product.

Parts manufacturers of brakes, clutches, transmissions, axles, steering columns, fuel pumps, and so on, can cooperate by preparing studies of what can be done and laying plans now. It is a big project and it cannot all be done in a moment, but we are sure much can be done if the automotive industry will take hold.

[1] See SAE Transactions, Vol. 50, October, 1942, pp. 439-443, 464: "Automobile Engineering Organization and Procedure," by C. R. Paton, R. F. Kohr, and M. A. Forester.

QUALITY CONTROL of Engineering Materials during Manufacturing and Processing

by R. H. McCARROLL and J. L. McCLOUD

Ford Motor Co.

WE have realized, as have many others, that the conditions prevailing in industries such as ours have made necessary better control procedure and better means for enforcement of this procedure.

It is our purpose, first, to outline the methods of quality control of engineering materials during manufacture and processing used by the Ford Motor Co., and then to illustrate how this procedure is applied to different types of production and fabrication.

We are well aware that many organizations have a definite quality control procedure, some placing most of the responsibility with various sections of their engineering department, others placing most of the responsibility in the hands of the production departments, and still others relying mainly on a separate inspection department. Various procedures have been worked out in different companies on the above basis and we do not doubt that the results have been quite satisfactory in most cases. However, we hope that by discussing the Ford procedure and, if you wish, comparing it with methods used in other places, we may all benefit in a manner that will help in the war effort.

Our outline for the subject matter to be covered will be as follows:

[This paper was presented at the SAE War Engineering-Annual Meeting, Detroit, Mich., Jan. 12, 1944.]

A. Material specifications – previous limitations and new relations to process specifications.

B. The origin and character of process specifications.

C. Organization of personnel for process specification enforcement.

D. Methods of test used to determine the effectiveness of the procedure and enforcement.

We will endeavor to cover this subject matter not in a highly technical manner, but in a manner that will be broad and general enough to have some interest for all types of engineers. By all types, we mean such classifications as executive engineers, designing engineers, production engineers, and testing engineers.

So that we may endeavor to fulfill these qualifying requirements, we have selected different types of production, fabrication, and treatment units for our illustrations.

1. As an example of raw material production we have selected a plant producing steel by the open-hearth process.

2. To illustrate the control of a type of plant fabricating raw material we have selected a steel foundry.

3. To illustrate process procedure on methods of treatment we have selected surface treatments, such as plating and painting.

This will be followed by a section devoted to a description of methods of tests on parts, subassemblies, assemblies,

QUALITY control of engineering materials has become increasingly important in recent years.

The methods of quality control used by one large automotive manufacturer, as explained in this paper, include a wider and better method of using material specifications in conjunction with process control.

The authors also describe the origin and form of their process control specification, and their method of organization for process control enforcement, using various types of industrial applications to illustrate how this control could be specified and used.

Finally, they describe methods of test on finished parts and assemblies of finished articles to determine the effectiveness of their control procedure and the quality of material, parts, or articles.

■ ■ ■

THE AUTHORS: R. H. McCARROLL (M '42) joined Ford Motor Co. a year after graduating from the University of Michigan in 1914, and has been there ever since. In charge of all chemical and metallurgical engineering of the company, Mr. McCarroll has developed many new alloys and methods for casting such parts as crankshafts, camshafts and pistons. He has played a major role in war production planning, and has served on the Technical Advisory Committee of the Automotive Division of OPM and on the Senior Technical Advisory Committee of the OPM Steel Division. J. L. McCLOUD has, like his co-author, been in the chemical engineering department of Ford Motor Co. since 1915. His only other connection, after receiving his chemical engineering degree from the University of Michigan in 1913, was two years with U. S. Rubber Co. Much of his career has been devoted to the study of paint and other materials.

FORD MOTOR COMPANY

OPEN HEARTH

PROCESS SPECIFICATIONS

APPROVED June 30, 1941

BUILDING SUPERINTENDENT

PROCESS CONTROL DIV.

CHEMICAL ENG. DEPT.

■ Fig. 1 (left) – Title page of open-hearth process specifications

OPEN HEARTH

INDEX

SECTION I	- SYNOPSIS OF OPERATIONS	PAGE 1
SECTION II	- CONSTRUCTION	4
SECTION III	- AUXILIARY EQUIPMENT	8
SECTION IV	- RAW MATERIALS	11
SECTION V	- OPEN HEARTH OPERATION	15
SECTION VI	- SOAKING PITS	23
SECTION VII	- ROLLING MILL	28
SECTION VIII	- PRODUCT SPECIFICATIONS	33

■ Fig. 2 (right) – Index page of open-hearth process specifications

and the finished article. These tests being made in order to determine the effectiveness of the quality control specifications and their enforcement.

An article appeared in Fortune entitled, "Quality Control."[1]

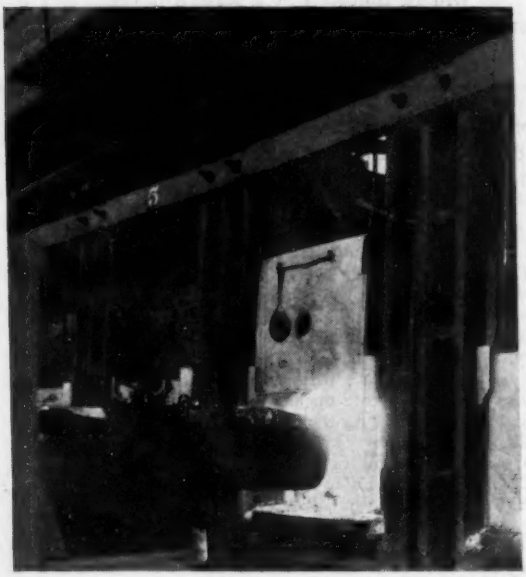
More recently this statistical analysis of quality was the subject of a course of intensive lectures sponsored by the University of Michigan at the Rackham Building in Detroit.

For these their subject was confined to the statistical analysis of quality by some measurements of value and methods of recording and interpreting these values. By this means, deviations out of control may be detected and the point of trouble sometimes located. However, if trouble occurs it was merely said that this matter was up to the engineers.

While we, of course, use this system of statistical analysis, we believe it is more important to keep out of trouble or to know how to correct troubles when they occur.

Our process control divisions do not concern themselves with physical dimensions, this being in the hands of our inspection department.

We realize that the better use of statistical data will improve our control in the phase of quality covered in this paper. However, please keep in mind that here we are concerned only with the quality control of manufacturing materials, as to their raw material composition, the details of their production, fabrication, and treatment.



■ Fig. 3 – Charging operations of the open-hearth floor

A. MATERIAL SPECIFICATIONS

Before the advent of material specifications, manufacturers were dependent on inadequate qualitative and quantitative tests. Small-scale tests being set up to try the comparative results with different lots or samples of materials furnished.

Fracture test on iron and steel, as well as other "rule of the thumb" procedures, were in order.

It was not until 1905 that the SAE issued their first real specifications on the composition of steels.

The ASTM started issuing material specifications about 1901 and as time progressed, more and more emphasis was placed on physical and behavior characteristics.

Later engineering prints have borne, not only the material specifications, but in the case of a material such as steel, the heat-treatment and the desired hardness. More recently these prints have come to include physical properties and behavior requirements, sometimes even leaving out definite chemical requirements.

It was realized that a better control of quality could be obtained by prevention.

The inadequate functioning of "trouble shooters," the many unauthorized changes, and the need for better process specification influenced us in setting up this control system under the chemical and metallurgical engineering department.

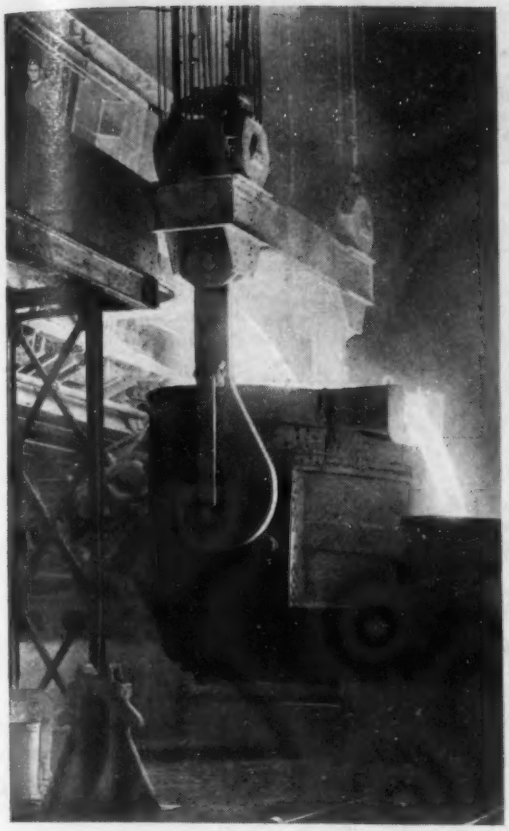
B. CHARACTER OF PROCESS SPECIFICATIONS

In the previous section we have given some of the reasons for the origin of process control specifications.

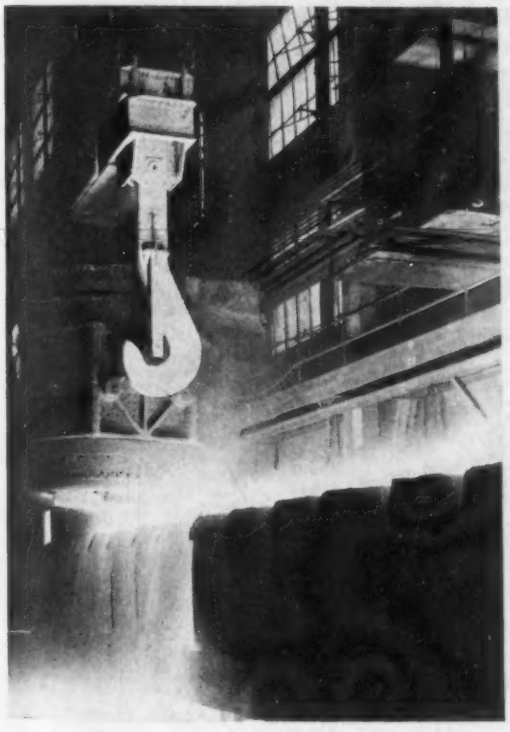
Their preparation and coordination in their present form in the Ford Motor Co. began about 2 years ago. This form will be covered in detail when we come to the illustrations mentioned.

For the present, let us only say that they include all material specifications, tying them into a completely detailed

[1] See *Fortune*, Vol. 28, October, 1943, pp. 126-127: "Quality Control – Its Demands Grow Tighter as War Get Together."



■ Fig. 4 – Tapping steel from the open hearth into a ladle



■ Fig. 5 – Teeming from the ladle into ingot molds

description of the proper operating and testing procedure, that they are prepared in cooperation with the production department and other interested engineers and that they may be changed only by agreement for improvement in which case the particular section is revised and approved in much the same manner that a design change is handled.

C. ORGANIZATION OF PERSONNEL

A large personnel is not needed but their selection as to fitness for the job is most important.

Also, much consideration must be given to the degree of cooperation that will prevail between others concerned, such as the general management, designing engineers, production division, inspection department, and the usual chemical and physical testing units.

To illustrate how the personnel is divided, there are, at present, about 945 people in our chemical and metallurgical engineering department, dispersed in 38 different locations. About 106 of these are entirely devoted to the process control work and spend most of their time out on the production lines, but get the reports and the cooperation of the various control laboratories and testing stations.

The men selected for process quality control positions have always had considerable experience in that particular production division but they may have widely differing qualifications and may be taken from either the chemical and metallurgical engineering department or some other department, such as the production department, the inspection department, the heat-treating department, or some testing laboratory.

It may be surprising but often, by proper selection and organization, the total number of men may be decreased rather than increased when the new control organization is put into operation.

These men see that all necessary materials, operations, and testing procedures are covered by proper specifications and that they are properly followed. They help to locate a condition not under control and aid in correcting the trouble at its source.

Of great importance is the fact that their reports and influence go beyond the production division supervision and building superintendents.

■ Open Hearth

Following the outline, we will give examples of these process control specifications. The first example is the open-hearth steel plant. The title page, as illustrated in Fig. 1, shows it to be written by the process control division of the chemical engineering department in cooperation with the production supervisors. It has the approval of both of these groups and final acceptance by the building supervisors and the head of the chemical engineering department.

In the open-hearth department, 12 of the 62 employees in the chemical engineering division are on this quality control work.

The process specification index (Fig. 2) shows its division into synopsis of operations, construction, auxiliary equipment, raw materials, operation, and so on. These are the essential points covered by this typical specification.

Figs. 3 to 5 illustrate the work in this department.

Fig. 3 shows the charging operations on the open-hearth floor.

Fig. 4 shows the tapping of the steel from the open hearth into a ladle, and Fig. 5 shows the teeming from the ladle into ingot molds.

To use specific examples of the quality control specifications, we will turn to page 20 in this book (Fig. 6). The

dry foundry sand to a depth of 2". This is to insure that the entire pipe will occur in the hot top section. When completely poured, the heat shall be held in the pouring pit before moving until sufficiently solidified. See figure 5, page 30 for recommended times for holding various sized ingots.

XIII. Sampling:

While teeming, it is necessary to take samples for chemical analysis and grain size. Chemical samples shall be taken from the first, center and last part of the heat. They are poured into a small test mold. A small amount of aluminum should be used in the test mold to provide a solid sample free from blow holes. The samples should be identified and delivered to the laboratory immediately after pouring.

Grain size samples shall be taken from the first and last of the heat. These samples should be removed from the mold and annealed in slag as soon as possible. No aluminum is to be used in grain size samples.

EQUIPMENT

1. Ladles:

The ladle into which the steel is poured is made of boiler plate and shall be lined with two courses of fire clay brick each 4½" thick. Externally, the ladle is bolted about the upper half with a heavy plate to which the rest for supporting it and the trunnions for lifting it are attached. One hundred and two hundred ton ladles, depending on the size of the heat, are used at the Ford Motor Company. The opening at the bottom of the ladle is provided with a fire clay nozzle 1 3/4" in diameter, which is opened or closed at will with a stopper. The stopper is made of clay bonded graphite and is mounted on a rod, protected by fire clay sleeve brick, that reaches to the top of the ladle. It is

■ Fig. 6 (left) – Page of specifications describing the time at which samples are to be taken for chemical analysis and grain size

TRACK TIME

The length of time elapsing between pouring and charging into the soaking pits is known as the track time. This time should not be any longer than the time necessary for solidification. Steel held in the molds after solidification tends to shorten mold life, sets up strains in the ingot and lengthens the time the ingots must remain in the soaking pits. The following chart indicates the recommended track time for various sized ingots.

DESIRED TIME INTERVALS IN HANDLING STEEL BETWEEN THE OPEN HEARTH AND SOAKING PITS

TYPE STEEL	MOLD SIZE	TIME HELD IN MOLDS AFTER POURING	TIME HELD IN MOLDS BEFORE CHARGING IN PITS	
			MINIMUM	MAXIMUM
Bar	20"	20 Min.	1 Hr. 30 Min.	5 Hrs. 00 Min.
	27"	20 Min.	1 Hr. 30 Min.	5 Hrs. 00 Min.
Sheet	(30"	1 Hr. 30 Min.	1 Hr. 45 Min.	5 Hrs. 30 Min.
	(32"	1 Hr. 30 Min.	1 Hr. 45 Min.	5 Hrs. 30 Min.
	(36"	1 Hr. 30 Min.	1 Hr. 45 Min.	5 Hrs. 30 Min.
A.Plate	(36"	2 Hrs. 00 Min.	2 Hrs. 15 Min.	5 Hrs. 30 Min.
	(52"	2 Hrs. 00 Min.	2 Hrs. 15 Min.	5 Hrs. 30 Min.
	(66"	2 Hrs. 00 Min.	2 Hrs. 15 Min.	5 Hrs. 30 Min.

If, for any reason, the above sequence cannot be carried out, tender steels must be given preference. Heats that are in the yard longer than 8 hours require special heating and soaking.

Figure 6.

At times when delays are inevitable, the track time is going to be correspondingly increased. To combat this difficulty, it has been found advisable to schedule as many types of steel as possible, which are least affected by a long track time, during these periods. In this manner any so-called tender steels may be given preference. (Types of steel with specifications of any element higher than the following shall be considered

■ Fig. 7 (right) – Page of specifications covering "track time"

(To properly determine the FeO content according to the method described below, it is necessary that the sample to be examined be taken at such time when the slag is free from fluorspar or silicon).

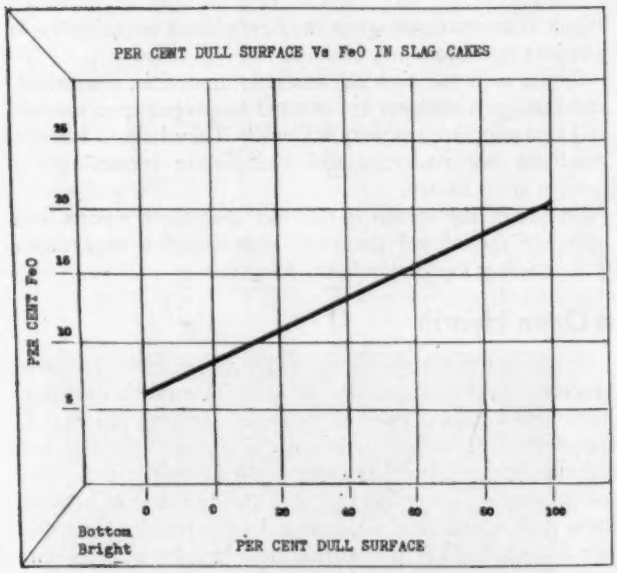


Figure 8

■ Fig. 8 – Page of specifications specifying the relation between a fracture of the slag sample to the ferrous oxide content

specification describes the time at which samples are to be taken from chemical analysis as well as in sampling for grain size.

Another important item covered in the operation of the open hearth is the "track time." On page 22 (Fig. 7) is the specification of the very important factor of time between teeming of the ingots and charging into soaking pits.

Slag formation is an extremely important part of successful operation of open-hearth furnaces, and on page 16 (Fig. 8) is specified the relation between a fracture of the slag sample to the ferrous oxide content. These are taken during the melting operations to guide the operators and show the process control men the progress of the heat.

As the steel is tapped into a ladle, additions of aluminum are made finally to deoxidize the metal. The exact quantity of aluminum is given in the chart shown in Fig. 9, according to the various types of steel regularly manufactured in the open hearth.

■ Steel Foundry

We again refer to our outline, and come to the second illustration of a process control specification covering a method of fabrication, such as casting.

Our illustration is the quality control in a steel foundry. This differs in character from the open-hearth steel control because the end products have the form and shape of finished parts for use in vehicles. The raw materials being here directly converted to usable parts.

The index of this process specification (Fig. 10) shows its division into sections and the details covered.

The synopsis of procedures for use in the Ford steel

ALUMINUM ADDITIONS

TYPE	LBS. PER TON IN LADLE	TYPE	LBS. PER TON IN LADLE	TYPE	LBS. PER TON IN LADLE	TYPE	LBS. PER TON IN LADLE
1010	2 #	2330	1 1/2 #	4320	3 #	A X	3 #
x 1015	1 1/2 #	2340	1 1/2 #	X 4340	1 1/2 #	A A	1 1/2 #
1020	1/2 #					A - Sol.	1 1/2 #
x 1020	1 1/2 #	3115	2 #	4615	2 #	A A	1 1/2 #
1025	1 #	3120	1 1/2 #	4620	2 #	A A.B.	1 1/2 #
1027	1 1/2 #	3135	1 1/2 #	4640	3 #	A A.L.	1 1/2 #
x 1030	1 #	3140	1 1/2 #			A A	1 #
1040	1 #	x 3140	1 1/2 #	4815	2 #	A A.L.	1 #
1045	1 #	3145	1 1/2 #	4820	3 #	A A	1 #
1050	1 #					B	1 1/2 #
1055	1 1/2 #	2240	1 1/2 #	5115	2 #	B - Sol.	1 1/2 #
1060	1 #	2250	1 1/2 #	5140	1 1/2 #	B B	1 1/2 #
1065	1 1/2 #			5150	1 1/2 #	B B - Sol.	1 1/2 #
1070	1 #	3312	2 #			B B	1 #
1075	1 #			6150	1 1/2 #	B B - Sol.	1 #
1080	1 #	4023	2 #			A B	1 #
1085	1 #	4027	2 #	8620	2 #	F	1 #
1090	1 #	4032	1 3/4 #	8630	1 1/2 #	B F	1 1/2 #
1095	1 #	4032	2 1/2 #			B F (1-4)	1/2 #
		4035	1 3/4 #	8725	1 1/2 #	B F.B.	1 1/2 #
1112	2 #	4037	1 3/4 #	8740	1 1/2 #	B F	1 #
x 1118	2 #	4040	1 3/4 #	8745	1 1/2 #	A F	1 #
1118	2 #	4042	1 3/4 #	8750	1 1/2 #	G.S.	2 #
		4043	1 3/4 #			G	2 #
x 1314	2 #	4047	1 3/4 #	9260	1 1/2 #	SCHE - Bar	1 #
x 1315	2 #	4050	1 3/4 #			F	1 #
x 1335	1 1/2 #	4063	1 3/4 #	9262	1 1/2 #	E111 - L	1 #
x 1340	1 1/2 #	4063	1 3/4 #			G	1 #
		4067	1 3/4 #			S.S.	1 #
1335	1 1/2 #					A.PLATE	1 1/2 #
1340	1 1/2 #	4130	1 3/4 #			Shot-Steel	1 1/2 #
1345	1 1/2 #	x 4130	1 1/2 #			Small Heat	Large Heat
1350	1 1/2 #	4135	1 1/2 #	04/07 Sheet		(FeO-12)4.2	(FeO-12)7
		4140	1 1/2 #	08/08 Sheet		(FeO-12)4.2	(FeO-12)7
2330	2 #	4150	1 1/2 #	11/14 Sheet		(FeO-14) 3	(FeO-14)5

■ Fig. 9 – Page of specifications giving the quantity of aluminum to be added during tapping

STEEL FOUNDRY

INDEX

SECTION I	- SYNOPSIS OF OPERATIONS	PAGE 1
SECTION II	- MELTING PROCESSES	5
SECTION III	- MOLDING & CLEANING	11
SECTION IV	- HEAT TREATMENT	16
SECTION V	- MATERIAL SPECIFICATIONS	19
SECTION VI	- WELDING	21
SECTION VII	- SPECIAL PROCESSES	22
SECTION VIII	- RAW MATERIAL SPECIFICATIONS	45

■ Fig. 10 – Index page of steel foundry specifications

SYNOPSIS OF PROCEDURES IN THE FORD MOTOR COMPANY STEEL FOUNDRY

INTRODUCTION

The Ford Steel Foundry was designed to handle castings ranging in weight from a few pounds up to as much as twelve tons. It has a rated capacity of 10,000 tons of finished castings per month and is one of the largest steel foundries of its type.

The melting processes used in the foundry differ with the type of metal made. All low or medium carbon steels for commercial steel castings are manufactured by the Triplex Process. Malleable iron castings are manufactured by the Duplex Process. Armor and similar castings requiring specific properties are cold melted.

Because of the variety in size of castings, it was necessary to divide the molding department into two sections. One part was mechanized to as great an extent as possible to permit the handling of small and medium size castings on large volume production. The other section was designed to permit floor molding of the large castings.

Although the foundry was primarily intended for steel castings, consideration was given in the selection of equipment for the manufacture of malleable and similar types of materials.

METALLURGICAL UNITS

The melting department in the foundry consists of four 96" cupolas, four six-ton side blown Bessemer converters and eight twelve-ton and two four-ton electric furnaces.

The cupolas are of the Whiting hot blast type. In front of each cupola is located a six-ton desulphurizing ladle.

The Bessemer converters are located directly opposite the cupolas with overhead cranes spanning the intervening space in order to facilitate transfer of the metal. The converters are of Tropenas type, that is, side blown and have a capacity of approximately six tons per blow. The blast is supplied through blowers delivering air at the pressure of five pounds.

The electric furnaces are located along the same way as the converters and cupolas so that they can be serviced by the same cranes. The furnaces are all fitted with swinging roofs in order to make them as flexible as possible as these units are used for both cold melting and for triplexing.

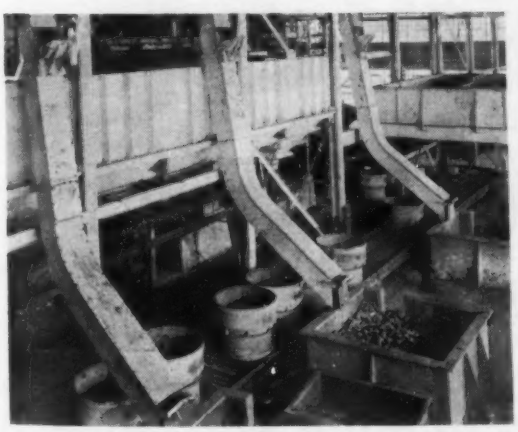
MELTING PROCESSES

Triplexing

The triplex system is so named because three metallurgical units are used in the process of manufacture. The first is the cupola, the second is the Bessemer converter and the third is the electric furnace.

The raw materials required for the making of the steel are kept in large concrete storage bins. An overhead crane fitted with a magnet,

■ Fig. 11 – Page of specifications giving synopsis of procedures for use in the steel foundries



■ Fig. 12 – Stock handling – cupola charging bucket conveyor line

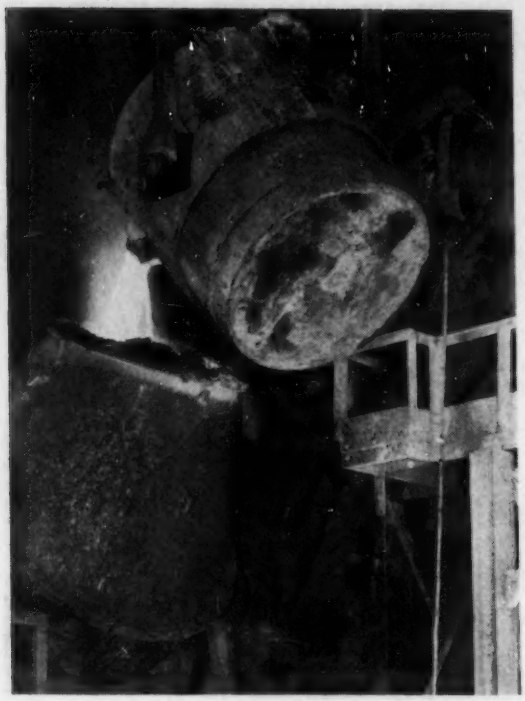
foundries (Fig. 11) shows the main metallurgical units and is illustrated by many pictures which serve to focus the attention of management with respect to the type of operations carried on in the plant as well as to guide the operational staff.

■ Fig. 13 – Cupola charging – bucket resting on "wishbone"

■ Fig. 14 – Desulfurizing ladle in front of cupola

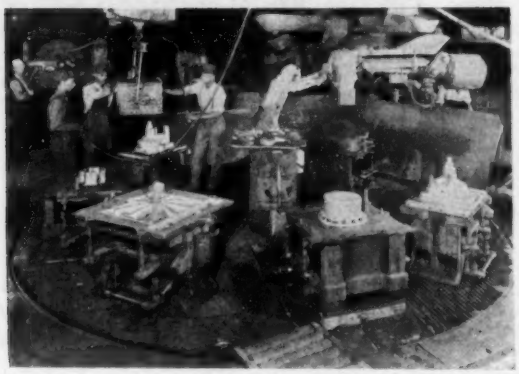
These include stock handling (Fig. 12), cupola charging (Fig. 13), desulfurizing (Fig. 14), charging the bessemer converter with desulfurized iron (Fig. 15), battery of electric holding furnaces (Fig. 16), molding on a turntable (Fig. 17), pouring of crankshafts in dry sand molds (Fig. 18), and the heat-treating furnaces in operation (Fig. 19).

The work of inspection (other than dimensional measurements) and the testing of finished parts, as well as of

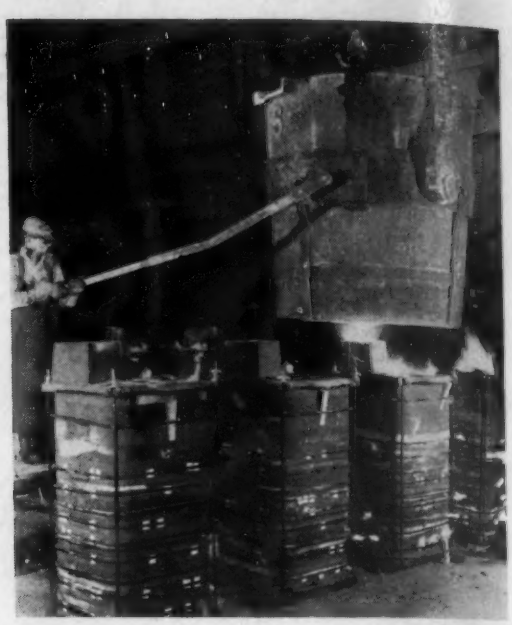


■ Fig. 15 – Charging bessemer converter with desulfurized iron

■ Fig. 16 – Battery of electric furnaces



■ Fig. 17 – Molding using turntable and sand slinger



■ Fig. 18 – Pouring crankshafts using bottom pouring ladles

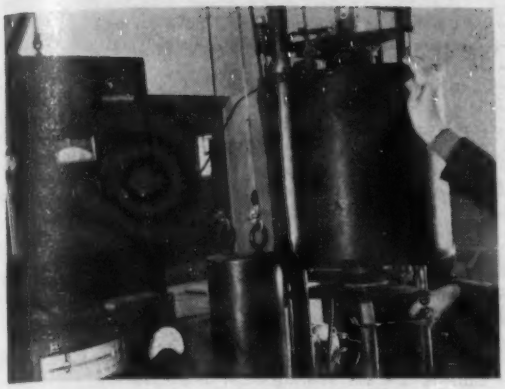
■ Fig. 19 – Discharging heat-treat furnace

■ Fig. 20 – Inspecting in the chemical and spectrographic laboratory

raw materials, is under the chemical engineering department.

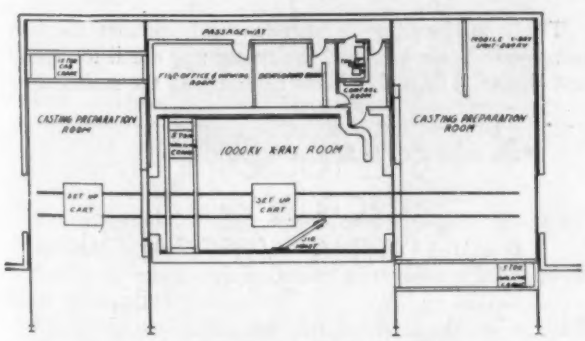
We have a chemical and spectographic laboratory (Fig. 20), a sand testing laboratory (Fig. 21), an X-ray laboratory (Fig. 22), which is equipped with a million-volt X-ray machine, and has adjacent developing and film reading rooms, as well as preparational facilities (Fig. 23) and a physical testing laboratory (Fig. 24).

The process specification gives detailed information regarding the operations as it did in the case of the open hearth. The melting processes are divided as indicated in Section 2 (Fig. 25), and following one of these details, we



■ Fig. 21 – Sand testing laboratory

■ Fig. 22 – X-ray laboratory

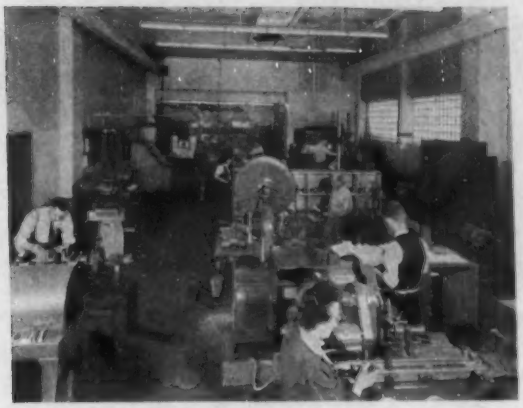


■ Fig. 23 – Preparational facilities

will turn to page 8 (Fig. 26), which gives the description of the slag under the triplexing procedure as to its composition and handling.

Passing over to page 28 in Section 7 (Fig. 27) we have a graph showing the relation between physical properties of test bars cast as such and the properties of cast armor qualification plates.

The process specification for a tank engine crankshaft



■ Fig. 24 – Physical testing laboratory

STEEL FOUNDRY SECTION II MELTING PROCESSES

	PAGE
TRIPLEXING	5
Preparation of Cupola Before Charging	5
Cupola Charges	5
Desulphurizing	6
Transfer	6
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Pouring	8
ACID ELECTRIC STEEL PRACTICE (COLD MELTING)	9
Raw Materials	9
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Meltdown, Oxidizing and Refining	9
Deoxidation Practice	10
Ladle Practice	11
Theoretical Considerations	11
BASIC ELECTRIC STEEL PRACTICE (COLD MELTING)	12
Raw Materials	12
Condition of Furnace	12
Meltdown	12
Refining	12
Deoxidation of the Steel	13

■ Fig. 25 – Index page of melting processes specifications

after each transfer, as well as once in between transfers, resulting in approximately four samples per hour.

Necessary corrections shall be made immediately upon return of the analysis from the laboratory.

Slag

The slag shall at all times be watched carefully, and the following procedure used:

After the second blow is transferred, all slag shall be pulled and a new slag added. This slag is made up of 350 pounds of silica sand, 30 pounds of limestone and 20 pounds of powdered silicon.

The third blow is then transferred and the pouring begins as described above.

The slag shall again be pulled after the sixth blow has been transferred and a new slag of the same composition as above made up.

From this point on, the slag shall be pulled after each eighth blow has been transferred and new slag of same composition made up.

The color of the satisfactory slag is apple green.

The slag shall be checked before each transfer of a blow and if at any time, due to contamination, the slag should become oxidized, i. e., turn brown, shall immediately be pulled and a new slag built up.

Slag analysis will be as follows:

FeO — 8 - 12%
MnO — 24 - 29%
SiO_2 — 58 - 62%

Pouring

The furnace spout must be at a temperature of 1950°F before pouring. The spout shall be shaped such that the stream of metal is well rounded, avoiding any splashing.

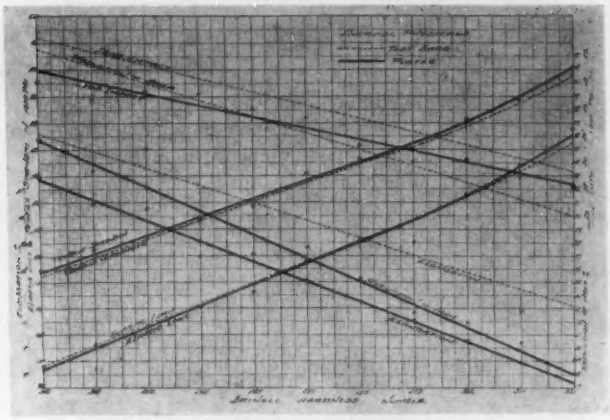
All ladles shall be at the minimum temperature of 1000°F before metal is poured into them and no wet clay or similar patching material shall be allowed while ladle is in use. In addition, no ladle shall be used if a heel or skull has been formed in the ladle due to use of cold metal.

One and one-half pounds Aluminum per ton of metal shall be added in the form of shot aluminum which shall be measured by using a standard size cup and shall be added to the stream when tapping.

All ladles shall be thoroughly slagged before pouring and in addition shall be skimmed during pouring to prevent any slag from entering the mold.

Pouring temperature for jobs weighing less than 100 lbs. shall be 2900 to 2950°F. Larger castings shall be poured at proportionally lower temperatures.

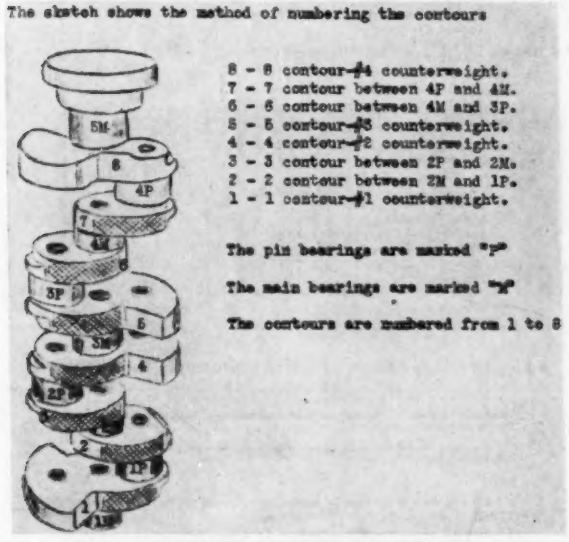
■ Fig. 26 – Page of specification giving description of the slag under the triplexing procedure



■ Fig. 27 – Page of specification with graph showing the relation between physical properties of test bars cast as such and the properties of cast armor qualification plates

THE FORD 8 CYLINDER TANK ENGINE CRANKSHAFT

This sketch shows the positions of the in-gates from the central feed to the crankshaft casting proper. The checkered portions on the sketch show the positions of the in-gates.


The sketch shows the method of numbering the contours
8 - 8 contour—#4 counterweight.
7 - 7 contour between 4P and 4M.
6 - 6 contour between 4M and 3P.
5 - 5 contour—#3 counterweight.
4 - 4 contour—#2 counterweight.
3 - 3 contour between 2P and 2M.
2 - 2 contour between 2M and 1P.
1 - 1 contour—#1 counterweight.
The pin bearings are marked "P"
The main bearings are marked "M"
The contours are numbered from 1 to 8

■ Fig. 28 – Page of specification showing specifications for a tank engine crankshaft

(Fig. 28) shows the shaft as a whole, and Fig. 29 shows the build-up of the core assemblies and the gates and runners for feeding this casting.

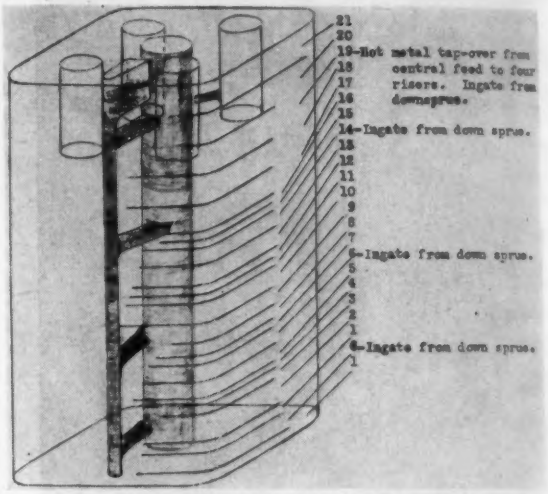
The specification for the steel foundry concludes with Section 8 (Fig. 30), showing the inclusion of raw material specification in the process specification for this plant.

■ Plating and Painting

We come to the third illustrations of process specifications. These are the methods of treatment in plating and painting. We will use an engine bearing for the plating example.

The first section of this specification is again a synopsis of the operations. The reasons for the base metal as well as the plated metal are described. Special equipment required is shown and sequence of plating is given.

#1 – Brick put in center – feed hole. Base for setup.
#6 – Core placed in this position to catch slag.
#1 – Core forms #1 main bearing.
#2 – Core forms #1 counterweight.
#3 – Core is top of #1 cntrwt. to center of #1 pin bearing.
#4 – Core is center of #1 pin bearing to bottom of #2 contour.
#5 – Core is #2 contour to center of #2 main bearing.
#6 – Core is center of #2 main bearing and #3 contour.
#7 & #8 – Cores are #2 pin bearing (#7-bottom and #8-top).
#9 – Core is #2 cntrwt. to center of #3 main bearing.
#10 – Core is center of #3 main bearing and #3 contour.
#10 & #11 – Cores are #3 pin bearing (#11-bottom & #12-top).
#13 – Core is #6 contour to center of #4 main bearing.
#14 – Core is #7 contour to center of #4 main bearing.
#15 – Core is #4 pin bearing center to bottom of #7 contour.
#16 – Core is #4 pin bearing from center to top.
#17 – Is #4 counterweight to center of #5 main bearing.
#18 – Flywheel end of crankshaft.
#19 – Start of riser with break-off.
#20 & #21 – Are riser cores.


19-Hot metal tap-over from central feed to four risers. Ingate from downsprue.
14-Ingate from down sprue.
6-Ingate from down sprue.
6-Ingate from down sprue.

SLAB CORES ASSEMBLY SHOWING GATING FOR THE FORD CAST STEEL 8 CYLINDER CRANKSHAFT

■ Fig. 29 – Page of specification showing specifications for the build-up of the core assemblies and the gates and runners for feeding this casting

The second section is detailed and specifies the bath compositions, the fixtures, the amount of metal to deposit, and so on. The make-up of the bath and the maintenance

STEEL FOUNDRY
SECTION VIII
RAW MATERIAL SPECIFICATIONS

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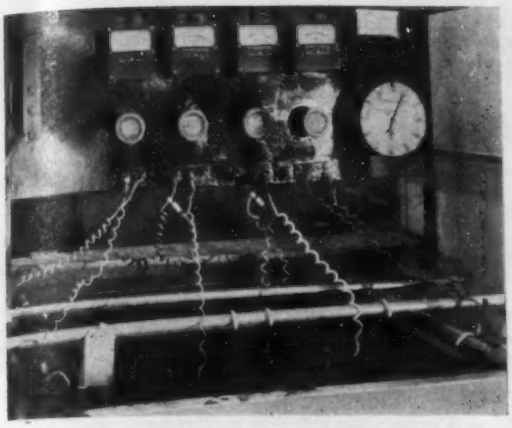
■ Fig. 30 – Index page for the raw material specifications



■ Fig. 31 – Plating bath

Fig. 32 – Fixture with uniform work-to-anode distances

of it is directed by the control men from the chemical department.

Plating to exact size requires amperage control much closer than usual.

Fig. 31 shows a plating bath with ammeters on each rack and a sweep-second clock to time the plating accurately. Besides accurately metered current and timing, the work-to-anode distance must also be controlled.

Fig. 32 shows a fixture with uniform work-to-anode distances. The electrical leads are direct to the anodes and the rack and anodes are lacquer covered except where current must flow through the solution. This lacquering gives bearings with even metal coats to the edges of the bearings.

Test pieces are plated twice daily to get the exact

■ Fig. 33 – Measurement of the plate on the surface of a bearing

amount of metal deposited in a given time. Knowing the amount of plate on a measured gage in terms of time at definite amperage per square inch, the bearing plating time and current is specified twice daily by the chemical engineering department.

The meters are checked daily with a master meter. When the amount of metal to plate is specified by thickness, it is gaged directly.

Fig. 33 shows the measurement of the plate on the surface of a bearing.

■ Painting

As another illustration of process specification under methods of treatment we will use painting operations.

We had these in civilian work and the form in war work is the same. In war work we had to make these compatible with Army painting specifications. Our specifications and often the Army ones show mandatory compositions of paint.

These specifications cover the operation in detail. The color, the name of the coat, the number, and sequence is given. We divide the vehicle parts into groups according to their individual requirements, wood parts, sheet metal, frames, underbody, engines, bodies, and so on. We specify the metal cleaners in composition as well as when used and how.

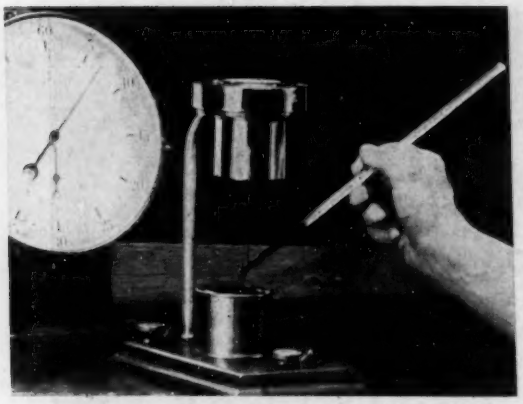
In the actual paint operation, the paint is brought to a consistency measured by viscosity determination. This is done in a Ford viscosity cup (Fig. 34). The cup was designed to permit control right in the shop. It is sturdy and simple.

Some particular paints that are used in dip tanks are so reactive with air that their condition must be watched. We have antioxidant in the paint in such tanks and run tests to see whether the reaction with air has started. We can then add antioxidant to keep the paint protected when required.

In Army as well as in civilian painting, desired colors are maintained. For this, color chips of standardized hue and luster are provided and maintained by the chemical engineering department control operators.

With paint of specified nonvolatile content and applied at specified viscosity, the applied and dried film has a thickness that is capable of direct control.

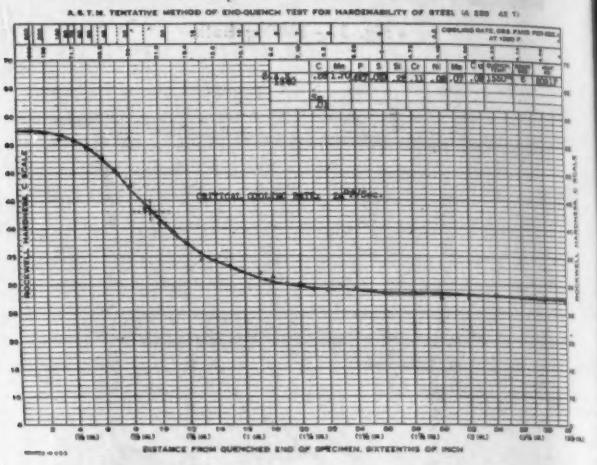
The film thickness in any given type of paint controls



■ Fig. 34 (left) – Viscosity cup to determine consistency of paint

■ Fig. 35 (right) – Measuring thickness of paint film

■ Fig. 36 – Finish is baked on in infra-red radiant heat ovens



■ Fig. 38 – Report of hardenability of a steel sample

Purpose: To determine the Austenitic Grain Size

REPORT OF MICROSTRUCTURE

DATE December 15, 1941 LAB 2990

Heat No. A 5666 Sample No.45

ANALYSIS C .29 - Mn .80 - Cr .67 - Ni .45

P .016 - M .019 - Si .27 - Mo .43 - Cu .10

NE 8620 Steel

TREATMENT Normalize at 1700°F.

Carburize at 1700°F - Furnace cool.

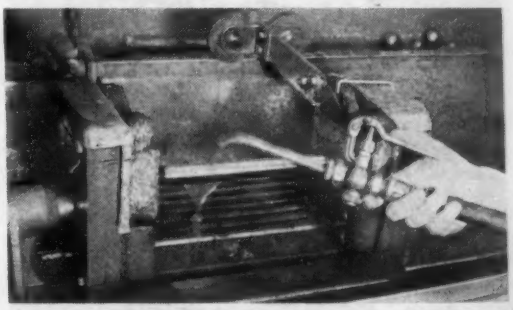
MAGNIFICATION 100 X

ETCH Nital

REMARKS Photomicrograph shows the

Austenitic Grain Size to be No. 6

■ Fig. 37 – Typical report on grain size of an open-hearth steel billet



■ Fig. 39 – Magnetic testing of a steel bar

its durability and we measure it on parts and bodies by direct-reading instruments (Fig. 35).

The baking of these finishes in infrared radiant heat ovens, a Ford development (Fig. 36) is controlled by actual surface temperature measurements on the parts.

■ Testing for Effectiveness of Control

"The proof of the pudding is in the eating thereof." As evident from the description above, what is raw material for one part of the plant may be a finished article for another part, and in general, the testing of this raw material in process is carried out on the job.

The quality of the open-hearth product is checked for grain size from the billet.

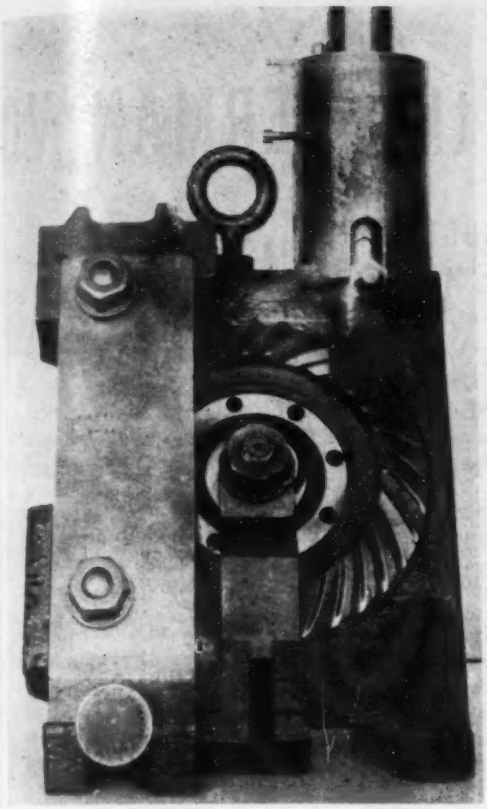
A typical report of this determination is shown in Fig. 37.

A hardenability sample is taken at the same time and a characteristic report is shown as in Fig. 38.

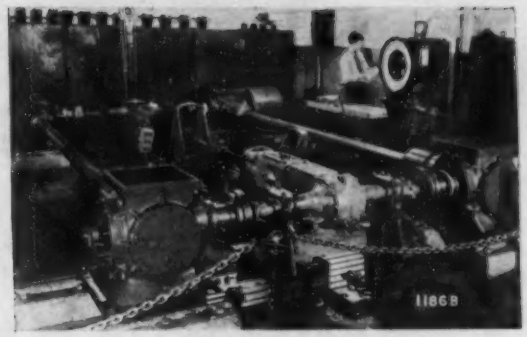
Later, as the steel is rolled into bars, a sample may be taken for magnetic testing. If there are seams, laps, or nonmetallic stringer inclusions, they show up here (Fig. 39).

We test the quality of gear steel by making gears. If we make transmission gears, we forge a number and finish them completely. In the cutting operation we determine the machinability. In the heat-treatment we watch that warpage and distortion don't occur.

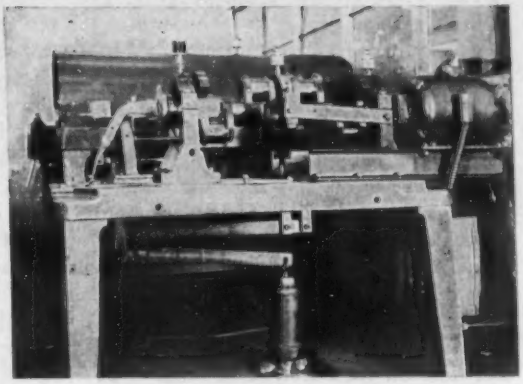
We test the finished gears for hardness and for tooth



■ Fig. 40 (left) – Checking tooth strength of differential ring gear



■ Fig. 41 (upper right) – Dynamometer testing of gears built into axles



■ Fig. 42 (lower right) – Endurance testing of a cast crankshaft

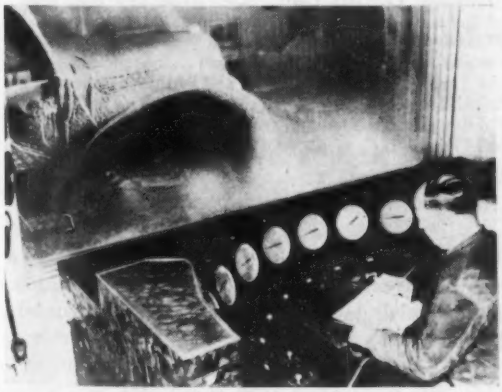
impact and static strength (Fig. 40). With gears of proved strength, we build up transmissions and run them in dynamometers. We make sure they are quiet and that no pitchline pitting or tooth breakage occurs.

With rear-axle gear steels, forgings are made, normalized, machined, and hardened. Spiral bevel gears are watched so that distortion such as unwind doesn't occur. Hardness, case depth, and tooth strength are checked. The gears are built into axles that are run in a dynamometer shown in Fig. 41.

To test its endurance we run it in a fixture stronger than the block it will run in so that any bending is in the shaft.

The quality control group runs such tests, as shown in Fig. 42.

When these parts and subassemblies are built into finished vehicles they are run in the weather tunnel (Fig. 43), or on our testing tracks, as shown in Figs. 44 and 45. Here all sorts of weather and road conditions are encountered.



■ Fig. 43 (left) – Testing in the weather tunnel

■ Fig. 44 (right) – Testing finished vehicle

This special rear-axle dynamometer runs one axle against another in a balanced fashion under controlled and measured loads and speeds.

The temperatures are kept above normal operating conditions so that lubrication conditions are more severe than may be encountered in service.

One of the products of the steel foundry is a cast crankshaft.

■ Conclusions

We hope that you agree with us that now, more than ever, there is need for better quality control of engineering material.

We have attempted to explain something of our methods.

1. By a wider and better method of using material specifications in conjunction with process control.

concluded on page 367

Particulars of the GERMAN VDM

by JOHN D. WAUGH
Lockheed Overseas Corp.*

THE VDM electric propeller (Fig. 1) is designed and built by the Vereinigte Deutsche Metallwerke, Aktiengesellschaft, of Frankfurt-am-Main-Heddernheim, and Hamburg, Gross-Borstel, Germany. This engineering company has spent many years of research and development on the propeller and has refined it to the point where it is used almost exclusively on all types of operational aircraft in the Luftwaffe.

The exceptions are the Junkers hydraulic motor operated propeller and the Argus aerodynamically operated type. These propellers have a more limited range of adaptability than the VDM and are thus used less widely.

First public showing of the VDM was held at the International Air Meeting in Zurich, Switzerland, 1937. From that time till the outbreak of war, VDM achieved distinction in being fitted to the Heinkel fighters which established world speed records of 392.5 mph and 463.9 mph, and was also fitted to the Messerschmitt 109 fighter that attained 469.2 mph, the present official world's land plane speed record. The Dornier Do.18, which flew to a world's record for long distance, likewise mounted VDM propellers.

In England the VDM was given spinning and flight endurance tests by the Royal Aircraft Establishment, Farnborough, in 1938, and was approved for use on civil aircraft. Shortly thereafter Constant Speed Airscrews, Ltd., Warwick, obtained the sole British license to manufacture the VDM propeller and patented quick-detachable spinner. The present conflict stopped the production of the propeller, but did not stop production of the spinner, which was improved by Constant Speed Airscrews, and is now used in quantity by the Royal Air Force.

■ General Description

The VDM propeller is strictly unusual when compared to an American design such as the Curtiss electric, which uses the same general idea of control and operation by electricity.

Pitch-change energy is derived from a small reversible electric motor mounted on the engine crankcase and connected to the propeller by a flexible shaft. This shaft enters a small primary drive reduction gear box attached to a large annular gear box fixed to the rear of the propeller hub.

The outside of the annular gear box is secured against rotation with the hub by a bracket to the engine nose case and houses the epicyclic pitch-change gears, some of which rotate with the hub.

Pinion gears engage the outlet drive of the annular gear box and drive worm shafts entering the hub blade sockets and mesh with worm gears integral with the blade adaptors.

THE VDM electric propeller is used almost exclusively on all types of operational aircraft in the German air force.

This propeller, Mr. Waugh concludes after his thorough study, is versatile, well designed, and well constructed; however, it is not superior to comparable British and American units.

Pitch-change rate of the model described here is inadequate for high-performance aircraft, but the propeller is very easily and quickly disassembled and serviced. The model is easy to manufacture, but, because of the large numbers of small parts, takes longer for inspection and assembly than for American propellers.

The VDM spinner is well worth study, as it is superior to American spinners, at least from the removal and installation standpoint. It takes only a matter of seconds to remove or install the VDM spinner, while American spinners take much more time and effort.

■

THE AUTHOR: JOHN D. WAUGH (J '42) is technical project specialist for Curtiss-Wright Corp., Propeller Division. Previously, he was an instructor in aircraft propellers for Lockheed Overseas Corp., in England. Here all types of propellers had been studied by company technicians and RAF personnel using American-built equipment. He also performed a liaison function between the company and British propeller manufacturers and British Government technical agencies engaged in propeller work of all natures. For a time he was senior instructor in propellers in the Air Force Training Program.

The *Aeroplane* drawing in Fig. 2 shows the general arrangement of the component parts and driving mechanism.

Since the motive power for blade displacement is led to the propeller by a flexible shaft, it is obvious that any means of rotating the shaft will serve the same purpose as the electric motor. Accordingly, the VDM people made known that their propeller could also be supplied with hydraulic, pneumatic, mechanical, or manual means of operation.

All VDM propellers are of hollow-shaft construction for the purpose of having all types and models readily adaptable to motor-cannon installations. Likewise the pre-war

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* Now with Propeller Division, Curtiss-Wright Corp.

LECTRIC PROPELLER

hubs were built with a standard shaft bore which could be fitted with spline sleeves of any dimensions, making a VDM applicable to any domestic or foreign engine shaft.

Blade retention is accomplished by simply screwing a blade into the hub adaptor and locking it against rotation by a wedge ring. This applies to duralumin, magnesium, or compressed wood blades, though dural is chiefly used.

Blade cuffs are not used in conjunction with VDM blades because the blades are effectively flared down to the root and the hub is of small diameter permitting satisfactory cooling of closely cowled radial engines.

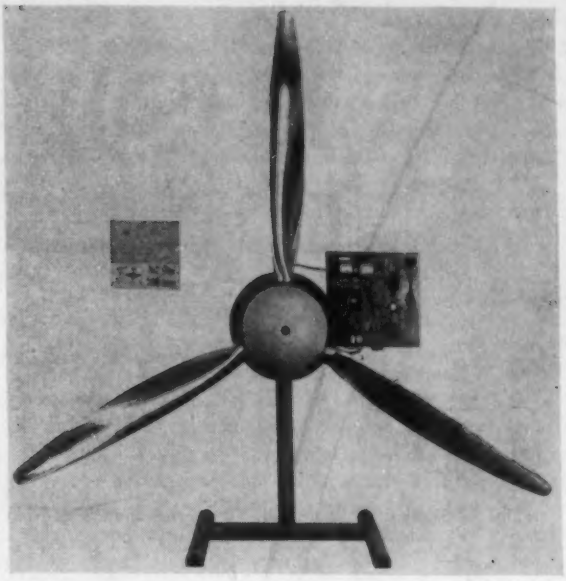
No special allowance is incorporated for the increased centrifugal twisting moment to low pitch of flared blades over nonflared blades because VDM blades are driven to low pitch and high pitch mechanically as compared to the Hamilton hydromatic, which depends upon the assistance of the centrifugal twisting moment to help motivate the blades to low pitch and must have blades designed with this factor considered. Of course, additional power is required to move the blades to high pitch when the centrifugal twisting moment is increased.

Control of the propeller is effected by conventional electric cockpit switches disposed cleverly for the convenience of the pilot. These switches enable the propeller to be operated on the ground with the engine dead, operated in flight in fixed pitch, selective pitch, constant-speed, and feathering. Reverse thrust control could be easily applied with no modification of the propeller. The blades will turn through 360 deg unless limited by electric motor cut-out switches.

Electric and mechanical pitch indicators are supplied and installed with VDM propeller control systems. The electric indicator is generally used on multiengined ships where mechanical drives would be too long for satisfactory indication. The mechanical indicator is primarily for the short cable drive a single-engined ship affords. However, both indicators are sometimes used together – the electrical indicator on the instrument panel and the mechanical indicator in the engine nacelle where the pilot can observe it from his seat in a multiengined ship.

The exact value of a pitch indicator is not known to the writer; but discussions with flying personnel sum up to the conclusion that it is just another gadget. The Army Air Forces have evidently been of the same opinion because no propeller on an Army ship has used an indicator since the Lycoming-Smith mechanical propeller was in general use.

The Germans must believe that for certain military operations, such as dive bombing or long formation flights, an indicator of blade position will assist the pilot in more speedily selecting and synchronizing a given set of conditions of engine speed, flying speed, and power output. For "recipe book" pilots, this would definitely be an advantage.



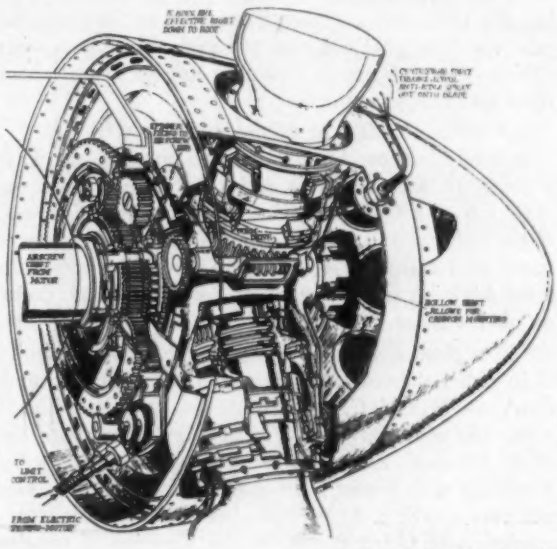
■ Fig. 1 – Complete VDM propeller and electric controls

■ Epicyclic Pitch-Change Gearing

The schematic diagram in Fig. 3 shows the disposition of the gears comprising the pitch-change train and illustrates the motor-to-blade relationship.

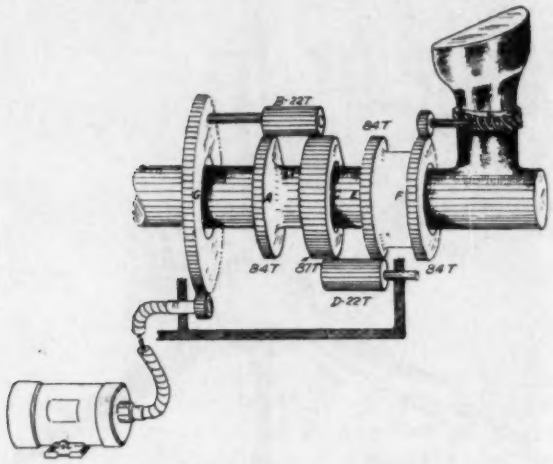
Gear *A* is splined to the hub and rotates at propeller speed. Gear *C* is free to rotate relative to the hub, and gears *E* and *F* are linked together and free to rotate relative to the hub.

Gears *A*, *E*, and *F* each have 84 teeth; gear *C* has 87 teeth cut on the same diameter blank as *A* and *E*. Thus, pinion gear *B*, with 22 teeth can mesh with gears *A* and *C*; pinion gear *D*, also 22 teeth, can mesh with *C* and *E*.



Copyright, *The Aeroplane*

■ Fig. 2 – Perspective-sectioned view – VDM propeller



■ Fig. 3 – Schematic diagram of VDM pitch-change gearing

Pinions *B* and *D* each use slightly less than half the face width of *C*, and use the rest of their width to mesh with their adjacent mating gears *A* and *E*.

At first glance this arrangement does not seem quite reasonable, but upon close examination of gear *C* the pitch of its teeth is found to be not exactly accurate with that of the pinions. This, however, has been so carefully calculated and the gears so precisely made that the arrangement works perfectly.

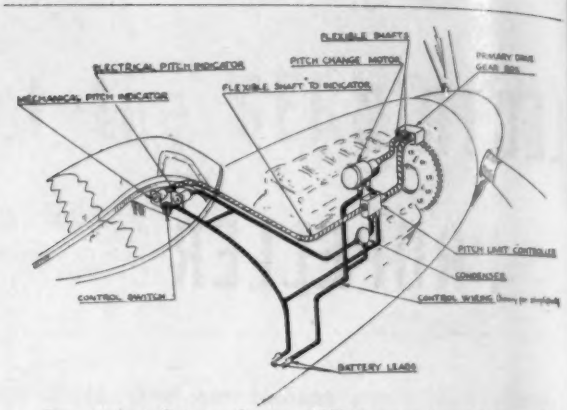
Gear *E* is linked to *F*, which meshes with the pinion and worm shaft driving the worm gear integral with the blade adaptor. Thus, it is apparent that any movement of the *E*, *F* combination relative to the hub will result in a pitch change.

Pinion *D* is free to rotate on a fixed shaft mounted in the gear housing, which is anchored to the engine. Pinion *B* is free to rotate on a fixed shaft mounted on the rear plate of the gear housing. This plate is fixed to gear *G* and is free to rotate relative to the gear housing and the hub, being isolated from each by roller bearings. Gear *G* has 150 teeth and is driven by a small 10-tooth gear in the primary drive gear box attached to the large gear housing. Pinions *B* and *D* are in triplicate, equally spaced about the front and rear gear housing plates.

Operation of this gear system is as follows:

In level flight condition, where no pitch change is taking place, gear *G* is stationary and a simple gear train results. $A/B \times B/C \times C/D \times D/E = 0$ and, as explained, *A* and *E* have the same number of teeth, *E* rotates at the same speed as *A* and since *A* is fixed to the hub, *E* rotates in the same direction and at the same speed as the hub; therefore, no pitch change takes place.

When pitch change is desired and the motor rotates gear *G* in either direction, planet gear *B* is rotated around the sun gears *A* and *C*, establishing an epicyclic train. One revolution of *G* moves *C* 3/87 of a revolution with respect to *A*. Gears *C*, *D*, and *E* can now be treated as a simple gear train with a gear ratio of 87/84 from *C* to *E*. Gear *E* will rotate $3/87 \times 87/84 = 1/28$ of a revolution for one revolution of *G* irrespective of the direction of rotation of *G* or the speed of *A*. *E* coupled to *F* and driving the worm gear to the blade completes the train and causes a pitch change.



■ Fig. 4 – Installation scheme of VDM nonautomatic propeller

■ Types of Installations

There are two types of installations of the VDM propeller in Luftwaffe aircraft:

1. The nonautomatic installation has both mechanical and electrical pitch indicators and depends upon pilot selection to operate the propeller at all times. This installation can be converted to automatic operation.

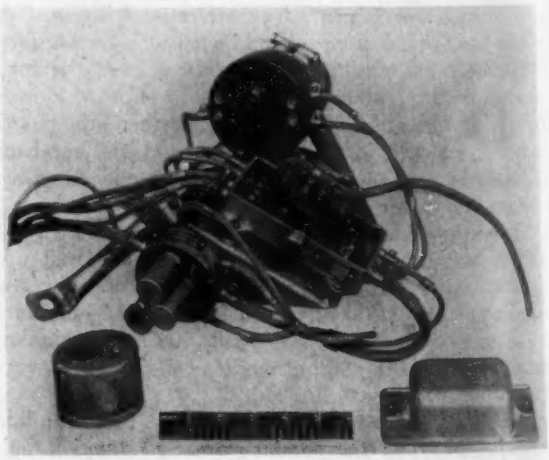
2. The automatic installation incorporates electrical and mechanical pitch indicators, a governor, and a relay, and will operate in constant speed after a fashion.

A description of these two systems follows.

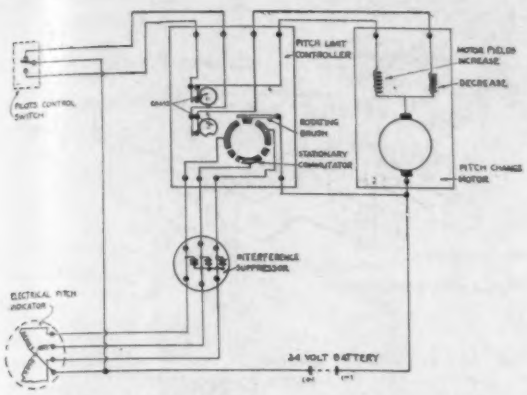
■ Nonautomatic System

Fig. 4 shows the nonautomatic propeller installation in an aircraft. Here it will be noted that a pitch-change motor and pitch-limit controller are connected to the propeller by two flexible shafts. From the other side of the pitch-limit controller another shaft goes to the mechanical pitch indicator in the cockpit. A control switch and electrical pitch indicator on the instrument panel and an interference suppressing condenser complete the control system. The motor, pitch-limit controller, condenser, and wiring are shown in Fig. 5.

By reference to Fig. 2 it will be noted that the motor and pitch-limit control shafts enter the primary drive gear box, where a reduction takes place. The motor shaft to blade reduction is 24,010:1, the pitch-limit control shaft



■ Fig. 5 – Motor, pitch-limit controller, and condenser of nonautomatic system



■ Fig. 6 – Typical VDM nonautomatic propeller wiring diagram

7139:1, making 1-deg pitch change equal 19.9 revolutions of the limit control shaft.

When the control switch is moved to the high-pitch or low-pitch position, current activates the reversible pitch-change motor and changes the pitch of the blades in a manner previously described. The primary drive gear box now rotates the shaft to the limit controller and mechanical pitch indicator. The shaft from the primary drive goes through the limit controller to the indicator and also operates a set of cams in the controller which will switch off the motor current when the blades reach a predetermined blade angle setting.

Electrical pitch indication impulses are also obtained from the pitch-limit controller. If feathering is desired, overriding the control switch in the high-pitch position will move the blades to the feathered position. This feature is found only on certain installations.

(a) The pitch-change motor for the nonautomatic system is of Bosch manufacture. Construction is of the completely enclosed series type; a pressed metal cover fits over the commutator end, and a flexible drive connection emerges from the anticommutator end.

The eccentrically mounted armature is carried in magneto-type bearings in the mild steel frame. Eccentric mounting of the armature allows for only one pole shoe, the magnetic circuit being completed through the pole case and armature. This pole shoe is made of mild steel and carries the two field windings of opposite polarity. One end of each winding is connected to one of the brushes. Flexible leads from the other brush and from the free ends of the field windings are taken to three terminals in the commutator end so that direction of rotation can be selected or changed by switching leads. Condensers are connected between the line brush and frame and between the free ends of both field windings and the frame.

At a speed of 2400 rpm and an input of 24 v, 8.2 amp, the torque is 2.4 lb-in. Overall length 5½ in., diameter 3 in., weight 4¾ lb.

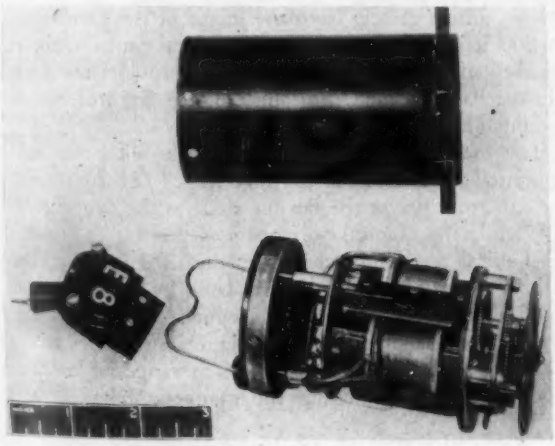
(b) The pitch-limit controller, as previously described, stops the pitch-change motor at the upper and lower limits of blade angle and sends impulses to the electrical indicator.

The motor control mechanism is gear driven at a reduction of 1000:1 by the flexible drive from the primary drive gear box. Two cams actuated by this reduction gearing open and close two sets of contacts which are in each of the two field circuits. The contacts are closed, completing each circuit in turn, until the preset high- or low-pitch position is reached. Here the proper contact is opened by its cam and the motor circuit is broken.

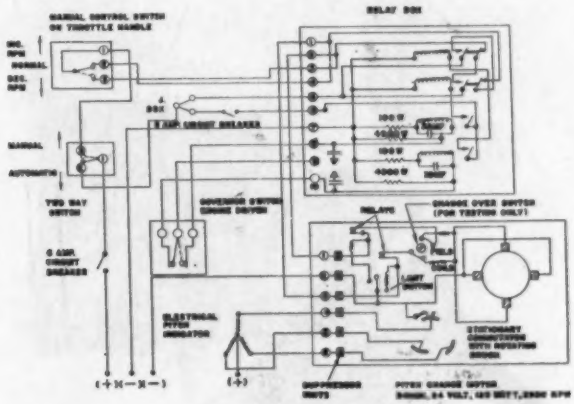
The pitch indicator operating mechanism consists of a fixed commutator having three main segments, each connected to the electrical pitch indicator. As the flexible shaft drives a rotating brush, at a reduction ratio of 15:1, around the commutator, a circuit is intermittently established through each segment of the commutator to the pitch indicator. Fig. 6 is the wiring diagram of the nonautomatic system.

(c) The electrical pitch indicator shown in Fig. 7 consists of three iron core coils mounted axially at an angle of 120 deg apart. A bipolar armature, of soft iron without windings, rotates on a spindle in the center of the three coils. This spindle drives the clockwork mechanism which operates the pitch-indicating pointers. The coils are connected to the three segments in the pitch-limit controller and are energized in turn, attracting the armature into rotary motion.

Each coil is provided with a U-shaped piece of soft iron pivoted on the side remote from the armature. When the coil is energized, this piece of iron is attracted to the iron core of the coil so that one tip projects through a hole in the core until it just touches an iron vane on the armature. This meeting short circuits the magnetic air gap between the iron core and the armature, thus acting as a magnetic



■ Fig. 7 – Electrical pitch indicator



■ Fig. 8 – Typical VDM automatic propeller wiring diagram – Messerschmitt 109E

lock preventing oscillation of the armature. An auxiliary lock for momentary holding takes the form of a spring-loaded pawl acting on a notched wheel in the clockwork gear train.

The face of the indicator is marked off like the face of a clock; with twelve equal divisions, each divided into five smaller units. Two clock hands, one short and one long, correspond to minute and hour hands. Each "hour" division equals 6 deg and each "minute" division equals 0.1 deg.

For every installation the take-off pitch is represented by 12:00 o'clock. So, if the clock hands sit at 3:30 o'clock, the pitch would be read as $3 \times 6 + 30 \times 0.1$ or 21 deg high pitch above take-off position.

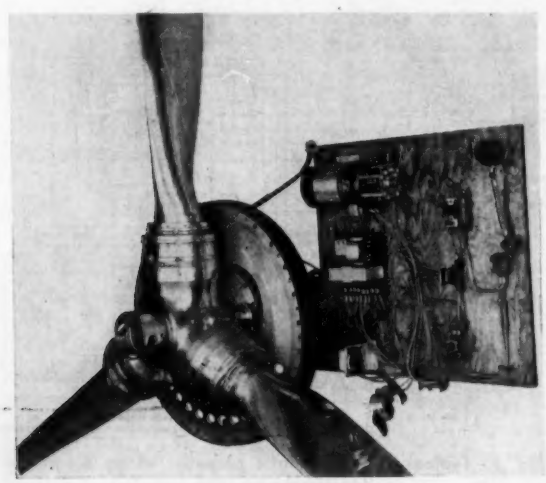
■ Automatic System

The automatic system consists of a pitch-change motor controlled by the pilot from a spring-loaded two-way switch located on the throttle handle, and by an engine driven governor actuating a relay. Selection of manual or automatic operation is obtained by a conventional selector switch. Fig. 8 shows the wiring diagram of the automatic system and Fig. 9 is a close-up of the actual components minus the governor. A circuit breaker of 6-amp capacity protects the automatic system from damage, but does not prevent use of the manual system should the automatic fail. Fig. 10 illustrates the automatic system installation.

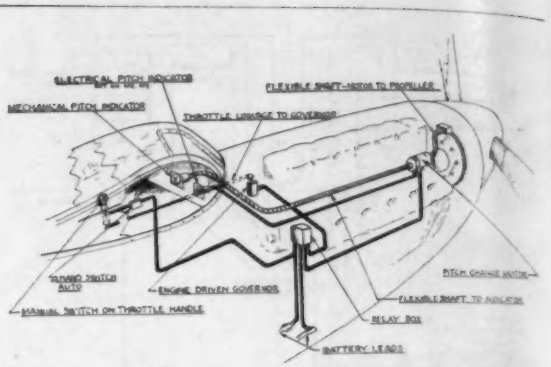
The manual or automatic switch is located just ahead of the throttle so that full-throttle position will put the switch in automatic, thus preventing overspeeding of the engine. However, the catch which puts the switch into the automatic position can be swung aside letting the throttle be opened while the propeller control system is in manual.

The propeller governor is a simple flyball-operated, single-pole, double-throw electric switch driven at one-half crankshaft speed by the engine crankcase gear box, which drives the machine gun interrupter and magneto. Throttle linkage to the governor operates on the idea of having a loading spring in the governor swinging in cam fashion against throttle-opening and flyball pressure.

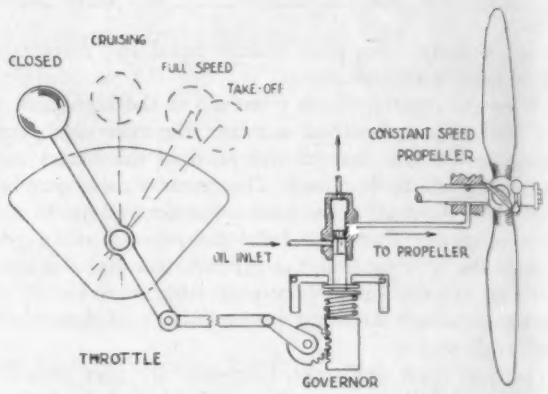
This single lever control, in which any rpm corresponds to a fixed predetermined value of manifold pressure, has virtue in that operation of the engine-propeller combination is much simpler for the relatively unskilled pilot with only slight loss of economy for some flight conditions.



■ Fig. 9 – Hub, motor, relay, and switches with covers removed



■ Fig. 10 – Installation scheme of VDM automatic propeller – Messerschmitt 109



■ Fig. 11 – Junkers-Hamilton automatic throttle-governor propeller control

Likewise, the manually operated low-pitch, high-pitch switch button which forms the throttle knob is a deliberate attempt, and accomplishment, to simplify operation for the pilot. Adoption of a similar system on American aircraft, fighters in particular, would probably prove very popular.

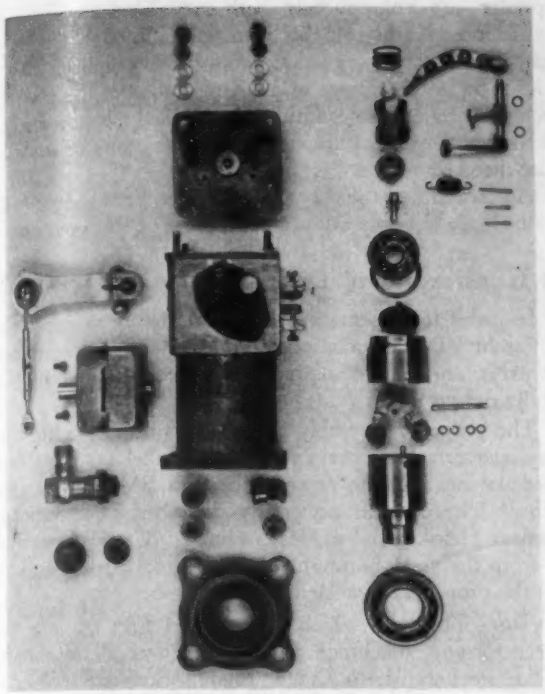
German views on automatic engine-propeller arrangements may be found in an article by K. A. Schmidt.[1]

Fig. 11 is a schematic view of an early German throttle-propeller linkage, which the Junkers company applied to the Hamilton licensed constant-speed propeller and which they built at that time.

(a) The Bosch pitch-change motor for the automatic system differs greatly from the motor of the nonautomatic in that it incorporates the pitch-limiting contacts, the electrical pitch-indicating transmitter and provides the drive to the mechanical pitch indicator. These features eliminate the pitch-limit controller.

The motor is a four-pole series type with two sets of oppositely wound fields for reversing. The front end of the armature drives the flexible shaft to the propeller gear box while the rear take-off operates gearing to rotate the mechanical pitch-indicator outlet once for every 3.3 revolutions of the armature. Also rotated by this gearing is a small disc mounting a button which contacts a pair of brushes connected to two magnetic relays in series with the field coils. The disc limits the pitch range by contacting the brushes and actuating the magnetic relays – one for

[1] See *Aircraft Engineering*, Vol. 12, August, 1940, pp. 233-236: "The Bases of V. P. Airscrew Design," by K. A. Schmidt. From *Luftwissen*, October, 1939, pp. 267-273.



■ Fig. 12 – Disassembled electric governor

high pitch and one for low pitch. To obtain different values of pitch, one of the brushes contacted by the rotating disc is adjustable by screwdriver from outside the housing.

The electrical pitch-indicator transmitter takes the form found in the pitch-limit controller, that is, a stationary 3-segment commutator with a rotating brush. All leads from the motor are equipped with interference suppressing condensers instead of the single large condenser found in the nonautomatic system. The motor is rated at 2500 rpm, 24 v, 120 w. Weight is 8 lb, 4 oz.

(b) The manual control switch which forms the knob of the throttle lever operates the pitch-change motor in either direction when the system is in manual. It is essentially a two-way press action switch, spring loaded in the central "off" position.

The switching mechanism consists of two silver-plated discs mounted on, but insulated from, two flat springs in which they "float." When the rocking lever forces one of the springs down, the "floating" disc contacts two small silver contacts mounted on short brass strips. The electric contact is made from the periphery of the disc contact to the fixed contacts.

The tendency of the discs is to rotate from the inertia of springing back, so a different point of contact is established at each actuation. A bakelite base forms the mounting pad for all contacts and springs. The switch is small, 2 x 2⅜ in. and weighs but a little over 5 oz.

(c) The electric governor, which is shown disassembled in Fig. 12, is manufactured by the Hamburger Metallverarbeitungs-Gesellschaft, of Hamburg. This company is evidently an accessories maker, constructing governors for VDM from VDM design.

The governor cap, body, and base are die cast from magnesium alloy and have very few machining requirements. Bronze bushings are pressed into the housing to carry small shafts and pins.

The body is divided into two main chambers; the lower chamber housing the speeder cup and flyweight assembly, and the top chamber housing the loading spring and actuating levers. One side of the top chamber is separate from the levers and houses only the current distributing contact arm.

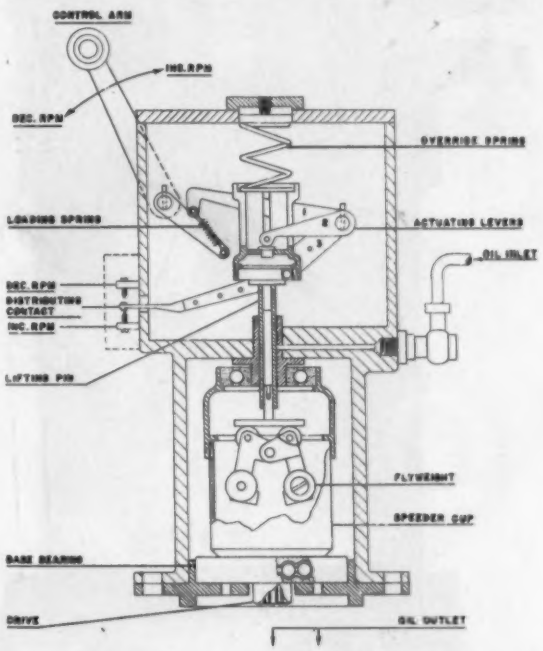
Oil from an outside pressure line is led to the lifting pin bushing connecting the upper levers with the flyweights in the lower chamber. From this bushing, oil distributed to the upper and lower chambers drains out the base. Small mesh wire strainers in the inlet elbow filter the incoming oil.

The control arm from the throttle linkage, which increases or decreases the tension of the loading spring, is equipped with a pointer marked off in the inevitable German vernier fashion. In conjunction with this pointer is a small plate screwed to the side of the governor and marked off in degrees from 0 to 80. As the control arm swings back and forth, the pointer with its vernier scale will indicate the degrees and fractions of degrees of travel.

The distributing contact lever emerges from its oil-tight compartment to meet the stationary decrease and increase rpm contacts mounted on the outside of the governor case inside a small aluminum cover.

Both moving and stationary contacts are made of silver and show no signs of burning. The increase rpm contact is adjustable by screwdriver from outside the contact cover. Both adjustable contacts are held in position by clamping screws.

The schematic cutaway view of the governor in Fig. 13 is intended to illustrate operation only. By reference to this drawing it will be seen that the small flyweights enclosed in the engine driven speeder cup will lift lever No. 2 via a pin when acted upon by centrifugal force. Levers 1, 2, and 3 are fastened to a common shaft by small taper pins. Thus, when lever No. 2 is lifted, No. 1 acts against tension applied by the loading spring. If the rotational velocity of the governor causes the flyweights to



■ Fig. 13 – Cutaway view – VDM electric governor

overcome the loading spring tension set by throttle linkage, levers 1, 2, and 3 are lifted to their upper limit. It will be noted that lever 3, the current carrying contact, will touch the decrease rpm contact and cause the relay to energize the pitch-change motor toward high pitch.

Conversely, if the loading spring tension is greater than the lifting force of the flyweights, lever No. 3 will be directed down to the increase rpm contact with a resultant correction of the blades to a lower pitch and higher rpm.

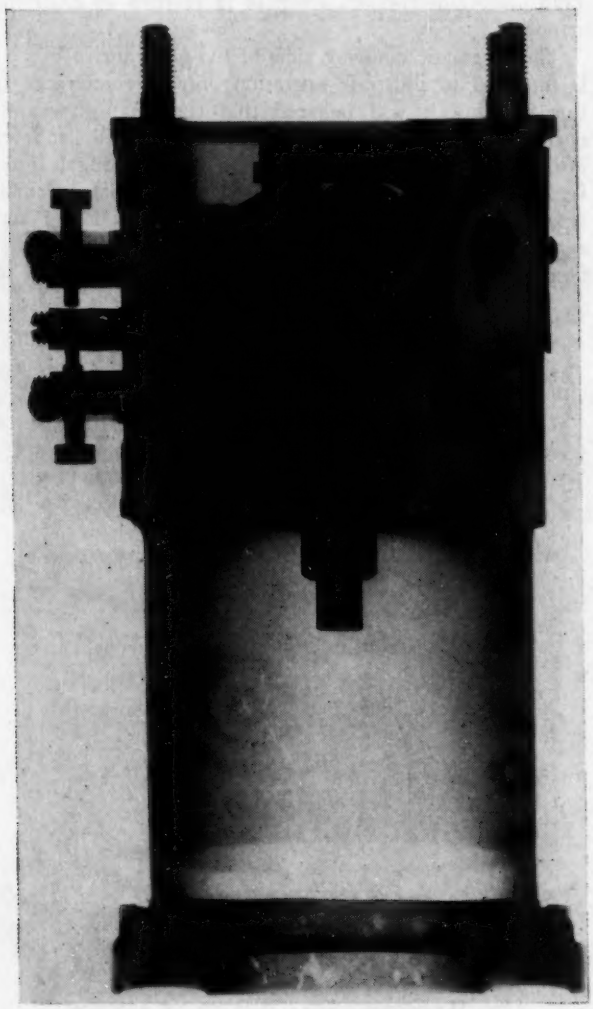
It is readily seen that this governor is limited in sensitivity and wholly lacking in any proportional correction of engine rpm. The Curtiss governor with its proportional correcting feature and the Hamilton governor with sensitive oil metering correction cannot even be compared to this German attempt.

Factory seals were on the specimen examined and all evidences were that it had not been in service long. Workmanship was of the poorest order yet found on a German device. Considerable hand fitting was roughly done. This points to either dilution of labor or production requirements; no similar indications, however, have been found on other propeller components.

Not even the most production-driven British or American propeller manufacturer would pass the fits found in this piece of work. The fit of the ball bearings gave the impression that gasoline had been used as a lubricant. The speeder cup consequently had enough play possibly to cause some binding of the lifting pin, although the design had clearly been intended to avoid this possibility.

Fig. 14 is the picture of an X-ray taken of the governor body casting. Close examination revealed gas porosity and shrinkage flaws in sufficient quantity to cause rejection by Lockheed standards.

Accessibility of the contact points, and light weight (1 lb, 10 oz), are the only admirable features of this governor.



■ Fig. 14 – X-ray of governor casting showing flaws

■ Constructional Details

In order to present as complete a picture as possible of different VDM installations and variations, different units, controls, and sources of data have been studied and liberally drawn upon.

The propeller assembly herein described is 11 ft, 6 in. in diameter and was taken from a Heinkel 115, twin float, mine-laying seaplane powered by two BMW 880 hp engines. The controls previously described came variously from a Heinkel and a Messerschmitt. The blade analysis is from the very thorough report of Rotol Airscrews, Ltd., on the propellers of a Heinkel 111 bomber.

Hub – The hub, which is shown in Fig. 15, is a one-piece forging, machined and ground over all in chrome-nickel steel similar to X4340. Blade sockets are recessed at the bottom to receive the inner journal bearing and are threaded at the rim with conventional V-threads to retain the blade bearing and adaptor housing.

Bosses on the sides of the blade sockets carry the worm drive shafts and house the bronze worms. The worm, which is denied end play by a close-fitting double set of ball thrust bearings, is keyed to its drive shaft, the shaft being carried by three sets of needle rollers running in direct contact with the hub.

An extension of the rear of the hub is splined inside to fit the engine shaft and tapered to fit the rear cone. The outside is splined and machined to receive the annular pitch-change gear box. A grease fitting between two hub sockets leads to the gear box shelf.

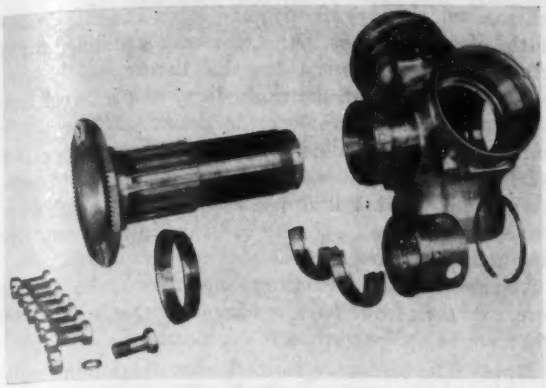
The center and front of the hub shaft bore are entirely hollow; the shaft splines being in the rear extension. This is a good feature from the standpoint of weight saving. A front cone taper and retaining nut snap ring recess are cut in the front of the hub.

Eight holes are drilled and threaded in each hub socket for the purpose of mounting the spinner and locking plates. These holes seem contrary to usual practice regarding a highly stressed propeller hub, but are evidently satisfactory in this instance because of thick walls and careful hand finishing of the holes.

The front of the hub is provided with a short annular extension which provides a seat for the spinner baffle.

The stub shaft which carries the hub is provided with a Hirth-type serrated flange held to the engine crankshaft flange by eight ground bolts. The rear cone is bronze, as is the split front cone. The retaining nut is drilled with small holes for locking purposes and large holes for installation and removal by a propeller bar.

The condition of the hub was found to be very good. Only the expected amount of galling between the hub and rear cone was evident. The rear shelf extension which carries the gear box was slightly pitted from fretting of the gear box on the shelf. This could be caused by improper



■ Fig. 15 – Stub shaft, hub, cones, and retaining parts

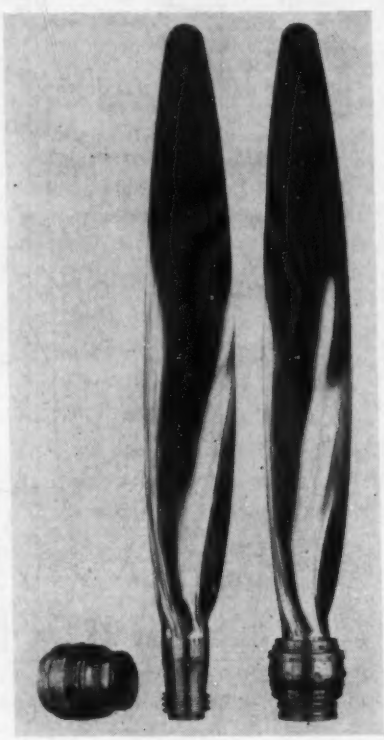
fit, insufficient area, poor retention, or vibration. However, not enough specimens have been examined to determine if this fretting is a habit of VDM hubs.

Cadmium plating on hub, stub shaft, retaining nut, and snap ring was in good condition, although quite thin.

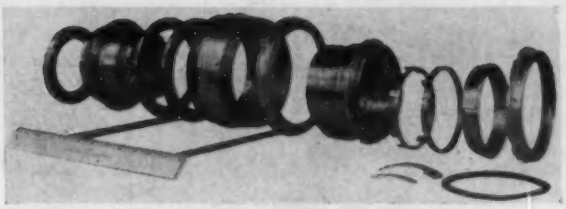
Blade Mounting – Blade mounting in the hub is accomplished by means of a unit assembly which houses an adaptor, threaded to mate the screw threads on the root of the blade, and equipped with radial bearings for bending loads and taper roller bearings for centrifugal loads.

Fig. 16 shows one blade mounting unit installed on a blade and one removed. Fig. 17 is an exploded view of the unit.

To explain the relation of parts and action of the unit, Fig. 18, a cross-section view, will best serve. In this view it will be seen the flanged engine shaft $a-b$ is fastened to the hub boss c by a retaining nut f. The hub blade socket d has an extension socket g threaded into it. This extension carries the outer journal bearing h, which consists of 142 rollers 0.235 x 0.235 in. in two rows and provides a



■ Fig. 16 – Dural blades with mounting units installed and removed

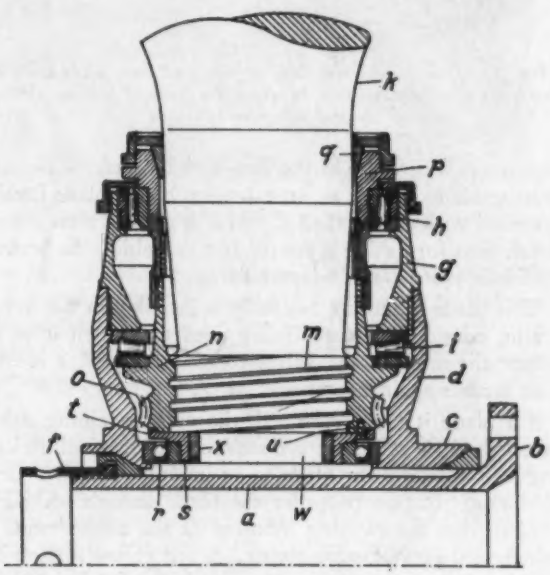


■ Fig. 17 – Exploded blade mounting unit

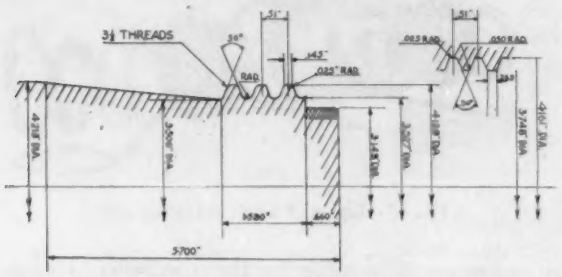
step and means of retention for the taper roller bearings i, which are 0.60 in. long and measure 0.368 in. in diameter at their large end. These taper rollers carry the centrifugal load of over 30 tons and are retained between two hardened races; the outer race tapered and the inner flat.

The dural blade k is held in the blade adaptor n by a screw thread m. The form of this screw thread and adaptor is shown in Fig. 19. A worm wheel o is cut and ground on the lower portion of the adaptor so that the blade may be rotated by the worm in the hub.

An extension p is screwed onto the adaptor, thus providing a race for the outer bearing and a cone seat for the conical locking ring q which is tightened down by a retain-



■ Fig. 18 – VDM blade mounting
a Propeller shaft
b Flange to engine shaft
c Hub boss
d Blade socket
f Retaining nut
g Socket extension
h Outer journal bearings
i Thrust bearing
k Blade
m Screw threads
n Blade adaptor
o Worm wheel
p Adaptor extension
q Conical locking ring
r Inner journal ball bearing
s Adaptor flange
t Centering projection of adaptor flange
u Securing screws – flange to adaptor
w Blade journal – inner bearing
x Blade journal shell



■ Fig. 19 – Thread form of VDM blade and adaptor

ing nut, effectively locking the blade from rotation within the adaptor. The adaptor extension is prevented from rotating relative to the adaptor by a castellated locking ring and circlip.

To establish a bottoming point for the blade, a flange *s* is centered by a projection *t* and secured to the adaptor by screws *u*. This flange then fits in the inner journal ball

■ Fig. 20 – One blade installed in hub and one blade mounting unit partially disassembled to show the conical locking ring, retaining nut, and felt seal

bearing *r*, which sits in the hub socket recess. The inner bearing blade journal *w*, an extension of the blade itself, is provided with a steel shell *x*, which is shrunk onto the soft dural, thus furnishing a strong and unyielding fit between the blade journal and adaptor flange.

This blade mounting assembly is installed in the hub as a unit, considerable force being used to tighten it to the proper aligning marks. When correctly aligned, a locking plate secures against rotation.

If a blade is to be removed, the outer retaining nut is loosened, thus releasing pressure on the conical locking ring and allowing the blade to be unscrewed easily. On a right-hand rotation propeller the blade threads are right-hand so that the twisting moment of the blades tends to tighten and prevent unscrewing. A felt grease seal on the outside of the cone locking nut excludes foreign matter. Fig. 20 shows one blade in its mounting unit and in the hub, also a unit removed from the hub with the felt grease seal, locking cone retaining nut, and locking cone.

The condition of the units was generally good, with grinding marks visible on all bearing surfaces. However, the thrust bearing races and outer journal bearing races were brinelled somewhat. The reason for this was not apparent because several other VDM races have shown no such indications. Heavy grease similar to Mobilgrease No. 2 was found on the bearings in the lower part of the hub sockets with very little on the outer bearings. This seems odd in view of the nature of centrifugal force, but is not odd in view of the construction of the blade mounting unit, which prevents application of lubricant except when disassembled.

It is readily seen that the design of this blade mounting has admirable features, such as the wide spacing of the bearings which carry the bending loads and independence of the thrust bearings. Also, for easy handling, the unscrewing of the blade is a desirable feature.

On the other hand, the multiplicity of parts and manufacturing operations are not too desirable. The simplest and most easily handled type of blade mounting is the bearing stack arrangement wherein a stack of ball bearings carries both centrifugal and bending loads in combination. The Curtiss in the U. S. and Schwarz in Germany typify this method of construction.

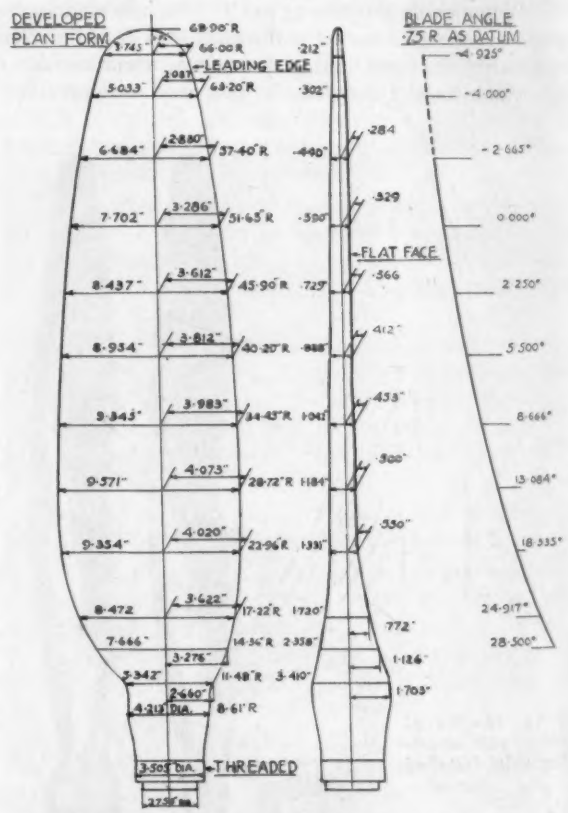
Apparently to avoid outstanding patents, the Germans developed their blade retention knowing that it was heavier and more complicated than foreign designs.

Blade – The blades are formed from duralumin forgings supplied by the Dürener Metallwerke A.-G. Tests of a Heinkel 111 blade have shown that the metal used conforms very closely to standard blade material.

Fig. 21 shows the plan form and angle distribution of the Heinkel 111 blade and illustrates how the airfoil section is carried down close to the hub. The method of screw thread blade retention was covered in the preceding discussion on blade mounting. The screw thread carries the centrifugal load, which is 33 tons at maximum rpm, while the bending load is carried by the locking cone at the outer portion of the hub socket and the inner journal.

The finish of the back of the blade was smooth while the face of the blade appeared as though it had been ground with about 120 grit. No anodic treatment was used – only a chromate primer covered with matt black paint.

Geometric properties of the airfoil cross-section are illus-



■ Fig. 21 – VDM Heinkel 111 blade – right-hand tractor

trated in Fig. 22. Here it will be noted that the leading edge of the blade is sharper and tapers more gradually than comparable British or American designed blades. Very likely this airfoil is a type developed by the German Institute for Aeronautical Research. However, there is a great independence of thought and action amongst the German propeller builders, which leads them to adopt various developments only when it suits their previous designs. The single feature which the three principal German propellers have in common with one another is the worm drive from the pitch changing mechanism to the blades.

The estimated stresses in the blade root at take-off rpm are:

Maximum tensile stress due to bending = 1.19 tons per sq in.

Tensile stress due to centrifugal force = 2.37 tons per sq in.

Total maximum tensile stress = 3.56 tons per sq in.

Chemical composition of the blade alloy in per cent:

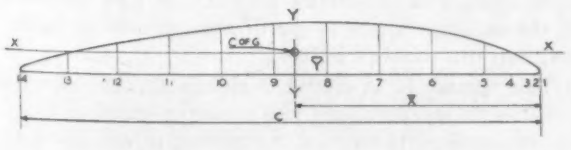
Copper	3.30
Iron	0.46
Silicon	0.43
Manganese	0.68
Magnesium	0.70
Titanium	0.02
Nickel	—
Chromium	—
Molybdenum	—
Aluminum	Remainder

The preceding blade data were obtained by Rotol Airscrews, Ltd., from a Heinkel 111 blade having a weight of 45.6 lb.

The Heinkel 115 blades, which were in the hub examined by Lockheed, have a different plan form and a weight of only 39 lb. Evidence of modification in service and our removal of metal in repair make this weight approximate.

Fig. 23 shows various propeller components and gives their Brinell hardness and tensile strength in tons per square inch.

Pitch-Change Gear Box – The annular gear box shown assembled and in relative position to the hub and shaft in Fig. 24 is mounted on the rear of the hub on an extension, also evident in the same figure. A set of splines prevents rotation and the three small setscrews apparent in the foreground secure the gear box axially. A small primary drive gear box is seen attached to the large annular gearing. This small box receives the flexible shaft drive from

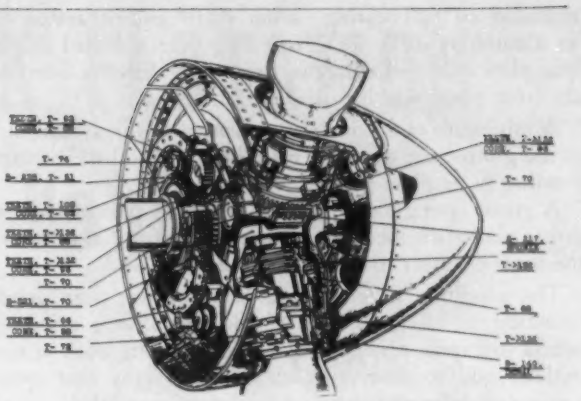


ORDINATE NO.	1	2	3	4	5	6	7	8	9	10	11	12	13	14
% OF CHORD	0	.0125	.025	.05	.10	.20	.30	.40	.50	.60	.70	.80	.90	1.0
% OF MAX. THICKNESS UPPER SURFACE	.155	.260	.382	.456	.635	.865	.974	1.00	.974	.890	.765	.590	.365	.070
% OF MAX. THICKNESS LOWER SURFACE	.133	.041	.027	.053	0	0	0	0	0	0	0	0	0	.070

AREA = .72 C²TM ȳ = .415TM x̄ = .464C

YY = .0396 C⁴TM IXX = .049 C⁴T³M IXY = .0059 C⁴T²M

■ Fig. 22 – Dimension and properties of VDM duralumin section taken at 0.75 radius



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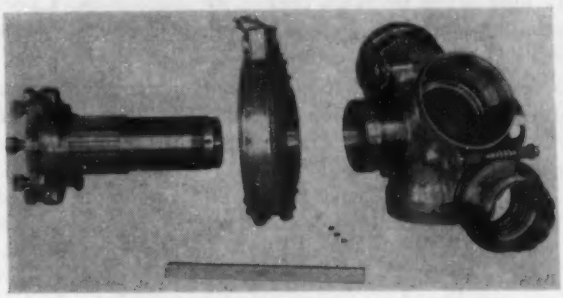
■ Fig. 23 – Hardness and tensile strength of various propeller components

B Brinell hardness

T Tensile strength, tons per square inch

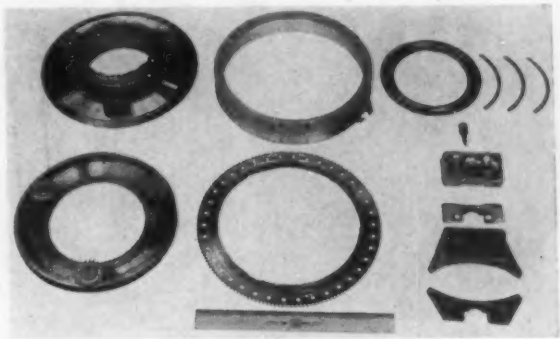


■ Fig. 24 – Stub shaft, gear box, and hub

the motor and provides a reduction of motor speed for the driving of the large ring gear which starts the pitch change previously described. Also, the small gear box operates the pitch-limit controller and mechanical pitch indicator in the nonautomatic system.

The primary drive gears are housed in a small aluminum-alloy gear box provided with ball bearings and mounted on the annular gear box by stamped aluminum brackets. A spring-loaded plunger impinging on a toothed wheel prevents rotation of the driving shaft from the motor, due to friction of the internal gearing attempting to rotate the large ring gear.

Fig. 25 is the layout of the gear box components. The front and rear plates are equipped with three planet gears



■ Fig. 25 – Layout of annular gear box and primary drive gear box

mounted on ball-bearing pinion shafts and separated by an aluminum shell. The large ring gear is bolted to the rear plate by small machine screws. A sleeve is fitted to the front plate, which carries the sun gears.

A minimum of friction is experienced in this system due to the profuse use of roller, plain bronze, and ball bearings running in graphite grease.

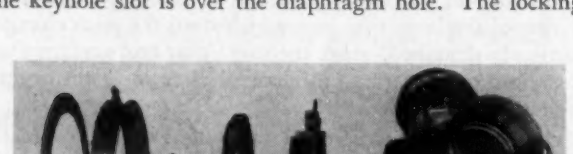
A grease fitting on the front plate allows replenishing of grease and a shielded perforation of the front plate vents the inner chamber to atmospheric pressure.

The condition of gears and bearings was very good. Grinding of all gears was of the first order; grinding marks still quite visible. The corrected tooth work of the middle, odd toothed, gear showed no signs that overlapping of different pitch gear teeth bore a penalty.

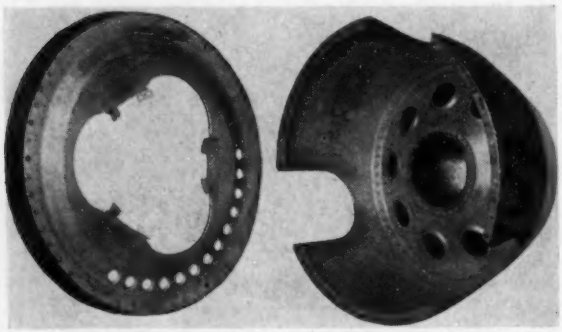
Fig. 26 is an exploded view of the entire pitch-change train. At the highest engine speed and rate of pitch change, the planet gears turn approximately 5900 + 400 rpm. The higher speed is encountered when the blades are turning to low pitch. Normally, the loads on the planets always act on the same side of the teeth and are very light when no pitch change is taking place. Under maximum conditions there is a tooth load of 165 lb at a velocity of 2800 fpm.

Spinner – The VDM spinner is one of the most unique features of the whole propeller assembly. It is composed of two major units; the diaphragm and the shell. These are shown in Fig. 27.

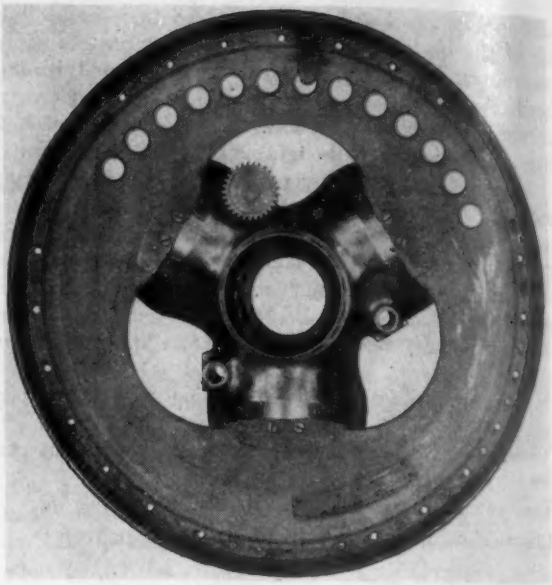
The diaphragm, a heavy gage dural stamping, is secured to the rear of the hub by brackets held to each hub socket by six capscrews. At the outer circumference of the diaphragm, a series of screws hold in place and allow limited rotation of a locking ring equipped with 21 keyhole slots designed to align exactly with 21 holes in the diaphragm when the locking ring is turned so that the round part of the keyhole slot is over the diaphragm hole. The locking



■ Fig. 26 – Exploded gear box in relative position to hub



■ Fig. 27 – Spinner diaphragm and shell



■ Fig. 28 – View showing method of spinner diaphragm attachment to hub

ring securing screws provide small stud projections on the front face of the diaphragm between each hole. Fig. 28 indicates these features.

The spinner shell is formed of two pieces. The back half is flanged at its rear and recessed to accommodate the three blades. The center baffle is spun from the same piece of sheet dural which forms the back half. To this back half the domed front is flush riveted. A bakelite ring is secured to the center of the support baffle by a riveted channel ring.

The flange on the rear shell half has a series of holes punched in it which coincide with the stud projections of the diaphragm and 21 groove-necked studs which align with the locking ring holes.

When the shell of the spinner is aligned with the proper holes in the diaphragm and pushed into place, the groove-necked studs project through the locking ring holes in the diaphragm and the stud projections of the diaphragm project through the holes of the shell flange, while the bakelite ring in the shell baffle fits snugly over the shelf cut on the front of the hub.

Now a flat, forked key is inserted in a slot on the periphery of the diaphragm and moved in lever fashion so that the narrow part of the keyhole slot in the locking ring is rotated into engagement with the grooved neck of the shell studs; stud projections prevent any axial movement of the shell in relation to the diaphragm and the bakelite ring supports the shell in front.

This spinner is, of course, a rigidly attached type, but could easily be redesigned into a cushion-mounted type if the vibration characteristics of a particular installation dictated such action. VDM has built several types of armored spinners, but not in great quantity. It seems that the weight added is not commensurate with the safety obtained, particularly with the VDM wherein the pitch-change mechanism is at the rear of the hub.

Hydraulically operated propellers with frontally exposed mechanisms, such as Hamilton and Junkers, might possibly use armored spinners to advantage, although experience

concluded on page 392

TIRE and TRACK FLOTATION of Military Vehicles

by Lt.-Col. J. E. ENGLER
Ordnance Desert Proving Ground
Camp Seeley, Calif.

SAND, mud, and snow can present obstacles to military vehicles which are almost as tough to negotiate as manmade traps. With proper vehicle design these former obstacles can be made to become insignificant.

Cross-country operation is affected by two vehicle characteristics, flotation and traction. What is adequate flotation on one terrain is inadequate on another, and this same statement can be applied to traction. With track-laying vehicles, these factors are rather absolute, but with wheeled vehicles, inflation and tire stiffness may be important.

One other important thing affecting wheeled vehicles is the number of wheels which are driving. Wherever the coefficient of traction is low it is imperative that every wheel contacting the ground be able to contribute its share of forward tractive effort.

All these things add up to say that the only connection between the engine and the forward movement of a vehicle is the tires or tracks on which that vehicle rests, and it is for this reason that the principles of flotation and traction must be adhered to or cross-country operation may be impossible.

CROSS-COUNTRY operation is something which not only armies must master but also any outfit which moves overland without the aid of roads or other surface improvements. Sand, mud, snow, and many combinations of these, as shown in Figs. 1 to 6, present probably the most severe obstacles; and it is on these types of terrain that flotation and traction must be given first consideration.

To some this may have little significance; to others who have recognized that their equipment is inadequate for cross-country operation it may arouse a great deal of interest; and to those who *must* get vehicles through such areas, the right kind of equipment is the difference between failure and success. Here's an example of what I mean:

Consider first a 6x6 truck similar to our 4-ton cargo type as it would be designed for normal road operation. See Fig. 7. Assume that the single tires are loaded to their full capacity and the vehicle weighs about 24,000 lb. Experience has shown us that in soft sand and at full inflation, this truck would be sluggish and barely able to move itself about. It could not climb any appreciable grade or tow any load back of it. If we reduce the tire inflations to 40% of what they were originally inflated, there would be a marked improvement in its sand performance. The truck can now pull a towed load of 3000 lb or climb a 12% grade. If we go further and apply tires to this truck whose rated loads are *twice* those actually used here, then the vehicle can pull 6000 lb or climb a 26% slope, which is sufficient performance to let it travel throughout almost all dune areas.

As a matter of interest, just how will this vehicle per-

[This paper was presented at the SAE War Engineering-Annual Meeting, Detroit, Mich., Jan. 11, 1944.]

■ Fig. 1 (left) – Medium tank in sand dune area

■ Fig. 2 (right) – Gasoline truck can also be equipped to negotiate sand dunes

■ Fig. 3 (left) – Typical mud where traction and ground clearance mean everything

■ Fig. 4 (right) – Field with bottomless mud underneath grass mat

form in mud or snow? Well, the effect of the reduced inflation and larger tires probably will *not* be noticeable except in certain specific conditions and the reason for this is somewhat involved so we'll come back to it later. Right now let's see why the vehicle just described performs as it does.

In order to get our terms straight let's start at the beginning. Flotation can be defined as the ability of a unit to stay on, or near, the surface of a particular type of terrain. What is good flotation in sand is of little significance in either mud or snow. Traction is the ability of a vehicle to cling to a surface or substance so that by clinging, it can move itself forward. Here again traction on one terrain may be useless or inadequate on another.

First let's describe sand – the toughest kind of sand which exists anywhere. Fundamentally it is similar to minute ball-bearings; when dry the particles are easily shoved apart, but when damp they may tend to cling together, forming a somewhat solid surface. The presence of silt and dust will produce the same binding effect as moisture. A tire or track in passing through sand will displace some of it to each side and compact slightly that portion directly underneath it – support is received both vertically and laterally. The extent of this displacement and the degree of compacting will depend primarily upon the track or tire dimensions and the load carried and secondarily upon the inflation. See Fig. 8. If the effective ground pressure is high, the tire or track will penetrate the sand to such a depth that even though the engine can spin the wheels, forward movement may be almost impossible. In such cases, additional engine power will be of no help because the tires are already slipping in the sand.

Inflation is important in that a lower inflation than that recommended for highway service effectively produces a larger, more flexible tire (Fig. 8) and instead of acting as a wedge the tread is allowed to be pushed inward, trapping the sand. The effect of inflation is reflected not only in the improvement of tractive ability but also in reduced rolling resistance. Fig. 9 illustrates how tractive effort and rolling resistance will vary with inflation. Note also that as tire deflection increases due to lower inflation, tractive effort increases rapidly while rolling resistance approaches a minimum, and any further reduction in inflation results in a great deal of work being done on the tire itself, causing rolling resistance to rise sharply. This variation in tractive effort and rolling resistance with change in inflation is true for all sizes of tires.

In soft sand, the tread design is important only as it affects tire flexibility. Difference in performance between two designs can be offset by adjusting inflation to provide the same flexibility.

Before we get to the practical application of the principles described above, let's discuss one other factor which is fully as important as anything mentioned up to now, and that is the necessity for all-wheel drive. There is no substitute for this, especially where maximum loads are to be carried cross-country. Here is a typical example:

Consider a large earth hauling vehicle of the tractor-trailer type usually found around construction jobs, but in this instance carrying a more practical load. See Fig. 10. This particular vehicle I have in mind, when fully loaded, weighs 96,000 lb. In soft level sand this tractor-trailer unit can barely move about. It will stall on the slightest grades, and is at all times right on the verge of being stuck. The 200-hp engine which it contains is sufficient to spin the front driving wheels at all times. Consider this same vehicle as having part of that 200 hp fed to the trailer axle so that now the unit is no longer a tractor-trailer, but is an all-wheel drive vehicle. See Fig. 11. Under these new conditions in the same soft level sand as before, this vehicle will not only move itself about with ease but will tow an additional trailer load of 17,000 lb or climb at least a 17% sand grade.

Let's review this: We have taken a 4x2 vehicle which is helpless in soft level sand and by converting it into a 4x4

■ Fig. 5 (left) – Rice paddies are difficult but sometimes not impassible for a light tank

■ Fig. 6 (right) – Half tracks find snow-covered hills require maximum traction and ground clearance

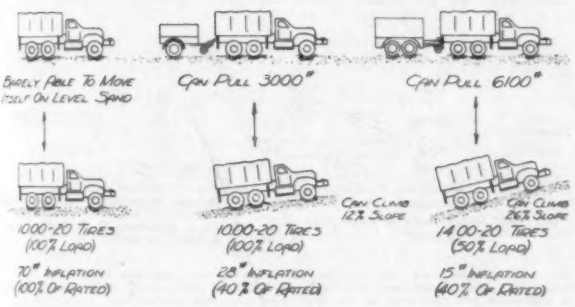
type we have increased its performance by several hundred per cent, all with no increase in total engine horsepower required. Why? Well, the tires are the only means of utilizing the power of the engine to produce forward movement of the vehicle. Engine power greater than the tires can effectively transmit will not be used unless that power can be distributed to more driving axles. This principle is just as true in mud or snow as it is in sand – it is true for all cross-country operation.

As to just how the 4x4 vehicle just described can be so far superior to the 4x2 type, here is the arithmetic explaining it (Fig. 12): The entire vehicle has a rolling resistance of 11,200 lb or 5600 lb for each axle. If we imagine the frame split in the center we will see that the front driving tires are pulling not only the measured 3000 lb drawbar pull but also the rear trailer tires, which have a rolling resistance of 5600 lb. Actually, then, the front driving tires are producing an effective tractive effort of 8600 lb. If we supply drive to the rear axle, making the vehicle a 4x4, then the whole unit can pull 17,200 lb or climb at least a 17% sand grade – it's as simple as that. Of course, in actual service, the shift of load in climbing grades will offset these figures slightly.

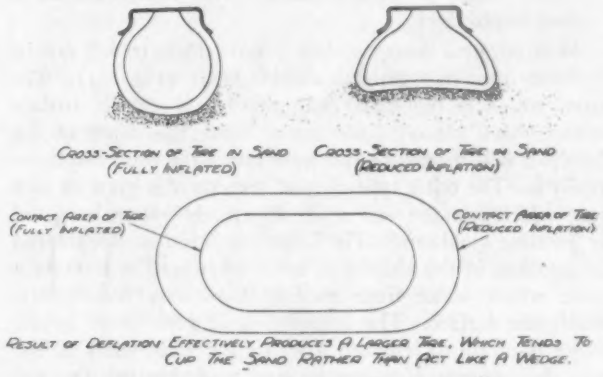
Up to now we've referred mostly to tires – what about tracks? In sand their performance is governed almost 100% by the degree to which they are loaded. We can't deflate them or easily replace them with an oversize as can be done with tires.

On hard surface the vehicle weight is supported on that portion of the track directly beneath the bogie wheels. In sand, mud, or other soft terrains, the weight is distributed throughout the track length. See Fig. 13. Experience has shown that when ground pressures are kept to 12 psi or lower, the vehicle can negotiate almost any sand area encountered. However, this assumes that the load is evenly distributed on the tracks so that when moving forward the back half does not carry most of the weight, in which case ground pressures in that portion would rise considerably and the vehicle would dig in rather than move forward. To keep from throwing tracks in sand is another problem and one which we won't discuss here.

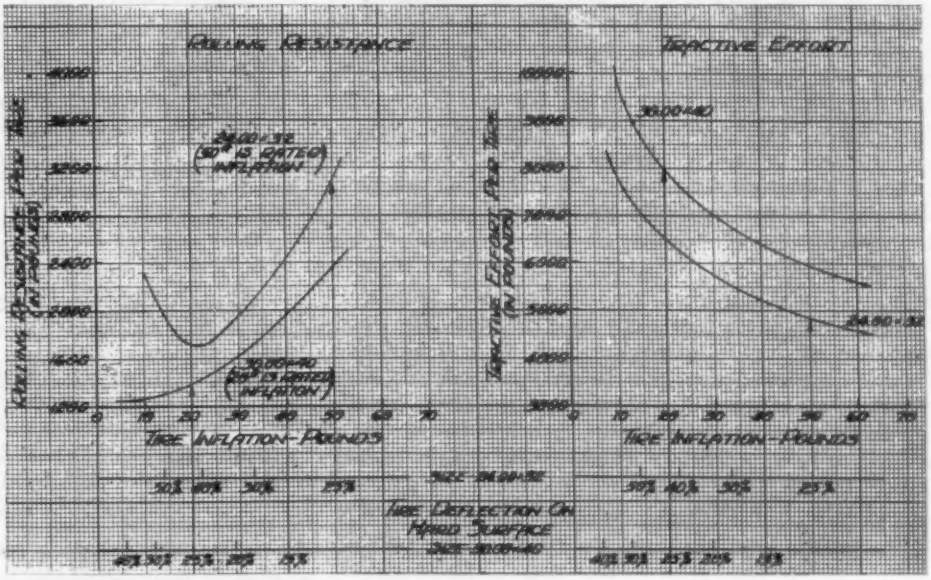
Mud differs from sand in that it resembles a liquid. It is compressible only in the sense that water can be squeezed out of it. A tire or track in going through mud will either


BARELY ABLE TO MOVE ITSELF ON LEVEL SAND
CAN PULL 3000"
CAN PULL 6100"
1000-20 TIRES (100% LOAD)
70" INFLATION (100% OF RATED)
CAN CLIMB 12% SLOPE
1000-20 TIRES (100% LOAD)
28" INFLATION (40% OF RATED)
CAN CLIMB 26% SLOPE
1400-20 TIRES (50% LOAD)
15" INFLATION (40% OF RATED)

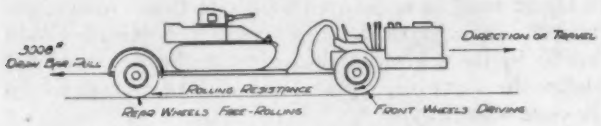
■ Fig. 7 – Applying flotation to a 6x6 truck for operation in soft sand – gross vehicle weight: 24,000 lb


CROSS-SECTION OF TIRE IN SAND (FULLY INFLATED)
CROSS-SECTION OF TIRE IN SAND (REDUCED INFLATION)
CONTACT AREA OF TIRE (FULLY INFLATED)
CONTACT AREA OF TIRE (REDUCED INFLATION)
RESULT OF DEFLATION EFFECTIVELY PRODUCES A LARGER TIRE, WHICH TENDS TO CUP THE SAND RATHER THAN ACT LIKE A WEDGE.

■ Fig. 8 – Action of tires in sand



■ Fig. 9 – Effect of tire inflation in soft sand – load per tire: 23,000 lb



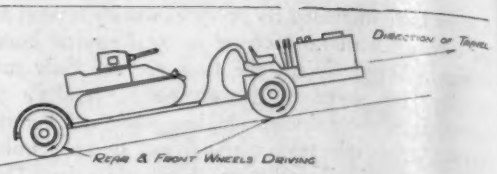
Tire Size 24.00×32 | Vehicle Gross Weight 96,000# (Distributed Evenly on All Tires) | Inflation 50#

Measured Rolling Resistance of Loaded Unit 11,200#

Performance

Above Vehicle Able to Move About on Level Sand and Exert a Drawbar Pull of 3000#, But is Unable to Climb any Appreciable Grade.

■ Fig. 10 – 4x2 type vehicle in soft sand



Tire Size 24.00×32 | Vehicle Gross Weight 96,000# (Distributed Evenly on All Tires) | Inflation 50#

Performance

Above Vehicle Able to Climb a 17% Grade in Sand or Exert a Drawbar Pull of 17,000# on Level Sand.
If Inflation is Reduced to 20# in All Tires, The Maximum Drawbar Pull Will Be Increased to 23,000#.

■ Fig. 11 – 4x4 type vehicle in soft sand

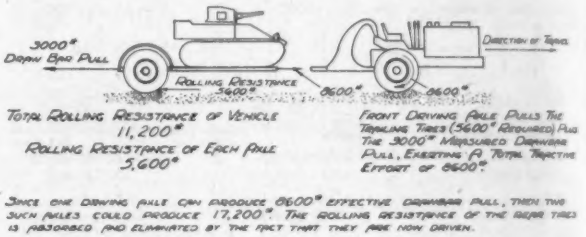
sink down until it reaches a solid base or, if the mud is quite thick, sink down until it has gained sufficient additional contact area to support it. Compared with sand, mud is very easy to displace, so easy in fact that a tire, even with much reduced inflation, will maintain its rounded shape and gain none of the benefits of deflection. As a result, adequate flotation in mud requires much less ground pressures than are necessary for sand. These ground pressures are so much lower that to design a wheeled cargo or combat vehicle with ideal flotation is almost impossible.

Mud covered areas, as they affect vehicle travel, can be divided into two general classifications (Fig. 14). The first, which is the most common, has a muddy surface with a hard subsoil underneath. Here the depth of the hardpan will determine just how severe the mud condition really is. The other type of mud consists of a grass or root mat which rests on top of an apparently bottomless mud or possible quicksand. The latter condition is encountered in marshes, in the vicinity of some lakes, and in mountain areas where water from melting snow may flow underneath the surface. The negotiation of such areas by vehicles is of course affected a lot by the consistency of the soil, the amount of water present, and whether the soil tends to settle and pack or to stay in suspension.

In connection with vehicle suspension design, some factors are more important than others, depending upon the type of mud. Let's consider some specific examples: In shallow mud with a hard subsoil just underneath the surface, traction is the most important thing, for without it the vehicle cannot move. You may expect that it can be secured by a rugged tread design, but this is only true sometimes. If the tread pattern fills with mud it is equivalent to none at all. Chains, grousers, or other traction devices must then be resorted to, and in almost every instance these will satisfy the traction requirement.

Adequate ground clearance is the next important factor, and here again what is adequate or necessary depends upon the mud condition. If it is sandy soil or contains decayed vegetation, it is likely to clean easily and cause less drag on the tires or tracks. Under such conditions a vehicle can sink deeper into the mud and still pull itself forward, assuming it has traction. If the soil is of the gumbo type, being very adhesive, then it is possible for a vehicle to be stalled before its underparts actually contact mud. In both cases, greater flotation would tend to lessen slightly the penetration and so would be desirable.

On terrain which has a relatively thin layer of sod on top with almost a bottomless mud underneath, flotation is all important. It makes no difference how much traction or ground clearance a vehicle has, if it can't stay on the surface it can't move. And just how much flotation is needed to let the vehicle stay on the surface? That is indeed a problem, because the thickness of the mat and the loading per square inch which it will support will vary all the way from a solid substance to something little better than lily pads on the surface of a lake. If we take the practical viewpoint, we would probably decide that any vehicle which could go where a man could walk would undoubtedly be damn good and have plenty of flotation.



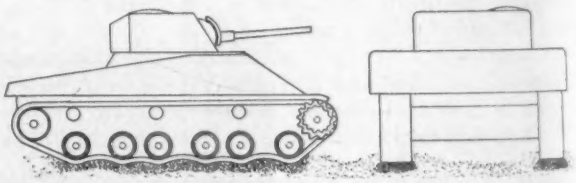
Total Rolling Resistance of Vehicle 11,200#

Rolling Resistance of Each Axle 5,600#

Front Driving Axle Pulls the Trailing Tires (5600# Required) Plus the 3000# Measured Drawbar Pull, Exerting a Total Tractive Effort of 8600#.

Since one driving axle can produce 8600# effective drawbar pull, then two such axles could produce 17,200#. The rolling resistance of the rear tires is absorbed and eliminated by the fact that they are now driven.

■ Fig. 12 – Why a 4x4 vehicle is superior to a 4x2 vehicle in soft sand



■ Fig. 13 – Action of tracks in sand – load distributed throughout track length (exaggerated condition)

Muddy Surface With Hard Sub-Soil Just Underneath

Muddy Surface With Hard Sub-Soil At Greater Depth

Grass Or Root Mat Covering Bottomless Soft Mud

Vehicles Stalled In These Areas Will Be Due To

1 Lack of Traction
2 Lack of Ground Clearance

1 Lack of Ground Clearance
2 Lack of Flotation, Especially Where the Hard Sub-Soil is More than a Foot Below the Surface
3 Lack of Traction

1 Lack of Flotation

■ Fig. 14 – Different types of mud areas as affecting vehicle travel

Earlier in this discussion I mentioned that deflation of tires would have little effect in improving the mud ability of a truck. There is one exception to this rule and it is in such an area as we've just discussed, where staying on top of a supporting mat is all important. Here, as in sand, deflation will effectively produce a larger, more flexible tire and in some places, a greater distribution of the load will mean the difference between go and no-go. However, even with low ground pressures it is possible for a vehicle to sink in due to its gross weight being greater than the mat will support.

Snow fundamentally differs from both sand and mud in that it is compressible. When traversed by either tires or tracks it is compressed rather than displaced or pushed aside. The amount of compression (or penetration) will depend upon the load and upon the consistency of the snow; and this may vary from little more than air to almost a solid. A tire or track will penetrate sufficiently far to compact a bearing surface capable of supporting the load. As a general rule, penetration to a depth much greater than the ground clearance of the vehicle will probably stall the vehicle. Since flotation on snow requires a ground pressure of 2 psi or less and these are almost impractical for anything except very light weight vehicles, it follows that as high ground clearance as practical is desirable.

Traction in snow is just as essential as it is in mud, but is more variable. An aggressive design on both tracks and tires may give excellent snow performance when the temperature is below 0 F. At temperatures nearer freezing their tractive qualities may appear almost nil for the reason that, under pressure, the snow will tend to form ice and the rubber tracks and tires cannot bite into ice although they could grip the snow. Since such temperature changes must be expected in all regions where snow exists, the most logical means of ensuring traction is to provide some device which will cut into the ice. In the case of tires, chains can be used, but for tracks the problem is more difficult and is far from settled.

At this stage we've touched on the major factors affecting vehicle travel in sand, mud, and snow, and now let's summarize. Satisfactory negotiation of these areas requires certain vehicle characteristics which can be grouped in three classifications: (1) flotation, (2) traction, and (3) all wheels driving (track vehicles not handicapped in this respect). As pointed out neither of the first two terms are absolute, what is satisfactory for one condition is inadequate for another and if we design for the worst condition, we become impractical.

But regarding the third characteristic, the necessity for all-wheel drive cannot be overstressed as there can be no halfway mark. It should be remembered that on any type of terrain where the coefficient of traction is low or where the tires penetrate the surface itself, vehicle performance is strictly limited by what the tires can convert to effective tractive effort. For example, a 7:50-20 tire in soft sand when loaded to 2500 lb at 55 psi inflation could contribute absolutely nothing to the forward movement of the vehicle because it would penetrate too far into the sand. A reduction in inflation would help and allow it to put out possibly 300 lb in forward tractive effort. Consider the same tire in mud but this time with almost no penetration. Here the coefficient of traction would govern and the tire might pull 200 to 1000 lb. On pavement this tire could contribute 1800 to 2200 lb of tractive effort. What I'm driving at is that the effective tractive effort of driving wheels is so materially reduced in any off-the-road operation, regardless of how much engine power there is available, that the climbing of any grade or the towing of any trail load may be impossible unless every tire contacting the ground contributes its share of tractive effort. In addition to reduced tractive effort of driving wheels, trailing wheels may have rolling resistances as high as 40% of the load carried or even greater.

Probably at this point there is a big question in the reader's mind as to how we can make logical use of this information, and so in this connection I have prepared a few short rules which if followed will go a long way toward removing the obstacles to cross-country operation.

Concerning wheeled vehicles:

1. For satisfactory sand performance, choose a tire size whose rated load for highway operation is at least twice that which it will be required to carry in service. Inflation can be reduced to permit negotiation of sand dune areas if required. If a larger tire than just described is selected, sand performance will be improved to the extent that less deflation will be necessary. The tire should have a flexible tread design, with a minimum number of plies, and should be used as a single. If used as a dual its sand performance will be reduced 25%.

2. For generally satisfactory mud and snow performance, traction and ground clearance are of primary importance. Tires selected should therefore have an aggressive tread design and should be relatively large in outside diameter rather than in cross-section. To provide some flotation, choose a tire size whose rated inflation is about 12 psi for highway operation at the service load. Deflation below this rated pressure will give added flotation in some instances. If it is desired to approach more nearly the flotation ideal, the tire would have to be large enough to support the rated load at 2 psi highway inflation. The tire itself should be used as a single; if used as a dual, performance will be reduced.

3. All-wheel drive is necessary on any vehicle designed for sand, mud, or snow. If not available, additional flotation for all wheels, additional traction for the driving wheels, and in some cases added ground clearance, will partially compensate for this disadvantage.

Concerning track-laying vehicles:

1. For satisfactory sand performance, provide sufficient track area to ensure a ground loading of not more than 12 psi. The load should be evenly distributed throughout the track length and vehicle center of gravity kept low. Traction qualities are unimportant.

2. For generally satisfactory mud and snow performance, traction and ground clearance are again of first importance and maximum traction should be provided. A track loading equivalent to that satisfactory for sand will also provide some flotation here, but for ideal flotation the track area should be sufficient to ensure a ground loading of about 2 psi. Here again, the load should be evenly distributed throughout the track length and vehicle center of gravity kept low.

And, in conclusion, regardless of what your reaction is, it is common sense to recognize that the tires or tracks on a vehicle are the only connection between the engine and the actual movement of that vehicle and that the principles of flotation and traction must be followed or cross-country operation will be impossible.

A Correlation of the TEMPERATURE—V/ Gasolines with

by E. W. ALDRICH[1],
E. M. BARBER[2],
and A. E. ROBERTSON[3]

VAPOR lock occurs when fuel vapor forms in the fuel system of a vehicle to such an extent that it disturbs the flow of liquid gasoline enough to interfere with normal engine operation.

From past experience it is known that fuel systems vary in their ability to handle vapor and that gasolines vary in their tendency to form vapor. Thus, in connection with the occurrence of vapor lock it is important to know something about the vapor-forming characteristics of gasolines; the vapor-handling characteristics of vehicle fuel systems, and the fuel system conditions that tend to promote vaporization of the gasoline.

The present report presents the results of an investigation of the vapor-forming characteristics of gasolines in the form of a correlation whereby vapor-forming characteristics can be estimated from the customary gasoline inspection test data. The results are applicable in considering vapor-lock test data when it is important to know the vapor-forming characteristics of the gasolines employed in the test and, in particular, when it is necessary to apply vapor-lock test results to fuels other than those employed directly in the testing.

The information used in this report has been obtained from the files of The Texas Co., Standard Oil Development Co., and General Motors Research Laboratories Division. The information assembled from these sources consists of a description of the experimental methods employed by each of the laboratories to measure vaporization characteristics of gasolines and actual measurements of the vaporization characteristics of 208 gasoline samples for which inspection data in the form of Reid vapor pressure and ASTM distillation test data were available. These samples cover a range of Reid vapor pressure values from 2 to 19 psi and a range of temperatures from 130 to 265 F at the 50% evaporated point on the ASTM distillation.

■ Experimental Determination of Curves

The apparatus employed by the three laboratories is quite similar in principle, though minor differences in detail and method of operation exist and appear to cause slight differences in results.

Basically, the apparatus consists of a graduated burette, immersed in a bath so that it can be adjusted to the desired temperature. The bottom of the burette is connected to a mercury manometer and leveling tube. In the operation of the apparatus a sample of liquid gasoline of known volume is introduced into the burette, the pressure on the burette being maintained at any desired value by adjusting the leveling tube. By suitable adjustment of the bath temperature and of the leveling tube, the gasoline sample is caused to vaporize. Readings are taken to determine the temperature and the volume of vapor contained in the burette when equilibrium has been attained after each increment of vaporization. For the determination of each sample, readings are taken at several points covering the range of vaporization that is of interest.

The experimental result is presented in the form of a curve relating temperature to the volume of vapor formed per unit volume of gasoline sample charged to the apparatus and such a curve is designated as a temperature V/L curve.

■ Comparability of Experimental Data

The data used in development of the correlation originate from three independent sources and test apparatus

[1] National Bureau of Standards.
[2] The Texas Co.
[3] Standard Oil Development Co.

[This paper was presented at the SAE War Engineering-Annual Meeting, Detroit, Mich., Jan. 14, 1944.]

THE results of an investigation of the vapor-forming characteristics of gasoline are here presented in the form of a correlation whereby vapor-forming characteristics can be estimated from the customary gasoline inspection test data.

These results are applicable in considering vapor-lock test data when it is important to know the vapor-forming characteristics of the gasolines employed in the test and, in particular, when it is necessary to apply vapor-lock test results to fuels other than those directly employed in testing.

THE AUTHORS: E. W. ALDRICH has been employed in the Automotive Powerplants and Lubrication & Liquid Fuels Sections of the National Bureau of Standards since 1928. During this period, Miss Aldrich, a Tulane University graduate, has engaged in studies of and related to fuels and lubricants. E. M. BARBER, supervisor of engineering research for Texas Co., has been working on fuels and lubricants, with special emphasis on aviation gasoline, at the company's Beacon Research Laboratory for the past six years. He is an alumnus of Pennsylvania State College. A. E. ROBERSTON, research chemist for Standard Oil Development Co., joined the company in 1929, the year he graduated from CCNY. Included in his research studies have been analyses of low temperature hydrocarbon gas, vapor pressure methods, and gasoline development and performance. His first technical position was with Brooklyn Union Gas Co. as analytical chemist.

V/L CHARACTERISTICS of Motor Inspection Test Data

(Compiled from a Report of the CRC Coordinating Fuel Research Committee)

Table 1 – Comparison of Samples from Different Sources

Sample Source	Inspection Data				Temperature, F			
	Rvp	10%	20%	50%	At V/L=4	At V/L=10	At V/L=30	At V/L=45
Laboratory A	8.6	145.5	167.5	229.5	145.5	151.0	167.0	176.5
Laboratory C (Group 2 data)	8.4	148.0	174.0	229.0	146.0	153.5	169.5	179.0
Laboratory B (Preferred)	8.9	145.5	171.5	242.0	140.0	147.0	164.5	175.0
Laboratory C (Group 2 data)	8.9	143.0	169.0	227.0	138.5	146.5	163.0	173.0
Laboratory A	8.5	133.5	160.0	227.0	130.5	138.0	158.0	168.0
Laboratory B (Nonpreferred data)	8.5	135.5	163.0	212.0	132.0	139.0	155.0	163.5
Laboratory C (Group 2 data)	8.3	132.0	162.0	223.0	131.5	138.5	154.5	165.5
Laboratory A	9.2	125.0	140.0	194.0	121.5	126.0	139.0	148.5
Laboratory B (Nonpreferred data)	9.2	129.5	142.0	186.0	123.0	127.0	135.0	139.0
Laboratory C (Group 2 data)	9.3	123.0	145.5	204.5	119.5	125.5	139.0	147.5
Laboratory A	10.0	123.0	142.0	210.0	120.0	124.0	138.5	147.0
Laboratory B (Preferred data)	10.0	122.0	139.5	205.5	119.5	124.5	137.0	145.0
Laboratory C (Group 2 data)	9.8	120.0	142.0	205.0	117.0	122.5	138.0	145.0
Laboratory A	10.7	115.5	133.5	198.5	114.5	119.0	134.5	145.0
Laboratory B (Preferred data)	11.0	115.0	136.5	207.5	113.5	119.0	134.0	143.0
Laboratory C (Group 2 data)	10.6	114.0	136.0	201.0	111.0	117.0	130.5	139.5
Laboratory A	11.9	114.5	137.0	205.5	113.0	118.5	134.0	143.5
Laboratory B (Nonpreferred data)	11.7	109.5	134.0	206.5	111.0	116.0	131.0	141.0
Laboratory C (Group 1 data)	12.0	111.5	132.0	214.5	106.0	113.0	130.0	139.5
Laboratory A	13.3	106.5	115.5	151.0	102.0	105.5	116.0	122.0
Laboratory C (Group 1 data)	13.0	107.5	115.0	146.5	96.0	102.0	111.0	116.0
Laboratory A	14.0	103.0	126.5	202.0	98.5	103.5	120.0	130.5
Laboratory B (Preferred data)	14.0	106.0	132.5	214.0	100.5	106.0	122.0	133.0
Laboratory C (Group 1 data)	13.8	109.0	127.0	205.0	95.0	101.0	118.0	124.0

which had not been standardized. Therefore, before investigating the possibility of obtaining a single correlation to fit all of the data, it was essential to have some assurance that, when identical samples were tested, identical results or at least reasonably comparable results would be obtained. For such a purpose it is evidently desirable that all three laboratories should obtain data on identical samples so that the comparability of testing could be judged directly. In the present instance there has not been the opportunity to conduct tests on identical samples. However, some evidence concerning the comparability of the results can be obtained by a comparison of the *V/L*-temperature data on samples selected from the data submitted to have approximately the same Reid vapor pressure and ASTM distillation values. Such a comparison is shown in Table 1, and from an inspection of this table it appears that the results of tests by the separate laboratories are reasonably comparable.

Correlation

Having established that the test results from the three separate sources were reasonably comparable, a correlation was developed for predicting temperature-*V/L* curves from inspection test data. This correlation is presented in Fig. 1, and consists of four graphs which give the numerical data to be used in the following four equations. The equations apply to calculations at 760-mm pressure.

$$T_4 = T_R + \Delta T_4 \quad (1)$$

$$T_{10} = T_4 + \frac{6}{41}(T_{45} - T_4) + \Delta T_{10/30} \quad (2)$$

$$T_{30} = T_4 + \frac{26}{41}(T_{45} - T_4) + \Delta T_{10/30} \quad (3)$$

$$T_{45} = T_{20\%} + \frac{1}{8}(T_{50\%} - T_{10\%}) + \Delta T_{45} \quad (4)$$

Where:

T_R is determined from the Rvp according to the relationship of Fig. 1 (1)

ΔT_4 is determined from the $(T_{20\%} - T_{10\%})$ according to the relationship of Fig. 1 (2)

$\Delta T_{10/30}$ is determined from the $\left(\frac{T_{50\%} - T_{10\%}}{T_{20\%} - T_{10\%}}\right)$ according to the relationship Fig. 1 (4).

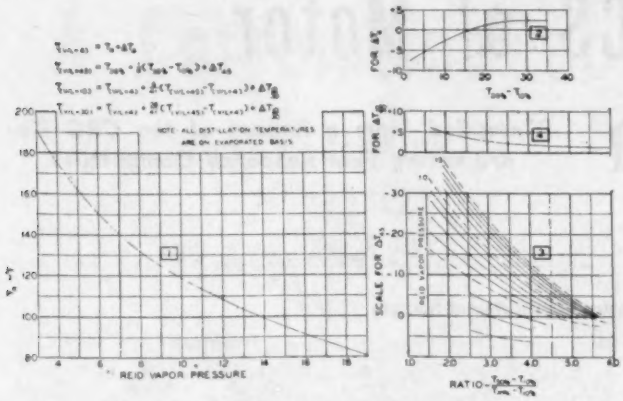
ΔT_{45} is determined from the $\left(\frac{T_{50\%} - T_{10\%}}{T_{20\%} - T_{10\%}}\right)$ and the Rvp according to the relationship of Fig. 1. (3)

Rvp is the Reid vapor pressure.

$T_{10\%}$, $T_{20\%}$, and $T_{50\%}$ refer to the ASTM distillation Temperatures, (F) at 10, 20, and 50% evaporated.

T_4, T_{10}, T_{30}, and T_{45} refer to the Temperature (F) at V/L of 4, 10, 30, and 45, respectively.

A sample calculation to illustrate the method applying the correlation is shown by Table 2.


NOTE: ALL DISTILLATION TEMPERATURES ARE ON EVAPORATED BASIS
REID VAPOR PRESSURE
SCALE FOR ΔT_{10}
REID VAPOR PRESSURE
RATIO $\frac{T_{50\%} - T_{10\%}}{T_{20\%} - T_{10\%}}$

■ Fig. 1 – Chart for estimation of gasoline vapor-forming characteristics from inspection test data

The suitability of the correlation for representing the experimental data is shown by Table 3, which presents a summary of the algebraic and arithmetic average deviations (deviation = calculated temperature — observed temperature) determined for the various groups of data and for the whole group of 208 samples. Table 3 also shows the distribution of the deviations.

From an inspection of Table 3, it is clear that –

a. The level of results is somewhat different in the different data groups. This may result either from lack of standardization in the temperature-V/L determinations or from inaccuracies in the inspection test data.

b. In the various groups of data, the algebraic deviation is a relatively large fraction of the arithmetic deviation. Thus, if the separate groups of data were adjusted for this difference in level, the deviations would be significantly reduced.

c. Even with the differences in level that exist, the overall agreement between the calculated and observed data is very satisfactory, the probable error being approximately 1.7 F and approximately 90% of the samples being correlated to within ±4.5 F.

d. It is highly probable that Rvp and ASTM distillation test inaccuracies contribute substantially to the deviations. This probability is brought out clearly by comparison of the deviations on the company B preferred and nonpreferred groups of data where the preferred data differ from the nonpreferred only in the care exercised in obtaining and checking the Rvp and ASTM distillation tests.

In addition to the wide range of differences in inspection test data, there were groups of samples of unorthodox composition included in the data. These comprised series of blends containing as much as 5% propane, blends containing as much as 50% pentane, depentanized samples brought back to vapor pressure by addition of as much as 20% butanes. For all such extremes of composition, no significant deviations from the correlation were noted.[4]

■ Correction for Pressure

Equations 1 to 4 given in the foregoing section permit

[4] A limited amount of data on samples having 90% points below 225 F and relatively high 50% points indicate that with this type of gasoline exceptional negative deviations may result at $V/L = 45$.

Table 2 – Sample Calculation of Temperature – V/L Curve from Gasoline Inspection Data

A. Inspection Data on Sample

Rvp = 10.4 psi

$T_{10\%}$ = 143.0 F

$T_{20\%}$ = 181.0 F

$T_{50\%}$ = 234.5 F

B. Derived Values from Inspection Data

$T_{20\%} - T_{10\%} = 38.0$

$T_{50\%} - T_{10\%} = 91.5$

$\frac{1}{8}(T_{50\%} - T_{10\%}) = 11.5$

$\frac{T_{50\%} - T_{10\%}}{T_{20\%} - T_{10\%}} = 2.41$

T_R [Fig. 1 (1)] = 116.0

ΔT_4 [Fig. 1 (2)] = +2.5

$\Delta T_{10/20}$ [Fig. 1 (4)] = +4.0

ΔT_{45} [Fig. 1 (3)] = −23.0

C. Calculated Temperature - V/L Data

Equation 1 $T_{(V/L\,=\,4)} = 116.0 + 2.5 = 118.5$ F

Equation 2 $T_{(V/L\,=\,10)} = 118.5 + \frac{6}{41}(50.5) + 4.0 = 130.0$ F

Equation 3 $T_{(V/L\,=\,20)} = 118.5 + \frac{26}{41}(50.5) + 4.0 = 154.5$ F

Equation 4 $T_{(V/L\,=\,45)} = 181.0 + 11.5 - 23.0 = 169.0$ F

D. Experimental Temperature - V/L Data

$T_{(V/L\,=\,4)} = 116.5$ F

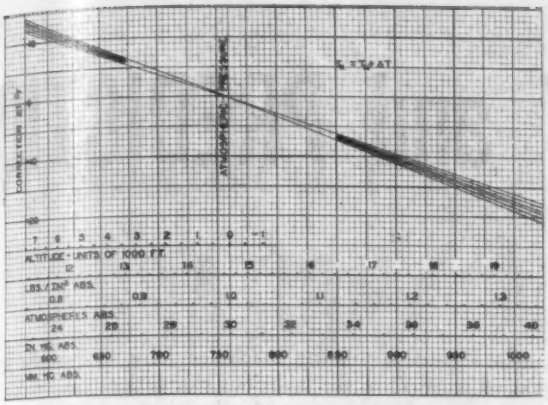
$T_{(V/L\,=\,10)} = 127.5$ F

$T_{(V/L\,=\,20)} = 154.5$ F

$T_{(V/L\,=\,45)} = 170.0$ F

Table 3 – Summary of Deviations

Data Source	Company A				Company B (Preferred Data)				Company B (Nonpreferred data)				Company C Data Group 1				Company C Data Group 2				All Data			
V/L Samples	4	10	30	45	4	10	30	45	4	10	30	45	4	10	30	45	4	10	30	45	4	10	30	45
Total Samples			49				23				76				35				25		208	207	207	208
Algebraic Deviation	+0.11	+0.50	−1.21	−1.21	+0.4	+1.0	+0.6	+0.3	−0.9	−0.6	−0.2	+1.0	+2.5	+2.4	+2.2	+2.9	+0.6	−0.1	−0.8	−1.9	+0.2	+0.4	0.0	+0.4
Arithmetic Deviation	2.1	2.4	2.0	2.1	1.6	1.6	1.3	2.0	2.7	2.4	1.8	2.6	2.5	2.7	2.9	4.1	2.2	2.2	2.0	2.4	2.3	2.3	2.0	2.6
Distribution of Deviations, %																								
0.0- 0.5	26.5	12.2	28.5	16.3	21.7	21.7	26.1	13.0	19.7	16.1	18.7	15.8	14.3	22.8	14.3	11.4	16.0	12.0	24.0	16.0	20.2	16.4	21.8	14.9
0.0- 1.5	46.9	36.7	46.9	34.7	60.9	60.9	65.2	47.8	46.0	42.7	52.1	38.2	54.3	37.2	37.2	25.7	48.0	52.0	60.0	44.0	49.5	43.5	50.5	37.0
0.0- 2.5	63.3	61.2	63.3	59.1	78.3	78.3	100.0	78.2	60.6	66.7	74.7	63.2	65.8	60.1	51.4	48.6	68.0	76.0	72.0	64.0	65.0	67.0	70.5	61.5
0.0- 3.5	79.6	77.5	79.6	73.4	100.0	100.0		82.6	69.0	74.8	90.8	77.7	74.4	71.5	71.5	60.1	84.0	88.0	80.0	88.0	73.0	79.0	84.5	73.0
0.0- 4.5	95.9	97.9	95.9	83.7				100.0	80.5	86.7	96.1	87.0	85.9	83.0	80.1	65.8	96.0	92.0	92.0	92.0	89.0	90.8	93.0	84.5
0.0- 5.5	97.9	97.9	97.9	95.9					89.6	97.5	98.8	94.9	91.5	88.7	82.9	77.2	100.0	92.0	96.0	100.0	94.0	95.5	95.6	93.3
0.0-> 5.5	100.0	100.0	100.0	100.0					100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0		100.0	100.0		100.0	100.0	100.0	100.0



■ Fig. 2 – Correction of boiling point of gasolines with change in pressure

estimation of the temperature-V/L curve of a gasoline at normal atmospheric pressure. In many instances it may become desirable to determine the corresponding curve for higher or lower pressures.

Given a temperature-V/L curve for normal atmospheric pressure, the corresponding curve for some other pressure can be obtained by making the following adjustments:

a. Adjust the temperature T_n by an amount dependent on the variation of boiling point with pressure to a new value T_x.

b. Adjust the V/L ratio $(V/L)_n$ by the following equation:

$$(V/L)_x = (V/L)_n \cdot \frac{P_n}{P_x} \cdot \frac{P_x}{T_n} \qquad (5)$$

Where:

P = Absolute pressure
T = Absolute temperature
$_n$ denotes the normal atmospheric pressure data
$_x$ denotes the new pressure condition

Fig. 2 may be used to estimate the boiling temperature T_x when T_n is known.

The adjustment for pressure is conveniently made by making a separate adjustment on each point and then plotting a new curve through the adjusted points.

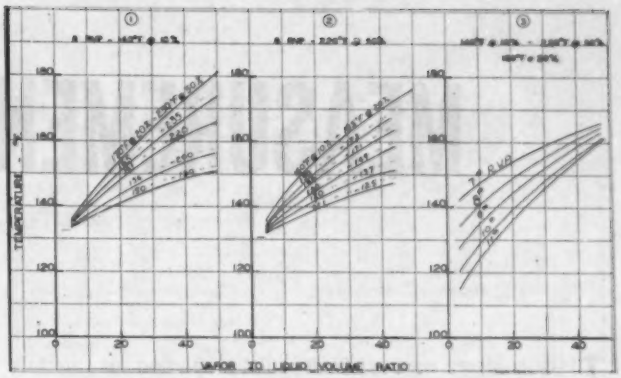
The pressure correction has received experimental test and may be considered to be of satisfactory accuracy.

■ Curves for Typical Gasolines

The effect on temperature-V/L curves of independently varying Reid vapor pressure, 10%, and 50% points is shown in Fig. 3 for several typical gasolines. Graph 1 of Fig. 3 shows variations of the temperature-V/L curves for four gasolines, all of which have 8-psi Rvp and 140 F at 10% evaporation. Graph 2 of Fig. 3 shows the variations of the temperature-V/L curves for five gasolines, all of which have 8-psi Rvp and 220 F at 50% evaporation. Graph 3 of Fig. 3 shows the variations of the temperature-V/L curves for five gasolines, all of which have 140, 165, and 220 F at 10%, 20%, and 50% evaporation, respectively. These three graphs represent in each case about the extreme commercial range that is likely to occur with the fixed variables that are indicated.

■ Conclusions

A correlation is presented from which it is possible to estimate temperature-V/L curves of gasolines from Rvp and ASTM distillation data. Within broad limits as to gasoline volatility and composition the accuracy of the correlation is compatible with the accuracy of the individual tests on which it is based. The results of the study show substantial variations in the amount of vapor formed at a given temperature, even with fuels having one of the usual front end volatility control points, such as Rvp, fixed.



■ Fig. 3 – Vapor-forming characteristics of several gasolines

QUALITY CONTROL

continued from p. 347

2. By describing the origin and form of our process control specification.

3. By explaining briefly our method of organization for process control enforcement.

We then used various types of industrial applications to illustrate how this control could be specified and used.

Any of the various industries within the Ford Motor Co. would have served as well. We might have used examples from saw mills to glass plants or from our airplane-engine factories to the bomber plant.

Lastly, we described methods of test on finished parts and assemblies of finished articles to determine the effectiveness of our control procedure and the quality of material, parts, or articles.

For obvious reasons, we cannot at this time, show comparative quality and production figures, but we can give you our assurance that the improved results we are obtaining are very encouraging. Production has been increased, scrap decreased, and quality improved.

If our attempt here in writing some of our thoughts on a method of improving quality through a better control of engineering materials during manufacturing and processing, is of help to others in the war effort, the Ford Motor Co. and the authors will be deeply gratified.

■ Fig. 45 – Testing finished vehicle

MEASUREMENT and PREDICTION of

THE subject, measurement and prediction of aircraft vibration, indicates that the paper deals with both experimental and analytical phases of the problem. With the wide variety of vibration measuring equipment available, a number of persons have been interested in having the data compiled so that it would be possible to determine the most suitable equipment for a specific problem. An attempt at this will be made in the first part of the paper.

Since the subjects of engine and propeller vibration have been covered by previous authors to a much greater extent than aircraft vibration, methods of studying and predicting the coupled modes and frequencies of aircraft structures will be discussed, with examples showing the correlation that has been obtained between experimental and analytical data. It is becoming more evident that uncoupled modes of vibration of aircraft structures may not be representative enough to conduct a satisfactory flutter analysis of an aircraft; therefore, methods of determining coupled modes of vibration have been studied.

All of the experimental measurements and theoretical computations on aircraft vibration have one basic objective; that is, to establish the maximum safe air speed of the aircraft. In reaching this objective, the following procedures have been and are being employed:

(a) Empirical approach – Incorporate stiffness, mass, and mass balance values in the aircraft known or judged to be adequate for the intended usage of the aircraft.

(b) Flight testing – Utilizing the procedure of (a) may not be sufficient for unconventional or high-performance aircraft. In such a case it may be necessary to install vibration equipment in the aircraft and explore the vibration response in flight.

(c) Prediction – The increasing size, cost, and performance of aircraft make it imperative that analytical methods be developed for accurately predicting the flutter speed of aircraft. The methods of Kussner, the classical theory of Dr. T. Theodorsen of the NACA, and the extensive work of Major B. Smilg of the Army, the Bureau of Aeronautics of the Navy, and other researchers permit the determination of the critical speed once the frequencies and modes of vibration are established. The use of equipment described herein permits these frequencies and modes to be established after the aircraft is built. The use of the analytical and tabulating machine method outlined in this paper permits the determination of frequencies and modes in the design stage.

Since available methods for calculating coupled modes of vibration are long and tedious, the Aircraft-Engineering Division initiated a study aimed at devising an additional tool for the engineer to expedite such calculations. After considerable exploratory work, methods of utilizing punched card calculating machines were successfully developed to perform the laborious calculations involved.

After procedures were developed for utilizing the punched card calculating machines for performing routine calculations of aircraft modes of vibration at zero air speed, the machine procedures were then extended to permit computations of aircraft modes of vibration at various air speeds and to solve the basic flutter equations which are involved in the determination of critical flutter speeds.

■ Vibration Equipment

Vibration measuring equipment can be divided into two main classes, mechanical and electronic. The mechanical instruments can be used for certain ground studies where some indication of amplitude or frequency is desired. Generally, they are not as accurate as the electronic instruments but can be used as a rough check. One exception is the rotoscope, which is a mechanical light chopper and can be used to view a relatively large field in daylight if the amplitudes of vibration are large enough to see by eye.

For flight testing and most ground testing, electronic equipment is the best suited, since amplitude, frequency, and phase can be all studied simultaneously. It also permits the study of transient effects.

Electronic equipment for aircraft vibration studies can

BOTH experimental and analytical phases of the aircraft vibration problem are covered by Mr. Critchlow, who has made a thorough study of this increasingly important subject.

As a result of this research work, methods of calculating coupled modes of vibration of aircraft structures have been developed that agree quite closely with measured data.

An important factor in cutting down the time, labor, and expense required in making vibration and flutter studies has been the adaptation of punched card calculating machines to perform the routine calculations.

■ ■ ■

THE AUTHOR: E. FOREST CRITCHLOW entered Government service in 1941 as chief of the flutter and vibration unit of the Aircraft Engineering Division of the Civil Aeronautics Administration. Before that Mr. Critchlow, who is a member of the NACA Subcommittee on Flutter & Vibration, was connected with Lycoming Division and Westinghouse Electric & Mfg. Co., where he was mainly concerned with vibration engineering problems.

[This paper was presented at the SAE National Aeronautic Meeting, New York City, April 5, 1944.]

f AIRCRAFT VIBRATION

by E. FOREST CRITCHLOW
Civil Aeronautics Administration

be divided into four groups according to their usable frequency range:

1(a) 0 to 100 cps – for measuring structural vibration, dynamic loads, static measurements, such as steady stresses, air speed, control surface position indicators, and control force indicators.

1(b) In order to get around the use of modulated or d-c amplifiers, some equipment is designed to work in the frequency range of 1 to 100 cps instead of starting at zero frequency. This greatly reduces the utility of the equipment since only steady-state vibrations can be measured with it.

2. 5 to 1000 cps – for measuring engine and propeller vibration, which usually involves a higher frequency range than structural modes of the aircraft.

3. 30 to 10,000 cps – for soundproofing and general noise measurement.

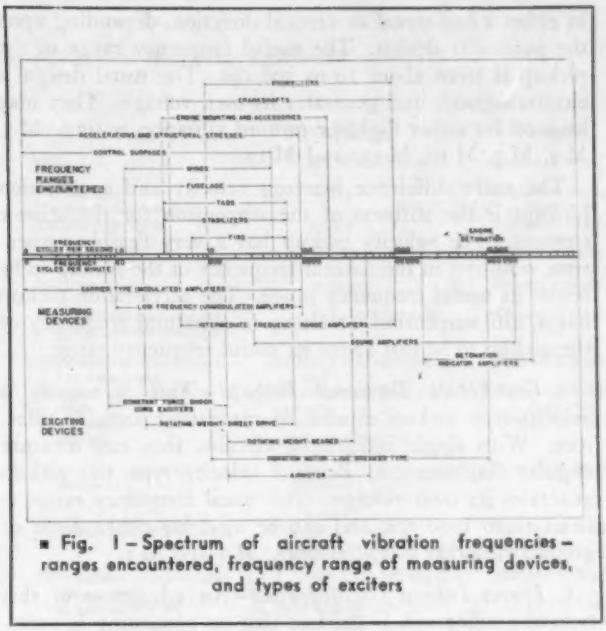
4. 5000 to 15,000 cps – for detonation indicators.

The essential elements of a system used to study vibration is a pickup of some type which is attached to the structure being studied. It may be a unit that responds to strain, position, pressure, acceleration, velocity, displacement, and so on, and may be linear or angular. The output of the pickup can, in some cases, be put directly into a recording or measuring device; however, in most cases it is necessary to put it through an amplifying device to increase the power of the signal in order to operate a recording device.

Some idea of the range and utility of the various types of vibration measuring equipment can be obtained from Fig. 1. This spectrum shows the range wherein various types of aircraft vibrations occur and gives some idea of the equipment that might be used for measuring them.

The most common types of electronic equipment suitable for aircraft vibration studies are listed as follows, showing whether they are suitable for flight or ground testing, and index letter "M" – indicating manufacturers that have available designs. The "key" to the "M"-designation is given in the section immediately following the listing.

1. *Wire strain gages* – These gages may be used for either strain measurements or as elements of special pickups for measuring motions, acceleration, and so on. Their frequency range extends from zero to values as high as



■ Fig. 1 – Spectrum of aircraft vibration frequencies – ranges encountered, frequency range of measuring devices, and types of exciters

50,000 cps. They may be used for both flight and ground measurements, and are available in various sizes and resistances up to 15,000 ohms. Attachment is achieved by cementing and if the gages are used at elevated temperatures such as for engine studies, they can be attached with special high temperature cement. M-1.

2. *Crystal-Type Acceleration Pickups* – This type of pickup is made by using a piezoelectric material which generates a voltage when a bending force is applied to it. When the bending force on the crystal is due to the acceleration acting on the pickup unit, it becomes an acceleration pickup. If the pickup is used with a circuit which incorporates a single or double integrating circuit, velocities or displacements, respectively, can be measured. The piezoelectric material is usually rochelle salts, but pickups using quartz are also available. Their usual frequency range is from 2 to 1000 cps. They are suitable for ground vibration studies and some limited flight measurements. M-2, M-7, and M-15.

3. *Carrier-Type Acceleration Pickups* – The carrier-type pickups are usually variable reluctance legs of a modulated bridge circuit. The high-frequency modulation is necessary to extend the frequency range of the pickup down to zero. They are for measuring acceleration, and, as in the case of the crystal type, can also measure velocities or displacements if single or double integrating circuits are used.

The usual frequency range is from 0 to 100 cps and can be made to cover higher frequencies, but due to the stiffer seismic element necessary, the sensitivity of the pickup is reduced. Either ground or flight testing is practical with this type of unit. Pickup can be mounted in any direction since gravity acceleration has a minor effect on the position of the flexibly mounted mass in the pickup. M-4, M-12, and M-13.

4. *Velocity Pickup* – As indicated by the name, this pickup is capable of measuring velocities, or if used with single integrating circuits will measure displacements. Since this pickup has a flexible type of suspension for the seismic element, it is affected by gravity so it can be used

in either a horizontal or vertical direction, depending upon the particular design. The useful frequency range of the pickup is from about 10 to 500 cps. The usual design is electromagnetic and generates its own voltage. They may be used for either flight or ground vibration testing. M-3, M-4, M-7, M-10, M-13, and M-17.

The main difference between velocity and acceleration pickups is the stiffness of the suspension for the seismic element. The velocity pickup has a very flexible suspension, resulting in the natural frequency of the pickup to be below its useful frequency range. The acceleration pickup has a stiff suspension, resulting in a natural frequency of the pickup to be just above its useful frequency range.

5. *Crankshaft Torsional Pickups* – This is usually a velocity-type pickup capable of measuring angular velocities. With single integrating circuits, they can measure angular displacement. Being a velocity type, the pickup generates its own voltage. The usual frequency range is from 10 to 1000 cps, and can be used for either flight or ground vibration measurements. M-3 and M-4.

6. *Direct Inking Oscillographs* – An advantage of this type of oscillograph is the fact that no processing is necessary to obtain records. It may be used in multichannel recording equipment. The usable frequency range varies from 1 to 60 or 120 cps. While generally employed for ground vibration measurements, it does have limited flight usability. M-2 and M-6.

7. *Photographic Recording Oscillographs* – This type of oscillograph is a multichannel recorder, and can be obtained in various sizes from 4 to 24 channels. The records are usually obtained on photographic paper and are from 2 to 10 in. wide, depending upon design. An amplifier is generally used to drive the oscillograph elements, since the signal from the pickup is too weak to operate it directly. However, one exception is manufactured by M-10, where the pickup operates the oscillograph element directly.

The frequency range and sensitivity is determined by the oscillograph element, which translates the electrical signal into mechanical motion. Different elements can, therefore, be used, depending upon the frequency range being studied. Most designs are suitable for both flight and ground testing. M-4, M-9, M-10, M-13, and M-17.

All the above are conventional recorders with the exception of M-17, which is a 13-channel variable area recorder developed for the Propeller Laboratory, Wright Field, Dayton, Ohio. In this design, 13 channels are recorded on 35-mm moving picture film.

8. *Oscilloscopes* – The oscilloscope is a cathode-ray device for translating the electrical impulse into a visible motion. It is the motion of a stream of electrons bombarding a fluorescent screen, so that the illusion of motion is obtained due to the persistence of vision. The screen can be photographed with an ordinary camera and therefore can be used for getting permanent records in making vibration studies. It is essentially a single element unit, but by using two pickups simultaneously in the horizontal and vertical channels, phase relation and relative amplitude can be determined by studying the Lissajous figures on the screen. The use of an electronic switch with the oscilloscope makes it a multielement unit.

Some designs will go down to about 1 cps, and due to the principle of operation, its high-frequency limit is far beyond the range required for aircraft vibration studies. It is suitable for making ground measurements and has been used for limited flight work. M-5, M-14, and M-17.

9. *Linear Amplifiers* – This type of amplifier does not use a modulating carrier system, and can be used with all types of pickups listed, with exception of the carrier type. They are used where the frequency range being studied does not extend down to include zero.

The design limitation that determines the usable frequency range is not confined to the amplifier alone, but is a combination of pickup, amplifier, and oscillograph (or measuring device).

The usual frequency ranges where this type of amplifier is used is from 1 to 100 cps, 5 to 1000 cps, and 30 to 20,000 cps. They can be used for either flight or ground investigation. M-3, M-4, M-8, M-9, M-13, and M-17.

10. *Carrier Amplifier* – This type of amplifier requires modulation with a high-frequency carrier system. They are used with carrier-type pickups (for example, strain gages, accelerometers, and pressure type) where zero frequency (static) or very low frequencies are being studied. Their usable frequency range is from zero to the upper limit determined by the oscillator (carrier) frequency and can be used for either flight or ground testing. M-4, M-7, M-9, M-12, and M-13.

11. *Vibration Meters* – This type of unit consists usually of a pickup, amplifier, and visual output meter. It is used for measuring amplitudes, velocities, or accelerations, depending upon the circuit arrangements and pickup used. It is useful in making quick amplitude comparisons, but does not determine phase or frequency. The usual frequency is from 2.5 to 750 cps, and it is useful for either flight or ground measurements. M-7, M-8, and M-16.

12. *Soundmeters.* The soundmeter is essentially the same as the vibration meter with the exception that its range covers the audio-frequencies from 25 to 7500 cps, and is used with a microphone instead of vibration pickup. It can be used for flight or ground testing. M-8 and M-17.

13. *Vibration and Sound Analyzers.* These are associate equipment with items 11 and 12 if the frequencies of the components are to be separated. M-8, M-11, and M-17. One exception is the 4-channel harmonic analyzer built by M-17 for the Propeller Laboratory at Wright Field. This unit is capable of simultaneous analysis of 4 channels from the 13-channel variable area recorder. Frequency range is from 5 to 3000 cps. They can be used for either flight or ground testing; however, some of the designs are only suitable for ground measurements.

14. *Mechanical Pickups* – Where a rough and quick determination of amplitude of a vibrating object is desired, certain mechanical pickups are available. They are self contained units where no associated equipment is required, and have some mechanical amplification so as to increase the vibrating amplitudes to readable values. They can be used for ground testing and certain types of flight testing, such as instrument panel roughness. M-7.

15. *Mechanical Frequency Indicators* – These are usually of a vibrating reed type, where the reed frequency is tuned to the frequency of the vibrating object, thus indicating the vibration frequency. They have limited ground and flight usage, but are very small and portable. M-18.

Table 1 – Chart Showing Classification of Vibration Measuring Equipment

Description	Use	Frequency Range, cps	Type	Range	Usable
Wire strain gage	Measurements of stress Special pickups	0 – 50,000+	Wire	±0.005 in. per in.	Ground or flight
Crystal-type acceleration pickups	Measurement of linear acceleration, velocity and displacement	1 – 1000	Piezoelectric Rochelle salts or quartz	±10 g	Ground or limited in flight
Carrier-type acceleration pickups	Measurement of linear acceleration, velocity and displacement	0 – 100	Variable reluctance	±20 g	Ground or flight
Velocity pickup	Measurement of linear velocity and displacement	5 – 1000+	Electromagnetic	±0.25 in.	Ground or flight
Crankshaft torsional pickup	Measurement of angular velocity and displacement	5 – 1000+	Electromagnetic	±10 deg	Ground or flight
Direct inking oscillograph	Multichannel recording up to 25 channels	1 – 60 1 – 120 0 – 100	Pen operated type element, crystal or electromagnetic	±½-in. amplitude	Ground or limited in flight
Photographic recording oscillograph	Multichannel recording 4 to 24 channels	0 – 1000+	Galvanometer-type element, Electromagnetic	Full width of recording paper	Ground or flight
Cathode-ray oscilloscope	Single channel for viewing	1 – very high	Amplifier and cathode-ray tube	Full sweep of cathode-ray screen	Ground or limited in flight
Linear amplifier	Increasing signal from pickup to recorder	1 – 20,000	Resistance-condenser coupled	Typical: Voltage gain 11000/1 0.1 meg in, 500 ohms out	Ground or flight
Carrier amplifier	Increasing signal from pickup to recorder	0 – limit determined by carrier frequency	Modulated with carrier frequency	Typical: $\frac{\text{Amp, d-c}}{\text{V, rms}} = 2.9$	Ground or flight
Vibration meter	Measurement of vibration amplitudes	2.5 – 750 10 – 4000	Pickup, amplifier and output meter	From 16 micro-in. 160 micro-in. per sec, 0.16 in. per sec²	Ground or limited in flight
Sound meter	Measurement of sound	25 – 7500	Microphone, amplifier, and output meter	24 db to 140 db	Ground or flight
Vibration and sound analyzer	Separate frequency components of complex wave	2.5 – 750 25 – 7500	Heterodyne or filter	Used with vibration and soundmeters	Ground or limited in flight
Mechanical pickups	Measurement of vibration amplitudes (visual)	15 – 250	Mechanical or mechanical-optical	0 – 0.015 in. 0 – 0.030 in.	Ground or limited in flight
Mechanical frequency indicators	Measurement of vibration frequencies	10 – 300	Vibrating reed	0.0001 in. and over	Ground or limited in flight
Stroboscopes	Viewing of vibrating objects as slow motion	6 – 10,000	Electronic or mechanical	Limited to visible amplitudes	Ground

16. *Stroboscope.* There are essentially two methods for viewing vibrating objects, to make them appear to stand still, or to move very slowly, in order to study their motion. One is by a flashing light (all other light excluded) so that the object is illuminated at certain intervals. If these intervals are properly regulated, the object will appear to move very slowly or to remain stationary. The other method is to interrupt the light getting to the eye by means of a light chopping shutter. The first type is used in darkness while the second type can be used with a bright light. Photographs can be taken using the first method if desired. This method is usable for ground testing. M-8, M-19, and M-20.

Table 1 summarizes the data on vibration measuring equipment.

The manufacturers indicated by the "M-" are as follows:

M-1 Baldwin Southwark Division, Philadelphia, Pa.
M-2 Brush Development Co., Cleveland, Ohio.
M-3 Commercial Engineering Laboratory, Detroit, Mich.
M-4 Consolidated Engineering Corp., Pasadena, Calif.
M-5 DuMont Laboratory, Inc., Passaic, N. J.
M-6 Electro-Medical Laboratories, Holliston, Mass.
M-7 General Electric Co., Schenectady, N. Y.
M-8 General Radio Co., Cambridge, Mass.
M-9 Hathaway Instrument Co., Denver, Colo.
M-10 Heiland Research Corp., Denver, Colo.
M-11 Hewlett Packard Co., Palo Alto, Calif.
M-12 Glenn L. Martin Co., Baltimore, Md.
M-13 Wm. Miller Corp., Pasadena, Calif.
M-14 RCA Victor Division, Camden, N. J.
M-15 Shure Brothers, Chicago, Ill.
M-16 Televiso Products, Inc., Chicago, Ill.
M-17 Western Electric Co., New York, N. Y.
M-18 Westinghouse Electric & Manufacturing Co., East Pittsburgh, Pa.
M-19 The Boulin Instrument Corp., New York, N. Y.
M-20 A. T. Ashdown, Ltd., London, England.

■ Coupled Modes of Vibration

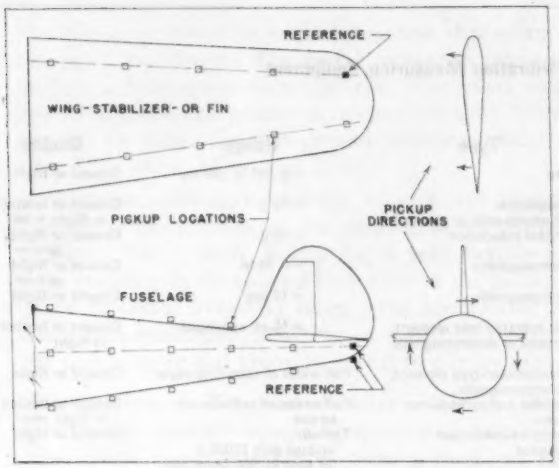
In connection with aircraft structures containing large discontinuities in mass and stiffness properties, it will be found in many cases that simple uncoupled modes of vibration do not represent the structural behavior with the degree of accuracy necessary to be used in conducting a satisfactory flutter analysis. With this in mind, the CAA has been conducting ground vibration tests on a number of aircraft structures to determine the coupled modes of vibration, and investigating the possibilities of predicting these coupled modes of vibration from design data so that a complete flutter analysis might be conducted while the design is still in the drafting room stage.

This, of course, required close correlation between test data and analysis, with a method of analysis that is capable of yielding coupled modes of vibration.

The method of analysis used for predicting the coupled modes of vibration is based on some of Jean Wylie's unpublished notes written while with the Civil Aeronautics Administration, and also work by Lt. M. A. Biot, Bureau of Aeronautics, Navy Department.

The method has been used successfully for computing wing, stabilizer, and fuselage modes of vibration. For wing modes the method has been used for single-, two-, and four-engined airplanes.

Method Used to Obtain Experimental Results – In order to determine the reliability of any method of analysis, it



■ Fig. 2 – Pickup locations for obtaining measured vibration data

was necessary to study these modes of vibration on the aircraft structure, so suitable methods were developed. Vibration equipment capable of recording a number of stations simultaneously was used, so that the frequency and relative phase and amplitude of various points on the structure could be obtained to define the deflection curves of the vibrating structure. Fig. 2 shows a diagram of pickup locations on the structure. The wing is analyzed by placing pickups along two lines, one near the trailing edge, the other near the leading edge, so as to determine two deflection curves of the vibrating wing. From the two deflection curves it is possible to determine the amount of bending and torsion present at each wing station. The stabilizer and fin are analyzed by the same method as used for the wing analysis.

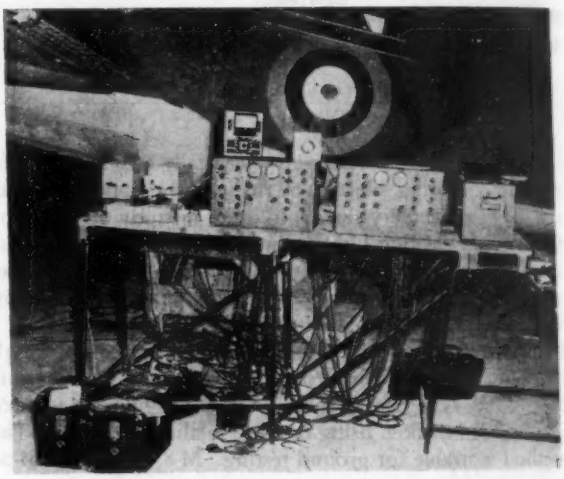
To determine the fuselage modes of vibration, the pickups are placed around the fuselage so as to complete a ring at each fuselage station being analyzed. In the case of wing, stabilizers, and fin, the pickups are so placed on the structure as to measure the motion perpendicular to the surface. In the case of fuselage analysis, the pickups are mounted so as to measure the motion tangent to the surface. This makes it possible to study fuselage side bending, vertical bending, and torsion all with one set-up.

For studying wing fore-and-aft modes of vibration, both vertical and horizontal components are studied, to determine the amplitude as well as the direction of motion. Figs. 3 and 3A show a typical set-up used to analyze the ground modes of vibration. The exciter used was a rotating unbalanced weight driven by a variable-speed transmission through a flexible drive shaft. The pickups are attached to the wing so that two deflection curves of the vibrating wing could be determined. The output of each pickup is put into a separate amplifier and then into a multielement recording oscillograph so the frequency, amplitude, and phase relation of each wing station studied can be recorded simultaneously, thus determining the wing deflection curves for each mode of vibration.

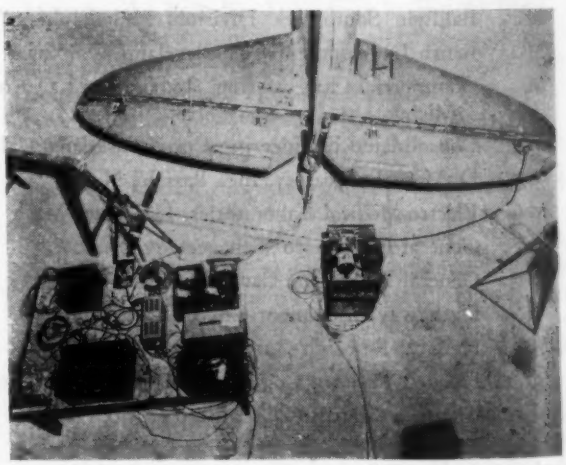
A typical set-up showing a method of exciting both sides of the airplane simultaneously is shown in Fig. 4. The important mode of stabilizer vibration being studied in this case was unsymmetrical, therefore, the weights of the two exciters were out of phase. In many cases where two wing coupled modes of vibration have very nearly the same frequency, it is necessary to attach exciters to the leading and trailing edges, set with the proper phase so as to separate bending, torsional, and fore-and-aft excitation.

Wing Analysis – Some examples of coupled modes of

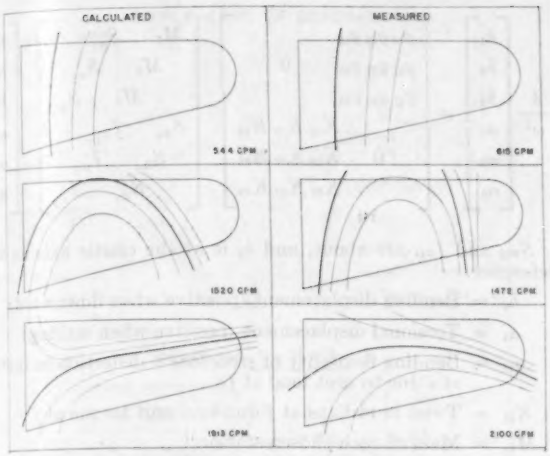
■ Fig. 3 – Typical ground vibration set-up – pickups attached to wing



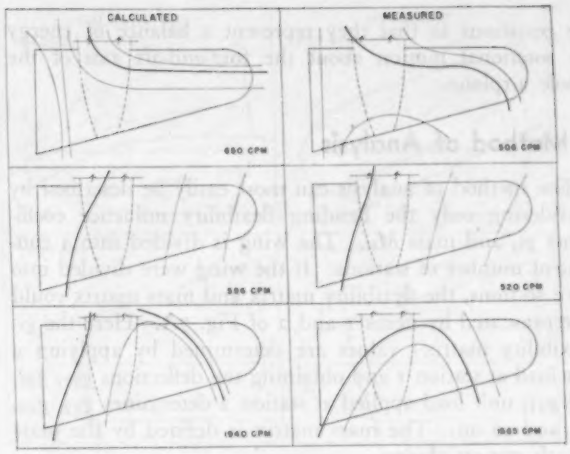
■ Fig. 3A – Typical ground vibration set-up – vibration recording equipment



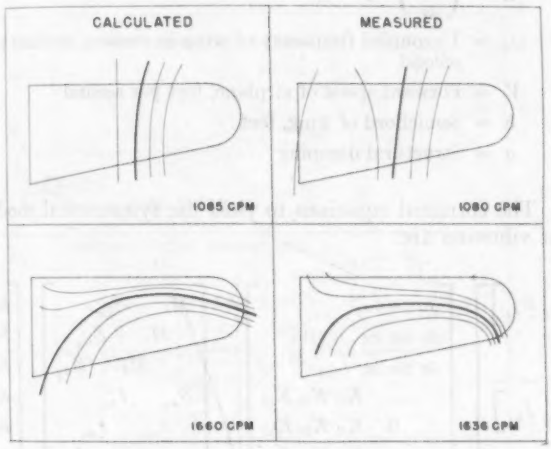
■ Fig. 4 – Double exciter set-up for exciting stabilizer vibration



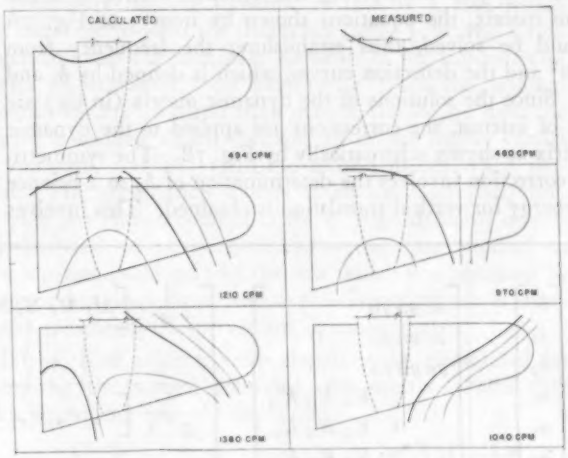
■ Fig. 5A – Measured and predicted wing vibration for a single-engine airplane – symmetrical mode



■ Fig. 6A – Measured and predicted wing vibration for a twin-engine airplane – symmetrical mode



■ Fig. 5B – Measured and predicted wing vibration for a single-engine airplane – unsymmetrical mode



■ Fig. 6B – Measured and predicted wing vibration for a twin-engine airplane – unsymmetrical mode

vibration of aircraft wings are shown by Figs. 5A, 5B, 6A, and 6B. Since the method for calculating the wing modes is broken into two parts, the figures are divided in the same manner. Two separate sets of equations are obtained for the wing analysis, one that yields the symmetrical modes of vibration, the others that yield the unsymmetrical modes. Fig. 5A shows the symmetrical modes for a single-engined airplane and Fig. 5B, the unsymmetrical modes. In this type of structure where there are no large concentrated masses, the coupling between bending and torsion is not as great as in the case of multi-engined airplanes, which is shown by Figs. 6A and 6B for a two-engined airplane. Of the six wing modes of vibration shown, there is not one mode that can be classed as an uncoupled mode, in which the torsion and bending are almost completely separated. The agreement between the calculated and measured values is considered good since the calculations were made from design data and were completed almost a year before the airplane was finished, and also since the calculations assumed a free body, and the airplane was tested while on its landing gear.

The calculations referred to above were by the Wylie method and are based on the iteration process of solving matrix equations. The matrix is simply a tabular method of representing the parts of simultaneous equations which contain the mass and stiffness properties of the structure, and the iteration process is a systematic trial and error method of determining the frequencies and deflection curves of the vibrating structure. Actually, instead of using the stiffness properties of the structure, the reciprocal of this or the flexibility is used, so that the first solutions that the iteration yields are then the lowest rather than the highest frequencies.

The matrix is first established for the wing as if it were anchored in a jig at the root, and is then corrected to represent the wing as a free body. Two corrections were made to the original matrix. The first corrected matrix yields the symmetrical modes of vibration, while the second matrix yields the unsymmetrical modes. The correction necessary to establish the symmetrical dynamic matrix is evolved by correcting the equations so that they represent a balance of energy for vertical translational motion of the whole airplane. The correction necessary to establish the unsymmetrical dynamic matrix is obtained by correcting

the equations so that they represent a balance of energy for rotational motion about the fore-and-aft axis of the whole airplane.

■ Method of Analysis

The method of analysis can most easily be described by considering only the bending flexibility influence coefficients g_{ij} and mass M_{ii}. The wing is divided into a convenient number of stations. If the wing were divided into three sections, the flexibility matrix and mass matrix could be represented by items 1 and 2 of Fig. 7A. Here the g_{ij} (flexibility matrix) values are determined by applying a unit load at station 1 and obtaining the deflections g_{11}, g_{21}, and g_{31}, unit load applied at station 2 determines g_{12}, g_{22}, g_{32}, and so on. The mass matrix is defined by the mass at each station chosen.

The combination of the flexibility matrix and mass matrix establishes the dynamic matrix for the wing in the jig. This is obtained by post multiplying the flexibility matrix by the mass matrix. By the repeated iteration process of post multiplying the dynamic matrix by a column matrix, the equations shown by item 3 of Fig. 7A would be solved, thus establishing the frequency from $1/\omega^2$ and the deflection curves, which is defined by h_i and α_i. Since the solutions of the dynamic matrix (in jig) are not of interest, the corrections are applied to the dynamic matrix as shown schematically by Fig. 7B. The symmetrical correction involves the determination of h_0 so a balance of energy for vertical translation is obtained. This involves the mass of ½ the fuselage. The unsymmetrical correction is made by determining the angle of roll of the fuselage necessary to balance the rotational energy of the airplane about the fore-and-aft axis through the fuselage.

The corrections necessary for the equations which combine bending, torsion, and unbalance are essentially the same as described for pure bending, and will be shown by the complete equations.

The fundamental equations upon which the method is based are shown as follows. First the uncorrected equations which would yield the frequencies if the wing were mounted in a jig are:

$$\frac{1}{\omega^2}\begin{bmatrix} h_1 \\ h_2 \\ h_3 \\ \alpha_1 \\ \alpha_2 \\ \alpha_3 \end{bmatrix} = \begin{bmatrix} g_{11}\,g_{12}\,g_{13} & & \\ g_{21}\,g_{22}\,g_{23} & 0 & \\ g_{31}\,g_{32}\,g_{33} & & \\ & K_{11}\,K_{12}\,K_{13} & \\ 0 & K_{21}\,K_{22}\,K_{23} & \\ & K_{31}\,K_{32}\,K_{33} & \end{bmatrix} \begin{bmatrix} M_1 & & S_{a_1} & & & \\ & M_2 & & S_{a_2} & & \\ & & M_3 & & S_{a_3} & \\ S_{a_1} & & & I_{a_1} & & \\ & S_{a_2} & & & I_{a_2} & \\ & & S_{a_3} & & & I_{a_3} \end{bmatrix} \begin{bmatrix} h_1 \\ h_2 \\ h_3 \\ \alpha_1 \\ \alpha_2 \\ \alpha_3 \end{bmatrix}$$

$S_{a_{ii}}$ and $I_{a_{ii}}$ are about, and h_i is at the elastic axis as the reference.

- h_i = Bending displacement (positive when downward)
- α_i = Torsional displacement (positive when stalling)
- g_{ij} = Bending flexibility of structure – deflection in inches at i due to unit load at j
- K_{ij} = Twist in radians at i due to a unit torque at j
- M_{ii} = Mass of each chosen station
- $S_{a_{ii}}$ = Unbalance of each station about elastic axis
- $I_{a_{ii}}$ = Polar moment of inertia of each station about elastic axis
- ω = Flutter frequency, radians per second
- $\Omega = \left(\frac{\omega_a}{\omega}\right)^2 (1 + jg)$
- ω_a = Uncoupled frequency of wing in torsion, radians per second
- V = Forward speed of airplane, feet per second
- b = Semichord of wing, feet
- g = Structural damping

The corrected equations to yield the symmetrical modes of vibration are:

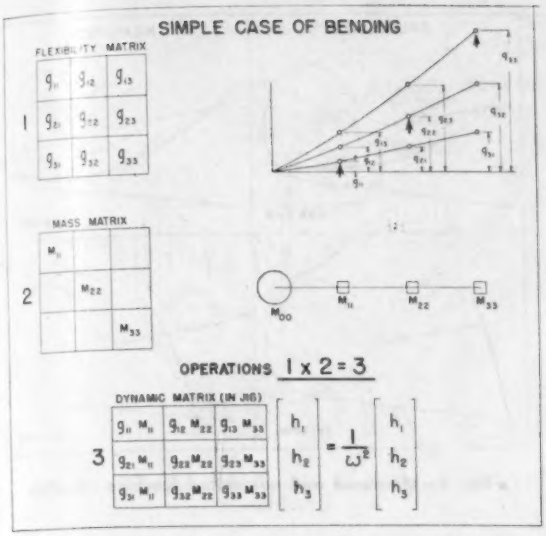
$$\frac{1}{\omega^2}\begin{bmatrix} h_1 \\ h_2 \\ h_3 \\ \alpha_1 \\ \alpha_2 \\ \alpha_3 \end{bmatrix} = \left\{ \begin{bmatrix} g_{11}\,g_{12}\,g_{13} & & \\ g_{21}\,g_{22}\,g_{23} & 0 & \\ g_{31}\,g_{32}\,g_{33} & & \\ & K_{11}\,K_{12}\,K_{13} & \\ 0 & K_{21}\,K_{22}\,K_{23} & \\ & K_{31}\,K_{32}\,K_{33} & \end{bmatrix} - \begin{bmatrix} \frac{1}{\Sigma M_{ii}} \begin{bmatrix} M_1\,M_2\,M_3\,S_{a_1}\,S_{a_2}\,S_{a_3} \end{bmatrix} \\ \hline \frac{1}{\Sigma I_{a^{ii}}} \begin{bmatrix} S_{a_1}\,S_{a_2}\,S_{a_3}\,I_{a_1}\,I_{a_2}\,I_{a_3} \end{bmatrix} \end{bmatrix} \begin{bmatrix} g_{11}\,g_{12}\,g_{13} & & \\ g_{21}\,g_{22}\,g_{23} & 0 & \\ g_{31}\,g_{32}\,g_{33} & & \\ & K_{11}\,K_{12}\,K_{13} & \\ 0 & K_{21}\,K_{22}\,K_{23} & \\ & K_{31}\,K_{32}\,K_{33} & \end{bmatrix} \right\} \begin{bmatrix} M_1 & & S_{a_1} & & & \\ & M_2 & & S_{a_2} & & \\ & & M_3 & & S_{a_3} & \\ S_{a_1} & & & I_{a_1} & & \\ & S_{a_2} & & & I_{a_2} & \\ & & S_{a_3} & & & I_{a_3} \end{bmatrix} \begin{bmatrix} h_1 \\ h_2 \\ h_3 \\ \alpha_1 \\ \alpha_2 \\ \alpha_3 \end{bmatrix}$$

which is reduced to the following before an iteration is undertaken:

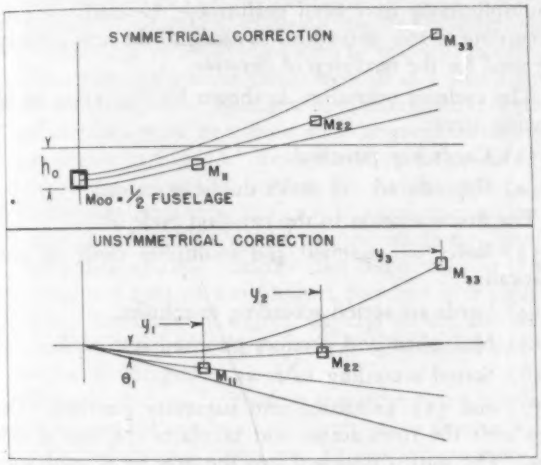
$$\frac{1}{\omega^2}\begin{bmatrix} h_1 \\ h_2 \\ h_3 \\ \alpha_1 \\ \alpha_2 \\ \alpha_3 \end{bmatrix} = \begin{bmatrix} \times & \times & \times & \times & \times & \times \\ \times & \times & \times & \times & \times & \times \\ \times & \times & \times & \times & \times & \times \\ \times & \times & \times & \times & \times & \times \\ \times & \times & \times & \times & \times & \times \\ \times & \times & \times & \times & \times & \times \end{bmatrix} \begin{bmatrix} h_1 \\ h_2 \\ h_3 \\ \alpha_1 \\ \alpha_2 \\ \alpha_3 \end{bmatrix}$$

The corrected equations to yield the unsymmetrical modes of vibration are:

$$\frac{1}{\omega^2}\begin{bmatrix} h_1 \\ h_2 \\ h_3 \\ \alpha_1 \\ \alpha_2 \\ \alpha_3 \end{bmatrix} = \left\{ \begin{bmatrix} g_{11}\,g_{12}\,g_{13} & & \\ g_{21}\,g_{22}\,g_{23} & 0 & \\ g_{31}\,g_{32}\,g_{33} & & \\ & K_{11}\,K_{12}\,K_{13} & \\ 0 & K_{21}\,K_{22}\,K_{23} & \\ & K_{31}\,K_{32}\,K_{33} & \end{bmatrix} - \begin{bmatrix} \frac{y_i}{\Sigma y_i^2 M_{ii}} \begin{bmatrix} y_1 M_1\,y_2 M_2\,y_3 M_3\,y_1 S_{a_1}\,y_2 S_{a_2}\,y_3 S_{a_3} \end{bmatrix} \\ \hline 0 \end{bmatrix} \begin{bmatrix} g_{11}\,g_{12}\,g_{13} & & \\ g_{21}\,g_{22}\,g_{23} & 0 & \\ g_{31}\,g_{32}\,g_{33} & & \\ & K_{11}\,K_{12}\,K_{13} & \\ 0 & K_{21}\,K_{22}\,K_{23} & \\ & K_{31}\,K_{32}\,K_{33} & \end{bmatrix} \right\} \begin{bmatrix} M_1 & & S_{a_1} & & & \\ & M_2 & & S_{a_2} & & \\ & & M_3 & & S_{a_3} & \\ S_{a_1} & & & I_{a_1} & & \\ & S_{a_2} & & & I_{a_2} & \\ & & S_{a_3} & & & I_{a_3} \end{bmatrix} \begin{bmatrix} h_1 \\ h_2 \\ h_3 \\ \alpha_1 \\ \alpha_2 \\ \alpha_3 \end{bmatrix}$$



■ Fig. 7A – Fundamental elements for obtaining calculated vibration data



■ Fig. 7B – Fundamental elements for obtaining calculated vibration data

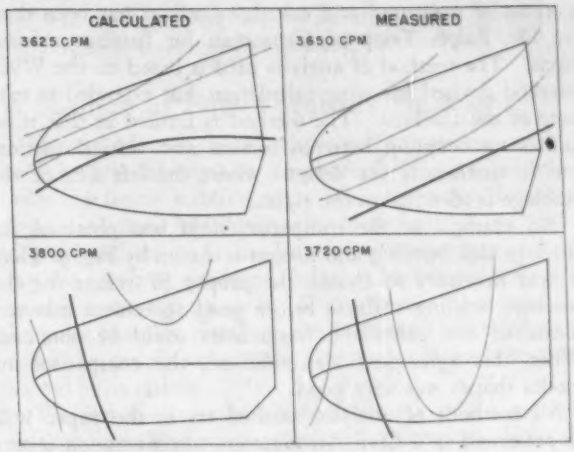
which, of course, is reduced to a square matrix before the iteration process is started.

The iteration process is a systematic method of ironing out the errors in the assumed column of numbers (deflection curve) that is chosen to be premultiplied by the dynamic matrix. If the correct deflection curve could be assumed, only one iteration would, of course, be necessary to verify the solution. However, if an arbitrary column of ones is chosen as the first trial, a number of iterations are necessary to obtain convergence, which occurs when the solution has been obtained.

The two foregoing equations will yield the first symmetrical and first unsymmetrical modes of vibration and frequencies for the wing. The bending and torsion at station 1 is defined by h_1 and α_1, and so on, and the frequency is obtained from the factor $1/\omega^2$.

■ Determination of Higher Modes

In order to determine the higher modes of vibration, the "dominate tendencies" of the fundamental modes have



■ Fig. 8 – Measured and calculated stabilizer vibration – unsymmetrical stabilizer

to be removed from the equations. This is accomplished by premultiplying the dynamic matrix by a row which will converge to yield the frequency factor $1/\omega^2$ and a row which contains the dominant elements of the fundamental modes of vibration. By subtracting from the original dynamic matrix an elimination matrix, the second dynamic matrix is formed which contains all of the higher modes of vibration, and will yield the second modes when iterated. The elimination matrix is obtained by post multiplying any chosen column of the original dynamic matrix (symmetrical or unsymmetrical, as they are handled as two separate problems) by the row which was obtained by iteration but properly ratioed so the chosen column is eliminated completely in the second dynamic matrix.

If the third column were chosen to be eliminated for obtaining the second solutions, the matrix would then look something like the following:

$$\frac{1}{\omega^2}\begin{bmatrix} h_1 \\ h_2 \\ h_3 \\ \alpha_1 \\ \alpha_2 \\ \alpha_3 \end{bmatrix} = \begin{bmatrix} \times & \times & 0 & \times & \times & \times \\ \times & \times & 0 & \times & \times & \times \\ \times & \times & 0 & \times & \times & \times \\ \times & \times & 0 & \times & \times & \times \\ \times & \times & 0 & \times & \times & \times \\ \times & \times & 0 & \times & \times & \times \end{bmatrix} \begin{bmatrix} h_1 \\ h_2 \\ h_3 \\ \alpha_1 \\ \alpha_2 \\ \alpha_3 \end{bmatrix}$$

The higher modes are obtained by the same process of elimination until as many modes as desired have been obtained.

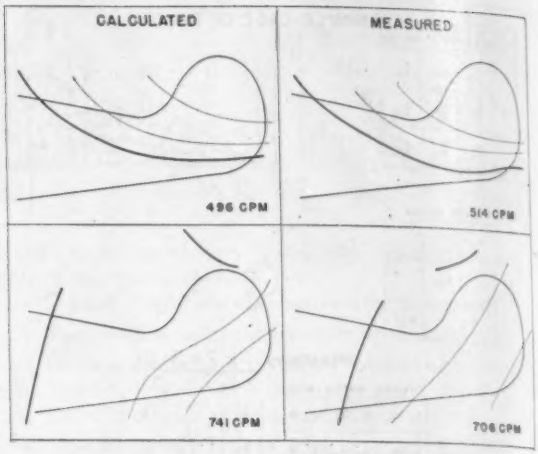
Stabilizer Analysis – With trouble having been encountered in a number of stabilizers on certain airplanes, investigations were conducted to determine the frequencies and deflection curves of the stabilizer modes that were involved. The experimental data were obtained first and calculations made later to determine necessary parameters to yield the known mode shapes and frequencies. This was done to aid in determining the actual stiffness properties present in the structure, and correlate with the stiffness properties obtained from design information. Fig. 8 shows the results of investigations on two different airplanes. It is interesting to note that, even though the modes are very different, the frequency of both stabilizers is about 3800 cpm.

Fuselage Analysis – Similar work to determine the com-

parison of measured and calculated values has been done by Dr. Ralph Tripp of Grumman for fuselage calculations. The method of analysis used is based on the Wylie method devised for wing calculation, but extended to take care of the fuselage. The method is limited in that it assumes no coupling between torsion and vertical motion, but is satisfactory for designs where the left side of the fuselage is identical to the right.

An example of the comparison that was obtained for fuselage side bending and torsion is shown by Fig. 9. Here it was necessary to choose the proper *EI* values for the fuselage bending stiffness before good agreement between measured and calculated frequencies could be obtained. When this agreement was obtained, the comparison of modes shapes was very good.

All methods of analysis touched on in the paper will be presented in a Civil Aeronautics Administration report which will be issued in the near future.



■ Fig. 9 – Measured and calculated fuselage vibration

■ Punched Card Calculating Machines

The iteration process of solving matrix equations is a very effective means for obtaining the coupled modes of vibration of the aircraft structure, but is a long and tedious process if run by using conventional calculating machines. The use of punched card calculating machines was, therefore, investigated to determine their adaptability to such a complex problem.

After quite a long search to find the proper place to undertake such a development, the Bureau of Census was interested and the work was put under the direction of Everett Kimball. He was able to see the pattern which the iteration of a matrix takes and apply it to the machines successfully.

The method used at first was limited to the range of numbers which appeared in the matrix, meaning that if some very large numbers occurred in certain cells of the matrix, the significance of the smaller numbers would be lost. This was later remedied by handling five significant numbers in every cell of the matrix and taking care of the decimal point of each individually.

A card specimen showing the various fields into which the numbers are punched is shown on Fig. 10. The trial, original or transposed matrix, row, column, number taken from cell of matrix, its power of 10, and whether plus or minus are punched in the first 15 columns of the card. Columns 16 to 24 make up the multiplier with its sign and power of 10. When these values are multiplied together, their product is punched in columns 34 to 43 with the proper sign and power of 10. The step of iteration being performed is shown in columns 54 and 55. After the multiplications have been performed, the cards are sorted according to row and added to establish the new column to be used for the next step of iteration.

The cycle of operations is shown by Fig. 11 as the following steps:

(1) Cards key punched.

(2) Reproduced – to make duplicate cards.

The first operation in the iteration cycle is:

(3) Rows are summed and multiplier cards cut automatically.

(4) Cards are sorted according to column.

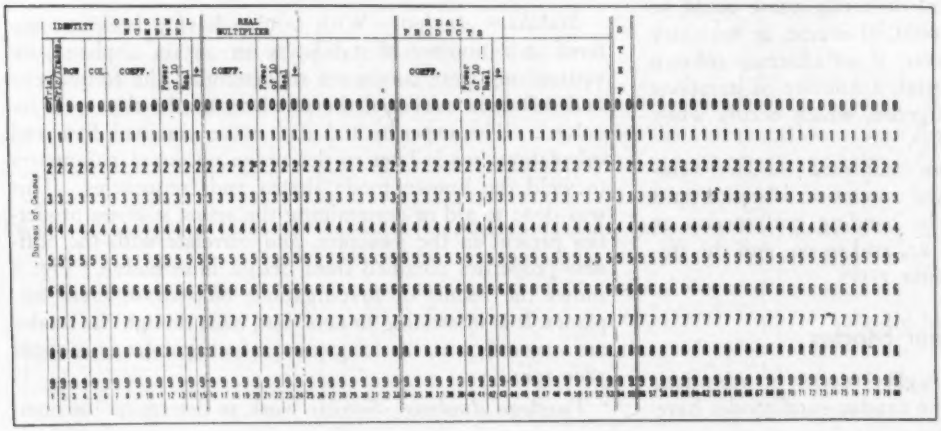
(5) Multiplied and product punched into card.

(6) Sorted according to rows.

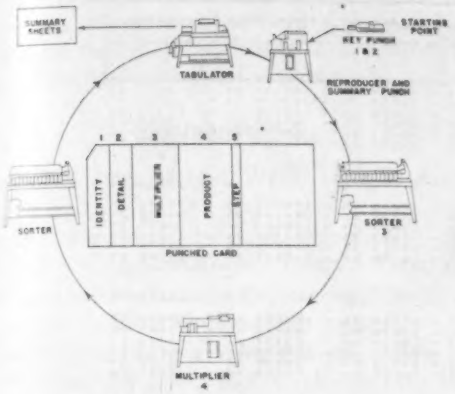
(7) and (3) Tabulated and summary punched. This step adds the rows across and tabulates the sum of each row. The sum is punched into the new set of cards automatically and the new set is ready for the second step of iteration, repeating the cycle until convergence is obtained.

With this system, it is possible to compute the coupled modes of vibration that exist at zero air speed, and have shown excellent agreement with the ground modes of vibration measured experimentally.

The next step was to adapt the machines to handle



■ Fig. 10 – Details of punched card used for calculating aircraft ground vibration modes – for real numbers only



■ Fig. 11 – Punched card machine cycle for iteration of matrices

complex numbers so that the modes of vibration at any air speed could be computed. This necessitated finding means of recognizing a varied combination of real and imaginary numbers, plus and minus values as well as a sign change when two imaginary numbers are multiplied together. These obstacles were overcome and the cards were modified as shown by Fig. 12. Two sets of cards are made up for the original number, one set for the real values, the other for the imaginary values. The columns added to the card as shown in Fig. 10 are the imaginary part of the multiplier which is punched into columns 25 to 33 and the imaginary part of the product punched into columns 44 to 53. Columns 14, 23, 32, 42, and 52 are also added to identify the numbers in the preceding columns as being either real or imaginary.

The cards are read by putting them through a tabulator, which records the results of the iteration, as shown by Fig. 13. The tabulating sheet lists all the information contained on each card plus the conformed products, which is the result of the process of converting the various powers of 10 to the same power of 10 and then printing the sum of the row. When these summaries are listed on the tabulating sheet, they are also automatically punched into a new set of cards, thus performing the first step of the new iteration.

The same numbers taken out of one step of iteration are fed into the next step of iteration, thus making any ratioing between steps unnecessary.

■ Solution of Flutter Stability Equations

Having adapted the punched card calculating machines to take care of complex numbers, it became possible to solve the flutter stability equations. Normally these equations are solved by putting the coefficients of the various degrees of freedom in the form of a determinant and solving the resulting quadratic, cubic, or higher-order equation to obtain the various roots. With the machines capable of iterating complex numbers, the coefficients were put in the form of a matrix rather than a determinant, and the matrix solved by iteration.

The iteration process has been set up on the machines so that several values of $v/b\omega$ and different values of frequency ratio can be computed simultaneously, so that the complete curves defining the stability of the aircraft modes being analyzed can be plotted for the speed range being studied.

One advantage of the iteration method of solving the stability equations is that of being able to study four or more degrees of freedom with a straightforward method of analysis, whereas by using determinants it is difficult to solve the higher-order equations that result.

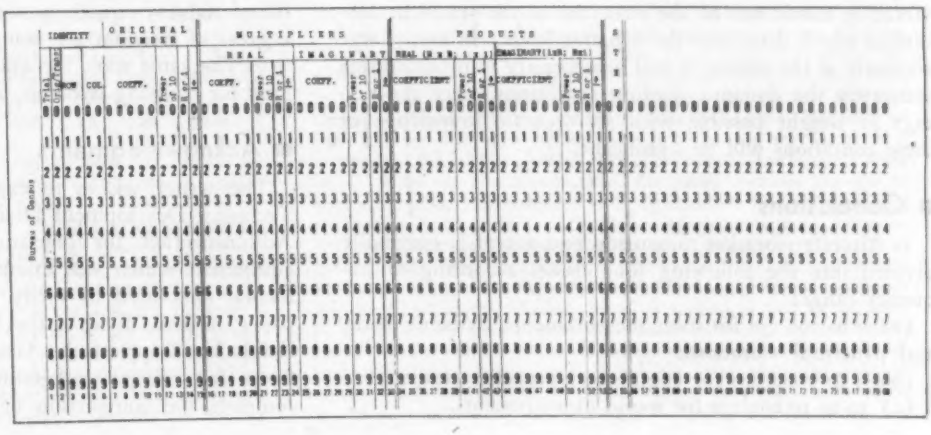
To prove the soundness of the principle, a set of equations using three degrees of freedom, which had been solved by the conventional method, was studied. The results are shown in Table 2, where several values of $v/b\omega$ were studied which represented the speed range of the aircraft.

Table 2

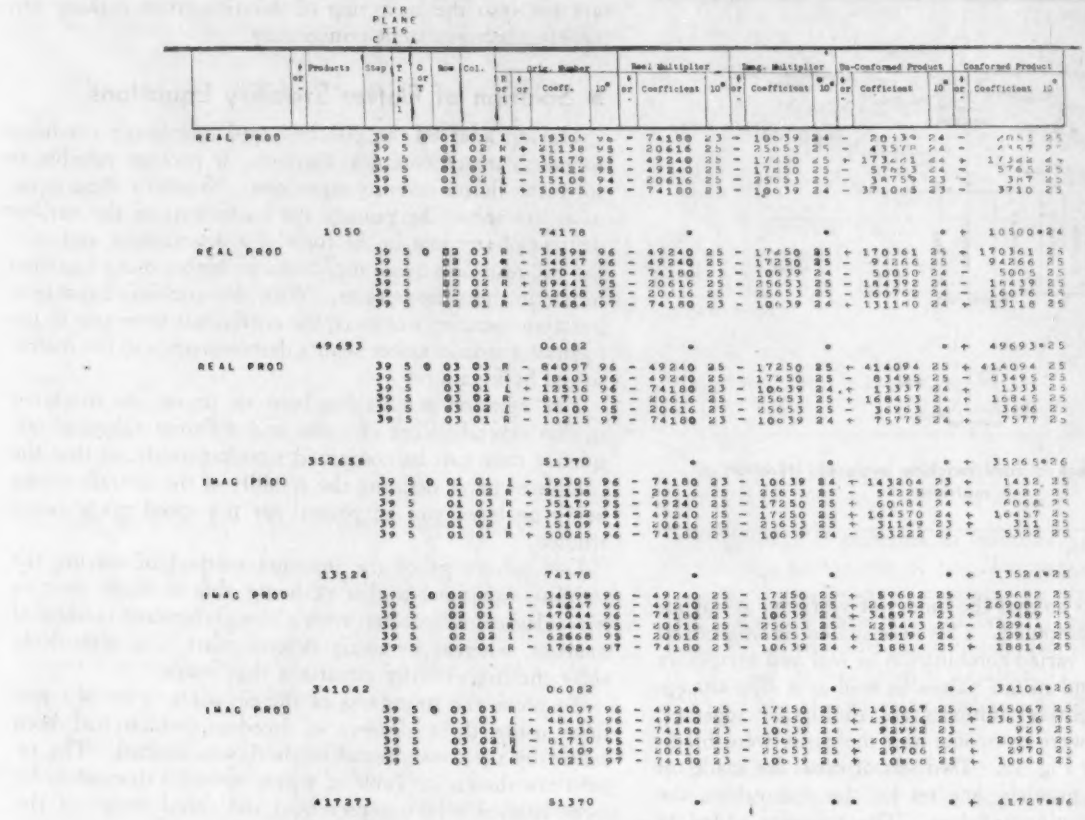
$v/b\omega$	Ω_1	Ω_2	Ω_3
By Determinant			
0.25	0.8578 – 0.0180i	5.748 – 0.1286i	2.866 – 0.4458i
0.83	0.9182 – 0.0625i	5.674 – 0.4358i	2.204 – 1.4690i
1.67	1.1010 – 0.0579i	5.483 – 0.9384i	−0.0130 – 2.8860i
By Matrix with Punched Card Machine			
0.25	0.8583 – 0.01946i	5.7412 – 0.12795i	2.873 – 0.4449i
0.83	0.8810 – 0.07830i	5.6729 – 0.44370i	2.2093 – 1.4478i
1.67	1.1014 – 0.05760i	5.4804 – 0.93900i	−0.0034 – 2.8852i

■ Dynamic Loads

Most aircraft companies have found that the measuring of dynamic loads has become a necessity either to check load assumptions or establish causes of structural failures. To make these studies, electronic equipment has been em-



■ Fig. 12 – Details of punched card used for calculating aircraft modes of vibration at various air speeds – for complex numbers



■ Fig. 13 – Summary sheet showing the results from punched card calculating machines

ployed which is very similar to that used for making vibration studies but capable of recording a different frequency range. The main difference being that vibration equipment need only measure frequencies down to one or two cycles per second, whereas dynamic load equipment is required to measure zero frequency conditions. Essentially, the two problems resolve to but one basic problem, vibration studies being made to determine the steady-state vibrating conditions of the structure and dynamic load studies being made to determine the transient conditions of the structure.

As in the cases of experimental determination where the same electronic equipment is used for studying both problems, so it is with the mathematical tools for studying the problems. The matrix representing the mass and stiffness properties of the structure will yield either the steady-state vibrating conditions of the structure or the transient conditions which determine the dynamic loads. In many large airplanes of the future, it will be necessary to predetermine accurately the dynamic loading conditions, since the savings in weight possible with an accurate knowledge of these conditions will be significant.

■ Conclusions

1. Aircraft vibration measuring equipment is essentially divided into the following four classes according to frequency range:

(a) 0 to 100 cps for static measurements, dynamic loads, and structural vibrations.

(b) 5 to 1000 cps for engine and propeller studies.

(c) 30 to 10,000 cps for sound measurements.

(d) 5000 to 15,000 cps for detonation indicators.

2. Good agreement has been obtained between measurements and calculations of coupled modes of vibration of aircraft structures.

3. The Wylie method of determining the coupled modes of vibration of single- or multiengined airplanes is sound, provided large concentrated masses, such as floats, do not extend out from the structure.

4. Punched card calculating machines can be successfully and economically used for making vibration and flutter studies as well as any other studies that use the iteration process of solving matrix equations. In fact, the possibilities of routine calculations on such machines appear unlimited.

5. The use of punched card machines for solving the flutter stability equations is economical if more than three degrees of freedom are being studied.

6. The same tools for making vibration studies can be used for studying dynamic loads.

■ Acknowledgments

The writer wishes to express his appreciation to the Technical Development Division of the Civil Aeronautics Administration for procuring the vibration measuring equipment which was essential for obtaining the experimental data used to verify the analyses covered by this paper; to Jean Wylie of the Douglas Aircraft Co. and Dr. Ralph H. Tripp of the Grumman Aircraft Engineering Corp. for valuable suggestions and calculations used to complete the comparison of experimental and analytical

data; also to Everett Kimball of the Bureau of Census, Department of Commerce, and F. R. Schmidt of the Aircraft-Engineering Division of the Civil Aeronautics Administration, for their untiring efforts in developing the punched card machine procedures as described.

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DISCUSSION

Suggests Alternate Iteration Method

–Lee S. Wasserman
Matériel Command, AAF

This paper by Mr. Critchlow shows that by the use of automatic punched card calculating machines, sufficient time can be saved to make practical the application of iteration methods in solving vibration and flutter problems involving many coupled degrees of freedom. Methods generally in use at the present time involve direct solutions of the stability determinants by methods substantially as described in AAF Technical Report No. 4798, and thus the number of degrees of freedom used in a particular problem is limited to a minimum (usually three or less) in order to avoid the excessive number of man-hours required to solve complex equations higher than the third order. For this reason it has been found necessary to represent the structure by a small number of degrees of freedom, which have been taken as the uncoupled vibration modes. Experience to date has indicated that reasonable correlation between theory and experiment is obtained with this method in a large number of cases involving ground vibration test data and flutter model test points.

The advisability of applying methods employing large numbers of degrees of freedom in aircraft vibration and flutter analyses is thus subject to some question. There are also other points involved in this question, namely:

a. How many degrees of freedom should be assumed? Should the analysis be stopped at the wing or tail or should the whole airplane be considered simultaneously?

b. Are the errors introduced by the simplifying assumptions involved in limiting the number of degrees of freedom comparable with those inherent in using potential airfoil theory without theoretically correct aspect ratio and compressibility corrections?

c. What compromises should be made between the number of man-hours expended and the probable accuracy of the results?

Present indications are that it may be advisable to carry out such complete investigations only in modes which are marginally critical on the basis of the simpler theory. Some work has been carried out at the Matériel Command in the last several months on iteration methods and the procedures developed are being applied to a large AAF airplane to determine their suitability for routine work. Present plans are that the matrix for this case will be set up by the Matériel Command and its solution will be carried out by the Bureau of Census under the direction of Mr. Critchlow and Everett Kimball. The results will be checked with those obtained from the simpler theory and a tentative decision will probably be reached on the advantages of the iteration method as used with automatic keypunch machines.

The following improvements have resulted from this Matériel Command investigation into the problem of applying iteration methods:

a. A simple method of eliminating the dominant root.

b. A method of estimating the values of the coordinates for the subdominant root from the iteration computation of the dominant root.

The first method is based upon the fact that the deflections corresponding to any one coupled mode are independent of the deflections corresponding to any other coupled mode. Thus, when a set of coordinates representing one coupled mode is known, the determination of its frequency and stability may be made independently of all the remaining degrees of freedom. For example, the equation for each degree of freedom will give the solution $\Omega = \Omega_1$ if the coordinates corresponding to Ω_1 are substituted in any of these equations. In this case the deflections $x_1, x_2, \ldots x_n$ in the various degrees of freedom are related to a deflection in one degree of freedom, say x_1, by certain constants; that is, $x_1 = x_1$, $x_2 = c_2 x_1, \ldots, x_n = c_n x_1$. Now then, if the substitutions are made that $x_2 = c_2 x_1 + x_{20}, \ldots, x_n = c_n x_1 + x_{n0}$, and the equations are written in terms of x_1 and $x_{20}, \ldots, x_{n0}$, the coefficients of x_1 in all equations will be proportional to $\Omega - \Omega_1$. The x_1 terms can be eliminated from all the equations and the number of independent equations is reduced by one so that the subdominant root is obtained in the next iteration.

The second method makes use of the way in which the iterations converge to the various deflection curves under consideration. From analogy to Stodala's process it may be seen that the deflection curves corresponding to each value of ω_i are divided by ω_i^2 by each iteration. Thus, the rate at which each mode is eliminated from the assumed deflection curve is directly proportional to $\left(\frac{\omega_i}{\omega_1}\right)^2$. Once the dominant deflection curve has been obtained it may be reasoned that in the previous iteration, the main error is caused by the deflection curve corresponding to the subdominant root. Thus, if the curve obtained from the final iteration is subtracted from that of previous iteration divided by ω_1^2 a good approximation of the subdominant deflection curve is obtained.

With respect to recent improvements in equipment, the Matériel Command has been using an electronic switch mixing circuit operating into a cathode-ray oscillograph for ground testing. This is a portable unit which permits the deflection curves to be obtained directly in any mode since the amplitude and phase at any station can be directly compared to that at any other station. Crystal pickups, although very light in weight, have not been found to be particularly rugged and reliable. Pickups developed by the Matériel Command include an amplitude gage and an accelerometer employing strain gages as the sensitive elements. These units are rugged and simple and the amplitude gage is particularly useful when the mass of available pickups is too heavy for the measurements contemplated.

Flight measurements have been made using carrier-type equipment as well as straight integrating amplifiers. The former has been used when any low frequencies are being investigated such as the measurement of the accelerations and strains. The straight integrating amplifier (frequency range from 3 to 1000 cps) has been used with both electromagnetic velocity and strain gage accelerometer pickups, with the latter used at low frequencies (3-30 cps). The recent developments in the electromagnetic pickups have allowed their universal use in any direction since the springs are relatively stiff and the effect of gravity is not important. The low natural frequency of these pickups has been obtained by utilizing the inertia of fluid pumped by the seismic element. Other instrument development work being carried out by the Matériel Command includes the use of sensitive oscillograph elements with direct input from pickups and strain gage units as well as the design of electromagnetic vibration exciting equipment. Recent vibration tests carried out by the Lockheed Aircraft Corp. using the latter type of equipment showed definite advantages such as simple and independent control of the magnitude and frequency of the vibratory impulses applied as well as the possibility of phasing the vibratory impulses from two or more vibrators so as to excite either symmetrical or unsymmetrical motions in bending and torsion. Such a procedure has a great advantage over the conventional methods of excitation when important symmetrical modes are obscured by unsymmetrical responses, and vice versa.

Getting a Better Grip o

by JOHN GAILLARD
Mechanical Engineer
American Standards Association

DURING the last few years, a number of industrial concerns have successfully applied an approach to the problem of quality control of manufactured products based on statistical theory and put into practice by means of a simple device, the quality control chart. Instructions for the use of such a chart are given in a set of three American War Standards approved and published by the American Standards Association.[1]

Applicable particularly to highly repetitive operations, the control chart method presents vast opportunities for getting a better grip on quality control, for example, in regard to physical and chemical properties of raw materials or the dimensional accuracy of interchangeable parts.

The main advantages of the control chart method are:

1. Reduction in rejections.
2. Possibility of spotting impending trouble in the production process and removing the cause of that trouble before rejections occur.
3. Increased speed of production through avoidance of rejections and interruptions of the process.
4. Reduction of cost of inspection.
5. More efficient use of materials due to the possibility of working to closer manufacturing tolerances and lower safety factors.
6. Better basis for judgment as to whether selective assembly can be replaced by assembly of interchangeable parts.
7. Better basis for harmonizing specification requirements with available facilities.

As many engineers do not use statistical theory as a tool, they may wonder what its application to quality control can add to the use of the highly refined inspection methods and devices now available to industry. If manufacturing concerns are checking their gages, and sometimes even their products, against gage blocks varying from their nominal sizes by no more than a few millionths of an inch, how can the theory of probability help the practical quality control man do a better job? The answer is that no matter how refined the physical inspection apparatus used in a plant may be, quality control can be made most effective only if based on certain principles of statistical theory. There is nothing academic in the use of these principles, in spite of the "highbrowness" of statistical mathematics. The quality control chart is a simple device which can be set up and used by every engineer. Also, the new method can often be introduced without necessitating any change in the existing system of inspection, at the same time utilizing the information contained in the inspection data to much better effect.

[This paper was presented at the SAE War Engineering-Annual Meeting, Detroit, Mich., Jan. 13, 1944.]

[1] "Guide for Quality Control," Z1.1-1941; "Control Chart Method of Analyzing Data," Z1.2-1941; and "Control Chart Method of Controlling Quality during Production," Z1.3-1943. Copies available from the American Standards Association, 29 West Thirty-ninth Street, New York City, 18.

■ Individual Quality

When quality control is discussed by automotive engineers, a problem that probably will come to their mind first is control of *dimensions*, that is, the degree of accuracy to be maintained in manufacturing mechanical components. A simple example is the problem of holding the diameter of a cylindrical pin within manufacturing limits specified by the designer. In this case, the *quality characteristic* under consideration is the diameter of the pin, and the quality requirement to be met is that the diameter of each individual pin must lie within the two given limits if it is to be accepted. Whether each unit of product meets this requirement can be determined in either of two ways. We can *measure* the diameter and decide whether the observed value falls inside or outside the specified limits. Or we can check the diameter with go and not go limit gages and get the answer without knowing the measured value. The latter method is most commonly used in the automotive and other mass production industries.

■ Several Quality Characteristics May Have to Be Controlled

If more than one dimension of a component must be kept within specified limits, each dimension represents a quality characteristic. Still other essential characteristics to be controlled may be the smoothness of the surface of a part, its hardness, the tensile strength of the material of which it is made, and so on. To get control of the quality of the product as a whole, we must get control of all of these quality characteristics individually. Therefore, a separate control chart (or charts) must be used for each characteristic deemed essential to satisfactory performance of the product. To concentrate on the principle underlying the quality control chart and to simplify the present discussion, we shall assume here that we are dealing with only one characteristic, the diameter of a cylindrical pin. Furthermore, we shall assume that we decide whether a pin is acceptable by *measuring* its diameter, and not by gaging it. It is realized that in mass production such a procedure would be impractical. However, this may be disregarded in the present discussion, since the principle underlying the control chart may be applied whether the parts are measured or gaged.

■ Quality Varies from Unit to Unit

The quality of a product varies from unit to unit. This is no news to the engineer. The very reason why he establishes manufacturing limits for a part is that he knows the shop cannot hold all parts of a given kind to an *exact* dimension. These limits are meant to restrict variation in size to an amount small enough to be immaterial to good performance of the part.

Being accustomed to gaging practice, the engineer usu-

UALITY CONTROL

THE number of rejections can be reduced or, in many cases, trouble can be spotted in the production process before rejections occur, so that production can be increased – these are some of the advantages to be obtained when a control chart is used to control the quality of manufactured products.

The quality control chart here introduced by Mr. Gaillard is based on the statistical method.

It is applicable particularly to highly repetitive operations, and its use presents opportunities for getting a better grip on quality control in regard to physical and chemical properties of raw materials or the dimensional accuracy of interchangeable parts.

■ ■ ■

THE AUTHOR: JOHN GAILLARD has dealt with industrial standardization since receiving his Doctor's degree from the Institute of Technology in Delft, Holland, his native country. Mechanical engineer on the staff of the American Standard Association, Dr. Gaillard is also a consultant on tolerance and gaging systems, and secretary of the ASA War Committees on quality control and on cylindrical fits. While still in Holland, he had been assistant director of the Central Standards Bureau.

lly does not care much in what way the quality of cceptable units varies within their specified limits. Rather, is main concern is whether the quality (in our example, ize) of each unit is acceptable. For this reason, the engineer so far has not shown great interest in the question as o how the quality of individual units is *distributed* hroughout the product as a whole, even though, for ractical reasons, he may be interested in certain aspects f this distribution. For example, he knows that the *minimum metal limit* of a part should be watched more losely than the maximum metal limit because a part ejected for being outside the maximum metal limit can till be salvaged by removal of more metal. On the other and, where the minimum permissible clearance of a fit is mall, the transgression of the maximum metal limits of he mating parts may be critical. Such considerations may ause the quality control man to watch the measured sizes f parts, even though their acceptance or rejection is etermined on the go or not go basis. However, as will e shown in the following, more detailed knowledge about he distribution of quality is essential in order to get the est possible grip on quality control.

■ Collective Quality

The engineer is concerned mostly with the question, "How large a percentage of the parts manufactured falls within the specified limits?" Or conversely, "How large a percentage of rejects do we get?" In asking these questions, he no longer focuses his attention solely on the quality of an individual unit of product, but also on the question how good the product is as a whole. This *collective quality* can be measured in different ways. If 1000 pins are measured (or gaged), and 980 are found to be inside the specified limits and hence, acceptable, the collective quality of the entire lot can be said to be 98% effective or 2% defective, or again, it can be said that the fraction defective is 0.02. All of these expressions are simple and practical measures of collective quality. Also, they are the only possible types of measures of collective quality in cases where a unit of product is accepted or rejected without its individual quality being measured, such as in go and not go gaging.

■ Quality Distribution

However, if we *measure* the quality of every single unit in a lot (such as the diameter of each pin), we can express the collective quality of that lot in another way also. By plotting each measured unit quality as an abscissa, and the number of times this value occurs in the lot, as the corresponding ordinate, we get points of a frequency polygon or a frequency curve representing the distribution of the unit qualities throughout the lot. Contrasting with such measures of collective quality as the "per cent effective," which merely gives a quality *rating* for the lot, a distribution curve gives us a complete picture of its collective quality.

■ Level and Dispersion of Quality

In applying the control chart method, we must have the information contained in the distribution curve, in a more condensed form. This requirement can be met by expressing collective quality in its two fundamental characteristics. These are the *level of quality* and the *dispersion of quality*.

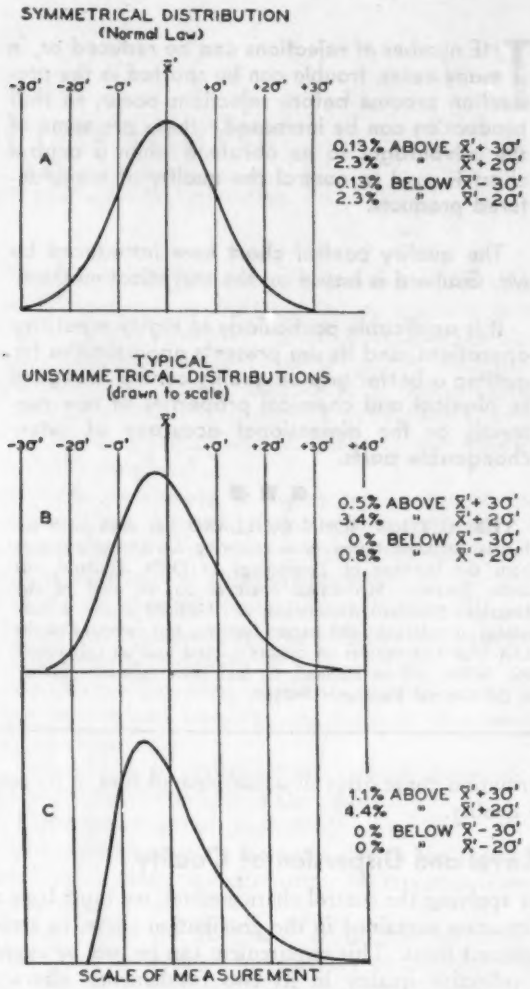
The *level* of quality can be simply expressed by the *average* of a number of quality measurements. For example, if a sample of n units is taken from the flow of product coming off the line, and the quality characteristic X (such as the diameter of a pin), measured on these units, gives the following observations: $X_1, X_2, \ldots\ldots X_n$, the level of quality of that sample is expressed by the average $\overline{X}$ (pronounced X bar):

$$\overline{X} = \frac{X_1 + X_2 + \ldots\ldots + X_n}{n}$$

The *dispersion* of quality can be expressed in several ways. A simple expression is the range of quality R. This is the difference between the highest value and the lowest value among a set of quality measurements, or $R = X_{max} - X_{min}$. For example, if the largest pin in a sample of 25 pins measures 0.9995 in. and the smallest pin measures 0.9987 in., the range for that sample is 0.0008 in.

Another measure of dispersion is the standard deviation, usually designated by the Greek letter σ (sigma). If quality measurements are made on n individual units of a sample, the standard deviation σ of these n observations from their average $\overline{X}$ (see above), is:

$$\sigma = \sqrt{\frac{(X_1 - \overline{X})^2 + (X_2 - \overline{X})^2 + \ldots + (X_n - \overline{X})^2}{n}}$$



If curves B and C were unsymmetrical in the opposite direction; the % values "above" and "below" would be interchanged.

■ **Fig. 1 – Percentages of quality values falling outside the three-sigma limits of distribution curves having different shapes**

This figure, which is reproduced from the American War Standard Z1.3-1942, also indicates the percentages falling outside the two-sigma limits; however, these percentages are not essential to the present discussion

The range R is simple to compute, requiring only a subtraction. Computation of the standard deviation σ requires a little more work, yet, nothing but simple arithmetic. The standard deviation is a more efficient measure of dispersion than is the range. The latter is used only for samples of 10 units or less.

■ Sample Qualities

A sample of n units taken from the flow of product gives us a picture of the state of affairs in the process at the time the sample was taken. We can formulate the information available in this picture by determining the sample quality, for example, in terms of $\overline{X}$ (sample average) and σ (standard deviation of the sample). A more continuous picture of conditions in the process during a given period will be obtained if we take a series of samples at regular intervals. The grand average of all of the sample averages will then give us the general level of quality of the samples taken. The standard deviation of the sample averages from the general level will give us an idea of the degree of uniformity of the product quality. The more closely the sample averages cluster about the general level, the smaller is the dispersion of quality and the higher the uniformity, according to the evidence supplied by the samples.

Suppose now that in determining such sample averages, we find one value that is particularly low. The question may then arise whether this low reading is due merely to an accidental piling up of low unit qualities in that particular sample, or whether it is a sign that there was trouble in the process which threw quality off balance. In other words, is that low reading for a sample quality significant or not? In the traditional form of quality control (that is, without the statistical approach), there is no answer to this question – we can only guess. The control chart method, however, does give us an answer, in a way to be explained later. We must first consider in some more detail the nature of quality distribution.

■ Causes of Variation in Quality

Quality varies from unit to unit in a product because in any production process,[2] innumerable causes of variation in quality are at work all the time. Some of these causes can be traced through their relation to the variations resulting from them and hence, are called *assignable causes*. Their elimination from the process will reduce the variation in quality and be a step in the direction of process control. Yet, after they have been removed, there will still remain variation in quality due to causes that cannot be found and hence, cannot be eliminated. There is nothing we can do in a systematic way to reduce this residual variation and we must, therefore, consider it as being due to chance. Accordingly, the causes that collectively are responsible for this minimum or irreducible amount of variation, are sometimes called a system of *chance causes*. However, the meaning of the term chance cause is relative, since a cause of variation that cannot be found today may become detectable in the future, due to technical progress.

A case in point is the influence of temperature changes on the accurate sizing of parts. There was a time when nobody paid attention to this matter, the dimensioning of parts still being so crude that the relatively small changes in the size of a part due to temperature variations in the workshop were insignificant. Later on, as manufacturing limits became closer, it was realized that the influence of temperature had to be taken into account – that is, temperature changes in the shop were recognized as an assignable cause of variation. However, for some time to come it was not deemed practicable to remove this assignable cause. At the present time this removal is achieved by controlling, for example, the temperature of the coolant used in grinding work to final size and by using a temperature-controlled room for inspection.

Some variations in the quality of a product are so insignificant from a practical point of view, that they do not warrant an investigation into their causes – even though these may be assignable and hence, not really chance causes. Therefore, instead of basing the investigation of a process on a classification of causes of variation into assignable causes and chance causes, as is sometimes done in the dis-

[2] The term production process is meant to include inspection procedure as well as manufacturing operations, since defective quality may be due to faulty inspection (chargeable, for example, to gage wear as well as to trouble in manufacturing (chargeable, for example, to tool wear).

cussion of quality control problems, the approach recommended by the American War Standards is based on a classification of variations in quality into (a) those which merit no investigation, and (b) those which are significant in that they are indicative of an assignable cause (or cause of trouble) in the process.

State of Control

As we have seen, gradual elimination of all assignable causes from a production process will lead to a condition where, with the process in use, we can do nothing further to reduce variation in quality. The process is then said to be in a *state of statistical control,* or briefly, in a *state of control.*

Here, we have a definite conception of control which is inherent in the process under consideration or, as we would say in mathematics, a function of the process. In this respect, the statistical approach has a distinct advantage over the traditional approach – that is, an approach made without the aid of the statistical method – where the attainment of control is *assumed,* for example, when the per cent defective has been reduced to a figure deemed to be satisfactory. This is a matter of personal judgment and the question whether the existing percentage can be reduced further is not answered in the traditional procedure.

How do we know that the state of control has been attained? While gradually removing assignable causes from the process, how can we determine at a given moment whether the variation in quality that is still left does not warrant any further investigation, or whether it still is significant and hence, calls for further reduction? That is, must we hunt for more assignable causes or can we stop hunting, assuming that variation in quality has hit the bottom? Is there any *criterion* that we can use in deciding whether a state of control appears to have been attained?

Quality Control Chart

Such a criterion is given us by the quality control chart, which is a graphical record of quality having two control limits placed on it in a manner to be explained in the next section. Data obtained from quality observations made on samples are plotted on the chart. If a plotted point falls *outside* the control limits, this should be taken as an indication that there is an assignable cause of variation (cause of trouble) in the production process. Thus, the quality control chart gives us a picture of the condition in the process and warns us when we must hunt for trouble. The control limits help us to classify causes of variation into the two groups mentioned before – assignable causes, and causes that merit no investigation.

Nature of Control Limits

How do we get the control limits that are placed on the control chart? To explain this, we go back to distribution curves. If we take a number of samples, measure the average quality $\overline{X}$ of each sample, and plot a distribution curve of the sample averages, the shape of this curve will vary with the conditions in the production process. Three different shapes of such distribution curves (*A, B,* and *C*) are shown in Fig. 1. The general average $\overline{X}'$ and the standard deviation are the same for all of the three curves. However, the shapes of the curves are different. Curve *A* is a normal curve, which is symmetrical, while curves *B* and *C* are unsymmetrical. Now, the normal curve *A* has the mathematical property that practically all (about 99.73%) of the values plotted fall within two limits located at equal distances $3\sigma'$ from the average $\overline{X}'$. These limits are called "three-sigma limits." As shown by the values given to the right of diagrams *A, B,* and *C* (Fig. 1), the percentage of values falling outside the three-sigma limits is higher for curves *B* and *C* than it is for curve *A,* the normal curve.

We can place such three-sigma limits on the control chart by computing their location, for a given size of sample, from quality measurements made on a series of samples and factors given in a table published in an appendix to the American War Standards. These three-sigma limits are then used as *control limits* while further quality measurements made on samples are being plotted on the control chart. If any such plotted point falls outside the control limits, this must be taken as a warning that there is an assignable cause of variation in the process which may spoil the quality of the product. That is, there is an indication of lack of control, and in order to get control, we must find the cause of trouble and remove it from the process. Practical experience with the control chart method, covering a wide range of industrial applications, has shown the effectiveness of such control limits as a means of "screening" significant variations in quality from nonsignificant ones, and as a warning system to look for a cause of trouble and remove it, as soon as possible after its presence has been indicated.

Application of Control Chart

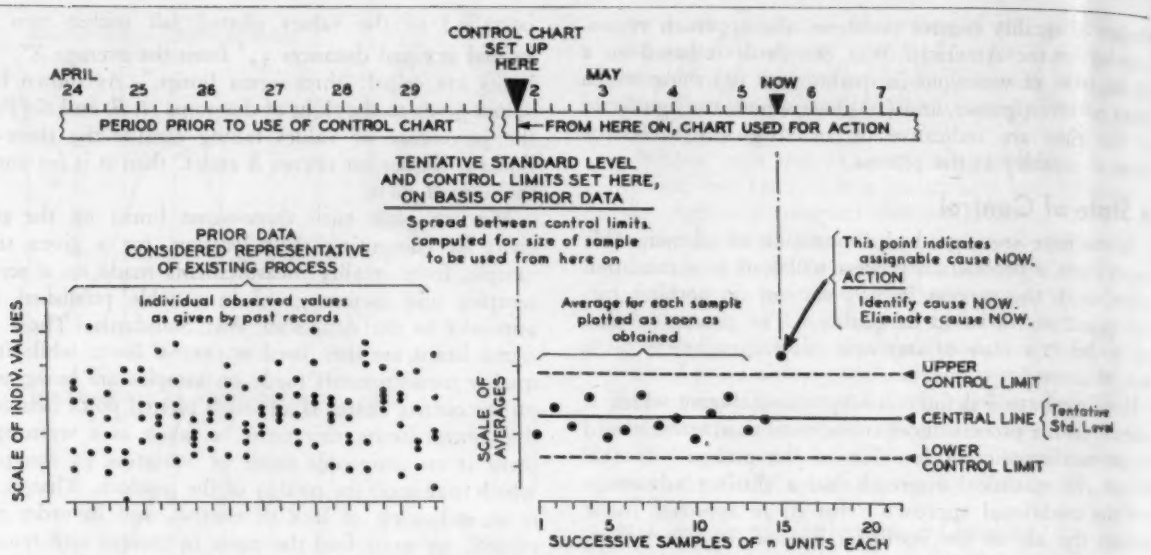
The practical use of the control chart for getting control of a process may be explained by reference to Fig. 2. Here it is assumed that before the chart was started, there was available a record of inspection data consisting of quality measurements made on units of product, and the times at which these measurements were made. From the records for the period from April 24 to 29, twenty-two subgroups of quality data, each consisting of five measurements made on individual units of product, were selected as being representative of the process to be controlled (left-hand side of Fig. 2). Four of such subgroups were taken for each day of the period under consideration. The average of the five measured values in each subgroup gives a picture of the level of quality in that subgroup.

Control Chart for Averages

Together, the 22 subgroup averages give a picture of the general level of quality of the product from which the 110 units were taken. To express this general level in a numerical value, the grand average of the 22 subgroup averages was computed and used to place a central line on the control chart (see right-hand portion of Fig. 2).

The dispersion of the 22 subgroup averages about their grand average is an indication of the degree of uniformity of quality in the product made during the period from April 24 to 29. Therefore, control limits applying to samples of n units each were computed from the grand average and factors given in a table and placed on the control chart above and below the central line (see "upper control limit" and "lower control limit," in Fig. 2).

The control chart for averages thus set up was used from May 2, on, as follows. Samples of n units were taken at regular intervals while the product was being made and the average quality of each sample was plotted on the chart as soon as possible after it had been inspected. So long as points thus plotted remain within the control


CONTROL CHART SET UP HERE
APRIL
24 25 26 27 28 29
MAY
2 3 4 5
NOW
6 7
PERIOD PRIOR TO USE OF CONTROL CHART
FROM HERE ON, CHART USED FOR ACTION
TENTATIVE STANDARD LEVEL AND CONTROL LIMITS SET HERE, ON BASIS OF PRIOR DATA
Spread between control limits computed for size of sample to be used from here on
PRIOR DATA CONSIDERED REPRESENTATIVE OF EXISTING PROCESS
Individual observed values as given by past records
SCALE OF INDIV. VALUES
Average for each sample plotted as soon as obtained
SCALE OF AVERAGES
This point indicates assignable cause NOW.
ACTION
Identify cause NOW. Eliminate cause NOW.
UPPER CONTROL LIMIT
CENTRAL LINE
Tentative Std. Level
LOWER CONTROL LIMIT
1 5 10 15 20
SUCCESSIVE SAMPLES OF n UNITS EACH

■ Fig. 2 – How inspection data taken from records of past production are used for setting up a control chart for getting control of the process while production is going on

limits, this means that there is no indication of lack of control in the process. However, if a point falls outside the control limits (like that plotted for sample No. 15 on May 5), this is a warning that there appears to be a cause of trouble (assignable cause). Therefore, we must investigate the process, try to find the trouble and remove it, if possible. A point outside the control limits is an alarm signal to take action. For this reason, control limits are also called *action limits*.

■ Revision of Control Limits

It is true that the control limits placed on the chart on May 2 are not based on quality data representing conditions in the process *at that time*, but since the basic data were recorded as recently as during the last part of April, they may be assumed to give a fairly approximate picture of the process early in May. In general, the first set of control limits on the chart should be based on the most recent quality measurements available.

Once the control chart has been started, the control limits should be periodically recomputed to be sure that they represent conditions in the process as they existed at a recent date. If the new limits are found to differ from those originally adopted, the latter are replaced by the new ones. (It will be noted that in Fig. 2 the central line is designated as a *tentative* standard level).

When all assignable causes have been eliminated from the process, the plotted points will be expected to remain within the control limits. Practice has shown that when at least 25 consecutive points fall inside the control limits, it may be assumed that a state of control has been attained. However, the plotting of data will be continued since new causes of trouble may sneak into the process again. While control may have been established, the problem of keeping it up is a continuous one. However, so long as control exists, it may be safe to extend the period of time elapsing between taking two consecutive samples. Evidence of lack of control may lead again to a shortening of this period.

■ Judgment of Lack of Control

In addition to being used for getting control of the process while production is going on, a control chart may be used also as an aid in judging whether control exists. We may be interested to find this out before starting a chart for process control. In the example of Fig. 2, we could decide, if we wished, whether or not there was control during the period from April 24 to 29, by plotting the subgroup averages computed for that period, against their three-sigma limits placed above and below their grand average (the central line in the right-hand portion of Fig. 2).

■ Other Kinds of Control Charts

So far, we have considered solely the use of a control chart for averages meant to keep track of the *level* of quality. In some cases, this is sufficient for the practical purpose. In general, however, we shall also have to keep track of *dispersion*, the other main characteristic of collective quality. This can be done by keeping, in addition to a control chart for averages, either a chart for ranges or a chart for standard deviations measured on samples.

Still another kind of control chart must be used where collective quality is measured in terms of the proportion between the number of acceptable units and the number of nonacceptable units in a given volume of product. Such a comparison is given, for example, by the fraction defective p, which is the ratio of the number of defective units to the total number under consideration; the per cent defective $100p$; or the number of defectives pn in a sample of n units.

There are also cases where it is necessary to use a control chart for the number of defects c in a sample of stated size, such as the number of flaws observed in a painted panel of given dimensions.

The procedure followed in setting up and keeping these various kinds of control charts is essentially the same as that described for a control chart for averages. What kind of chart or charts should be used depends on the requirements of the production problem under consideration. The American War Standards on Quality Control give advice in this respect and also present illustrative examples of typical problems that may arise in practice.

The Development of the MERLIN ENGINE

by J. E. ELLOR

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THE engine powerplant, with its installation, has always been somewhat of a nightmare to the airplane designer, and is still regarded by him as a necessary evil. There is no relaxation, however, in his entreaties to the engine manufacturers to furnish greater power outputs, at reduced weights if possible, and preferably without increase in frontal area.

Such requests are usually accompanied by a plea for the maintenance of these higher powers, from sea level to astronomical altitudes, together with a pressing invitation to do something about slaking the unsatiable thirst of the engine.

As an inducement to the power unit engineers in their endeavor to squeeze the proverbial quart out of the pint pot, it is not unusual for aerodynamic experts to produce curves indicating the very substantial gains to be obtained in top speed, rate of climb, range, and load carried, if only they could have improved engine performance.

It is, however, not necessary to preach to the converted, as the desirability and importance of producing higher performance engines of low weight and drag have long been appreciated by design and development engineers.

One has only to look back over the past few years, in order to realize the tremendous advance already made in airplane power units, which, in conjunction with the skill of the airplane designer, and the researches of the aerodynamic experts, has furnished the Allied air forces with planes superior to those of the enemy in very great numbers and at the right moment.

What is perhaps not fully appreciated, is the diversity of the problems and their solutions which lie behind each major improvement, and the fact that succeeding improvements are rendered more difficult, since they call for more careful research and experimentation to increase the store of knowledge of the physical properties and characteristics of available material.

Means must be found to overcome the prevailing limitations in mechanical operations and in the physical and heat-resisting properties of materials, and improved methods must be found for the treatment and processing of fuels, with a view to raising the detonation characteristics.

Further, during the search for, and application of, improvements, the engineers must always have in mind the fact, when considering the introduction of anything new, that it must be capable of being produced in quantity and there must be no sacrifice in the reliability of operation.

When considering the latest aero engines now being produced by manufacturers of repute, it is significant to note that the progress has centered around their original design and this supports the general contention that evolutionary rather than revolutionary development produces the best results, in the shortest period of time. Expressing this in a way very familiar to most engineers: "It is better to work with the devil you know, than the devil you don't know."

It is also rather significant to observe that the various firms who have had long experiences with either aircooled radial or liquid-cooled inline type engines, adhere religiously to the development of their own particular type. This, I am sure, will continue until one or the other, or both, ceases to make parallel progress.

Generally speaking, the trend in improving performance is to obtain the maximum power at sea level by means of supercharging, and to maintain the induction pipe pressure to the highest possible altitude, by progressively speeding up the supercharger as the altitude increases.

If there were no limits to the degree of supercharging, it is obvious that very high power outputs could be obtained from engines of small bore having the minimum number of cylinders. Since, however, detonation imposes a limit to the permissible charge density, the general line

ALTHOUGH the first Merlin engine was developed only a little over 10 years ago, its design was based on the Rolls-Royce engines that had already been successfully built, including the widely used Kestrel engine – the present-day Merlin being a direct scale-up of this engine.

Improvements during these years have taken the form of such changes as: alternate supercharger and airscrew reduction gear ratios; stepping up power ratings and supercharger performance; and improved cylinder cooling by changing over from pure glycol to pressure cooling systems.

The battle of Britain was responsible for some of the improvements in supercharger performance. Increased supercharger pressure raised the sea level power of the Merlin by 40%, giving the pilots of the Spitfires the advantages over the German flyers.

Further developments of the engine, many of them discussed here by Mr. Ellor, have made it possible for the RAF to retain this superiority.

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in the development to increase specific output has been first to obtain the maximum output from existing size cylinders up to the detonation limitation of available fuels, then to fit either more cylinders or larger cylinders.

Increasing the number and size of cylinders furnishes a direct increase in power output at any given altitude, but this is accompanied by a proportionate increase in weight and size.

■ Effect of Increasing Bmep's

The most fruitful field for exploitation is, therefore, to be found in raising the mean effective pressures on existing engines, as in so doing, higher outputs can be obtained with relatively small increases in weight without increasing the general overall dimensions. It is, indeed, a well-known fact that the greatest advance in pushing up engine performance has been due to progressively raising the mep by means of supercharging, and overcoming the physical and mechanical limitations as they arise by extensive and continued progress in fuels, materials, and design technique.

Fifteen years ago, mean effective pressures of 150 psi from engines having a weight/power ratio of 1.8 were considered good, whereas today on virtually the same engine, the maximum mep's attained have increased to between two and three times, and the maximum weight/power ratios reduced to less than one-half the former figure.

The rate of progress in this direction has, of course, been spread over a period of years, and a plot of mean effective pressure against time gives approximately a straight line, but showing an increase in slope for the past two or three years. This would appear to indicate that a continued advance towards higher powers, and lower weight/power ratios might be expected in the future on existing engines. The factors influencing the limitations to increased mean effective pressures, dealt with in the order of importance, are as follows:

1. *The detonation characteristics of the fuel, which determine the limiting pressure and temperature of the charge supplied to the cylinders.*

For a given cylinder compression ratio, the mean effective pressure is directly proportional to the charge pressure and varies approximately inversely as the charge temperature. With any one grade of fuel, the permissible charge pressure can be increased by correspondingly reducing the charge temperature without changing the detonation conditions.

It is, therefore, very important to have as low a charge temperature as possible, since its effect in increasing the mep is twofold, namely, permissible charge pressure is increased for the same degree of detonation, while the weight of charge is further increased by virtue of the lower temperature.

In passing, it should be noted that advantage can be taken of an increased knock rating fuel, partly to raise the manifold pressures, and partly to increase the cylinder compression ratio, if it is desired to bring down the specific fuel consumption.

It should, however, be borne in mind that the only virtue of a higher grade fuel lies in the raising of the detonation characteristics, as the energy value per gallon remains unchanged.

Rapid strides have and are being made by the petroleum companies in their research to provide higher grade fuels, and the author would like to pay tribute to the good work they are doing.

The reduction of the high charge temperatures generated during compression in the supercharger, can be effected by inter- or after-coolers, although a material drop of temperature can also be obtained by injecting liquid such as water into the charge, thus dropping its temperature by an amount equivalent to the heat required to evaporate this injected liquid.

There is also an added advantage to water injection in that the chemical reaction of the steam on its disassociation products promotes antiknock activity. While this is an efficient way in which to increase engine output from available fuels, the system required to operate it becomes very involved, and the additional installation weight could, perhaps, be more usefully exploited by increasing the amount of after-cooling and resorting to the use of higher octane fuels.

2. *The second factor is the strength of mechanical parts to withstand the higher loads transmitted as the mep's are raised.*

These limitations are usually shown up by failure during development testing, which necessitates the stiffening up of parts, changing to materials having improved strength and fatigue properties, and the processing of steels to furnish harder rubbing surfaces to improve wear characteristics. Bearing troubles with both frictionless and journal types have certainly not been conspicuous by their absence during the process of working up to higher mep's. Improvements in journal bearing materials from white metal to lead bronze, and now to silver lead, have contributed very largely in making the present power outputs possible.

Better heat-resisting materials and improved design technique have helped the exhaust valves and seats to withstand the greater heat in the exhaust gases discharged from the cylinders, while improved design in manufacturing technique applied to the airscrew reduction gear teeth have considerably raised the permissible loading. In this connection, extensive testing has shown that certain EP lubricating oils help appreciably in raising the power loading of gear teeth without adversely affecting other engine parts. The introduction of such an oil would be a very effective way of reducing the weight and size of a gear combination for any given power output.

These are some of the main items which have to be attended to in the struggle to maintain reliable operation as the mep's are increased, but the increase in weight per horsepower gained is surprisingly small, while the change in overall dimensions is approximately nil.

3. *The third factor is the ignition system.*

Although it is one of the smallest parts of the engine, the spark plug has one of the toughest jobs to do. Very little assistance is provided in getting rid of the increased amount of heat with increase in power on account of the electrical insulation. Thus, we find that with each marked step-up in mep, an improved spark plug has to be developed, possessing a higher heat factor, but still retaining the nonfouling characteristics when the engine is slow running.

The magneto also comes into the picture to a somewhat lesser degree, in that higher voltages are required for the spark to bridge the plug gaps against the higher charge density in the cylinders. This increases the tendency for electrical leakage in the magneto, particularly at high alti-

tudes where the surrounding density is low. To overcome this trouble, it becomes necessary either to redesign the unit providing increased leakage paths and/or to supercharge the magneto.

4. *The fourth and last factor to which reference will be made is the cylinder cooling limitation and the mechanical loss.*

In any particular engine, the quantity of heat rejected to the cylinder walls increases with the weight of charge inhaled, but not in direct proportion. The rate of increase for a liquid-cooled engine follows the approximate law: "weight of charge, raised to the power of 2/3."

Due to the elastic control of the cooling medium, experiments have shown that very high mep's can be obtained from liquid-cooled engines, without raising metal temperatures beyond the desirable limits. One very important factor is to maintain a reasonable cylinder-liner temperature to facilitate the dissipation of the additional heat absorbed by the piston, and the provision of a uniform flow of cooling medium across the cylinder heads, eliminating blind pockets.

The use of water as a cooling medium, operating under pressure to elevate the boiling temperature, definitely maintains a lower wall temperature than pure glycol and therefore has distinct advantages, particularly in the higher mep's.

As regards mechanical frictional losses, these increase at a very low rate with increase in mep at the same piston speed, while the cylinder pumping losses actually change from negative to positive values.

In fact with the higher manifold pressures it is possible to exceed 100% mechanical efficiency, in which case the charge pressure difference across the pistons during the suction stroke is more than sufficient to motor the engine.

Taking into account the power to drive the supercharger and expressing the total losses in terms of the available airscrew shaft horsepower, the overall mechanical efficiency with the engine-driven supercharger remains approximately constant.

In the foregoing paragraphs the author has confined his remarks specifically to the raising of powers by increasing the mep, as the history of Merlin development has been in this direction.

Early Developments

Before passing on to the actual development of the Merlin, it will be helpful to survey briefly the preceding types of Rolls-Royce aero engines on which the real spade work was done, thus enabling the first Merlin to get off the mark with such a good start.

Rolls-Royce interest in aero engines dates back to World War I and the years immediately following it, when the late Sir Henry Royce designed the well-known Eagle, Falcon, and Condor series. These were all naturally aspirated engines and followed the same basic design, having 12 separate composite, steel jacketed, liquid-cooled cylinders disposed at 60 deg in two banks. The powers, however, were relatively low compared with present-day outputs.

It will, of course, be remembered that the first direct North Atlantic crossing was made by a Vickers Vimy aircraft powered by two Eagle engines. It is also interesting to note that the airscrew reduction gears were of the epicyclic type and on one occasion, advantage was taken of the facilities offered by this design to experiment with a two-speed gear in order to increase the power available on climb by increasing the rpm of the fixed-pitch airscrew used in those days. The pilot was thrilled with the improved performance of the airplane both on climb and in level flight particularly, as he had been informed not to expect any speed increase under the latter conditions. When asked if he had changed gear before going into level speed, he replied, "Oh, no! I obtained much better results by keeping it in the high gear."

The introduction of the constant-speed airscrew has, however, robbed the two-speed gear of its principal virtue, but with the higher powers of today and the limitations of airscrew blade characteristics, it still possesses certain advantages, provided the increased weight can be kept down.

Inspired by the superior performance obtained from the Fairey Fox, an airplane of advanced design possessing low drag characteristics and powered by a Curtiss D-12 engine developing approximately 500 hp at sea level, the late Sir Henry Royce designed a new engine later known as the Rolls-Royce Kestrel. This was in 1926, and the principal dimensions of the present-day Merlin are a direct scale-up of the Kestrel engine.

The leading dimensions of the Kestrel were:

Bore and Stroke, in.	5 x 5½
Capacity, cu in.	1296
Total Piston Area, sq in.	236
Weight	
Naturally Aspirated Version, lb	890
Supercharged Version, lb	960

The first series was naturally aspirated and supplied with compression ratios of 6, 7, and 8:1. The two cylinder banks were monobloc type with the heads cast integral with the skirts and secured to the crankcase by long studs.

Wet liners were employed, designed for easy removal.

Such design, however, required great care in the machining and assembly in order to ensure tight joints between the liner and the head, to prevent leakage between gas and water compartments. This, by the way, was no easy matter and required great skill and expert men on assembly. Even with the greatest care, top joint leaks were one of our greatest troubles, together with the barrel distortion on account of the holding-down loads being transmitted via the top joints and through the liner.

This trouble was entirely eliminated at a later date by introducing a two-piece block which separated the head from the skirts and which was designed for the holding-down loads to be taken through the walls of the castings instead of through the steel liners.

This construction was introduced on 2-speed Merlins.

Straight spur-type airscrew reduction gears were adopted in preference to the epicyclic, in view of their simplicity and greater capacity to take increased loads. This change was also influenced by the fact that the airplane designers required a higher thrust line to permit larger diameter airscrews to be fitted to single-engined planes without unduly increasing the depth of the undercarriage.

In the very early days of the Kestrel development, a single-speed, single-stage gear-driven centrifugal supercharger was applied. This enabled the engine to maintain 480 bhp up to an altitude of approximately 15,000 ft. A specially designed Rolls-Royce carburetor was located at the supercharger inlet to take advantage of the heat of vaporization of the fuel during its passage through the impeller and to raise the supercharger compression ratio.

This arrangement added to the compactness of the design and facilitated installation and controls.

To prevent the pilot from opening his throttle too wide at the lower altitudes, thus safeguarding possible engine failure, by exceeding the permissible powers, an automatic boost regulator was developed and fitted. This consisted of an aneroid-operated relay valve which connected either side of a servo piston to the pressure or suction side of the supercharger. The actuating force due to this pressure difference on the two sides of the piston operated through a differential gear onto the carburetor throttles. Thus the pilot could move his throttle lever to the wide-open position and the boost control would adjust the actual opening of the carburetor throttle to maintain a constant, predetermined, boost pressure covered by the aneroid setting. The same design of unit is used on the Merlin engine of today.

A vast amount of concentrated effort lies behind the development of the Kestrel engine, covering many thousands of hours of laboratory and flight testing.

During this period, many changes of cylinder designs were investigated – alternative connecting rods – intensive valve and seat investigations – piston designs – and main and big end bearings. In addition to the work carried out on the multicylinder engines, a battery of single cylinders was busy on fuel, piston, and spark-plug research. Another group of technicians concentrated on the best means of cooling the engine and developing simplified and reliable systems. These included water cooling, evaporative cooling, pure glycol cooling and later pressure water cooling, which is now used exclusively on all the Rolls-Royce engines. Thirty per cent glycol is added, however, to prevent freezing while stationary on the ground.

A considerable amount of work was done on evaporative cooling both on the engine and in airplanes using the wing surfaces as well as standard honey-comb radiators to condense the steam. Its chief advantage lay in the fact that less water was carried, cooling resistance reduced and condenser leaks were not quite so serious on account of the relative volumes of steam and liquid. Its general adoption, however, was discontinued because of the complicated system, but from its development we obtained an efficient combined header tank and separator and a water circulating pump possessing good characteristics very essential for pressure cooling.

In parallel with the actual engine development, wind tunnel and full-scale flight experiments were carried out at our own powerplant and flying section. Here engines were submitted to aerial-type tests and intensive flight and laboratory tests were conducted on radiators and low drag cowlings. For many years, we have concentrated a large staff on installation problems located in a special factory and flying establishment some 18 miles away from the main plant.

This department is equipped with drawing offices, work shops, dynamometer test rigs, and a flight section, where we have had as many as 12 to 15 airplanes in operation at one time in connection with powerplant development.

To enable differences in drag to be observed more accurately, a Heinkel He 70 was purchased because of its low drag characteristics and spacious fuselage rendering it particularly suitable as a flying test bed. This plane had a Kestrel engine installed and did very useful work right up to the commencement of the present war, when, for obvious reasons, it had to be grounded. To obtain actual figures on drag losses on cooling systems, airplanes were flown without any provision for external cooling. To obtain basic comparisons with radiators and cowls of different sizes and location on the fuselage, a large tank of water carried in the fuselage was connected to the engine circulating system and allowed to boil off.

This permitted a flight period of about 40 min to take a series of speed observations. Variable-pitch airscrews and blade characteristics most suitable for our engines were developed at this establishment with a view to selling maximum thrust rather than just horsepower.

■ Interchangeable Powerplant Developed

Intensive work was also done on air inlet and exhaust systems, which led to the general introduction of the utilization of the energy in the airstream to add boost to the engine and to increase the thrust due to the ejector effect of the exhaust. All this progressive development and close coordination of energy on every single angle of the installation problem led to the development of the interchangeable powerplant now supplied by Rolls-Royce.

This policy was formulated by the British Air Ministry's directive that alternative powerplants should be available for all new military types of airplanes. Experience has shown that not only is this very desirable, but that it also works out to be very practical and useful, particularly in the case where the same powerplant is used on multi-engined planes of different design.

Even on single-engined fighters there does not appear to be any reason why the most compact engine mounting, cowling, and the like, should not be commonly used to facilitate servicing in the field.

Very important data were accumulated and later applied to the Merlin engine by the rapid development of the R-type engine designed specifically for powering the Supermarine Schneider trophy seaplane. These engines were later used to power various record-breaking boats and world land speed record cars, some of which have been operated in this country.

This engine was also a direct scale-up of the Kestrel, but in extracting the very high powers by raising mep's and piston speeds, there was scarcely a part we did not break during its development. It can be truly said of this R-type engine that the factors of safety on all parts were certainly uniform but very near to unity, thus limiting the operation at high powers to a very short life.

The bore and stroke were 6 x 6.6 in. and the total displacement 2240 cu in. The maximum power developed was 2783 bhp at 3400 rpm requiring a boost pressure of 73 in. of mercury, absolute. A maximum mep of 310 psi was recorded at 3000 rpm with a boost pressure of 67 in. of mercury and the power was 2600 bhp. The power to drive the supercharger was 360 bhp at 3000 rpm and 450 hp at 3400 rpm and the corresponding crankshaft horsepowers were 3333 and 2960, respectively. The maximum crankshaft mep at 3000 rpm was 355 psi.

Special fuels were used having a high latent heat to bring down the charge temperature and the octane values were approximately 92. The fuel constituents were 20% gasoline, 70% benzol, 10% methanol, with 4 cc of lead added per gallon.

With this background and all the development facilities, the Merlin engine commenced its career.

As previously stated, the design follows the Kestrel and R engines very closely. The bore and stroke of 5.4 x 6 in.,

giving a total capacity of 1650 cu in., were chosen to furnish an output of 1000 bhp at the moderate manifold pressure of 42 in. of mercury, absolute. The maximum power ratings of the first Merlins were 880 bhp for take-off and 1020 bhp at 15,000 ft altitude obtained from a single-stage, single-speed, gear-driven centrifugal-type supercharger, operating at a rotor tip speed of 1150 fps. Pure glycol was used as a cooling medium with the maximum temperature limited to 120 C, and the best grade of fuel obtainable at that time was 87 octane.

A special float-type carburetor was developed for this engine in conjunction with the S. U. Co.

This had direct needle jet controls operated by bellows which automatically corrected the mixture strength with increasing altitude and enrichened the mixture above predetermined boost pressures to permit maximum output without detonation.

The engine was also fitted with an automatic boost regulator scaled up from the Kestrel unit.

It was a little over 10 years ago when the first Merlin was designed, and up to date there are 77 different mark numbers. A new engine mark number is allocated when changes are made to power ratings, interchangeability of parts, and when installation in aircraft is affected. Then, as today, a feeling of relief and gratification was experienced when a new mark of engine passed a type test. The troubles and disappointments encountered during previous months of testing and redesigning are temporarily forgotten. This state of bliss is not permitted to last very long, however, as the flight tests which follow will certainly find other troubles to cause many headaches and sleepless nights. Fortunately, we had the Hucknall flight establishment where the Merlin flight developments were carried out in parallel with the dynamometer investigations, and the value of such a department cannot be too highly stressed as a means of ascertaining and correcting engine faults quickly.

Further, one of the principal delays in getting powerplants to operate in new planes is the lack of installational system data and experience. It is no secret that even today airplanes spend far too much time on the ground because of installation troubles, most of which can be eliminated if the same care and attention is given to the design and development of such items as cooling, oil and fuel systems, pipe lines, connections, mechanical and automatic controls as is given to the engine and airplane. Also, as the reliable operation of the engine depends on the satisfactory functioning of these systems, it is the author's opinion, based upon long experience, that the best results are obtained by making the powerplant the engine manufacturer's responsibility in collaboration with the airplane designers and accessory manufacturers.

Among the advantages derived are that the most suitable systems are evolved for the particular type of engine, while the latter is being developed for the different types of planes. The coordinated data can then be at the disposal of all parties and will, undoubtedly, reduce the number of different layouts in the same type of airplane and will enable the latest developments and improvements towards reliability to be incorporated.

Working upwards from the Merlin 2 to the Merlin 46, the main changes comprised:

Alternative supercharger and airscrew reduction gear ratios.

Stepping up power ratings and supercharger performance.

Improving cylinder cooling by changing over from pure glycol to pressure cooling systems.

The introduction of a two-speed drive to the single-stage supercharger in order to raise the full-throttle altitude, at the same time increasing the power available at the lower altitudes.

During this period the grade of fuel had been increased from 87 to 100 octane, the knock rating having been raised by progressive increments of tel.

The Merlin X engine was the first of the series to be fitted with a two-speed supercharger.

■ Exhaust Valve Corrosion

Exhaust valve troubles resulting from the introduction of leaded fuels and high ratings are well known to aero engine manufacturers.

The story of the prolonged investigation carried out by Rolls-Royce into the cause and prevention of lead attack is a long one, but in view of this work we consider that corrosion of exhaust poppet valves will not present any obstacle to the increasing of mep's for some time to come. The standard Merlin exhaust valves are sodium cooled in the stem only and are fully coated over the face and head with Brightray. This is an alloy composed of 80% nickel and 20% chromium, which possesses sufficient ductility to prevent cracking as a result of the head flexing and resists the formation of lead oxide deposits, thus eliminating the preignition experienced with noncoated valves.

Experience has shown us that in-line monobloc cylinder construction gives more trouble from lead attack of exhaust valves, than the separate-cylinder arrangement. This is presumably due to the valve-insert distortion being aggravated by the thermal expansion effect in adjacent cylinders, and one of the essentials of a successful monobloc design is to isolate the cylinder heads from one another as far as possible.

■ Cylinder Heads

In the early days of the development of the Kestrel blocks, the cylinder heads were connected solidly by metal without any provision for expansion, and considerable trouble arose from head and valve-seat distortion, until we separated them by saw-cutting the metal between each head. Subsequent designs provided for expansion in the casting, and thermostats were introduced in the cooling system to prevent cyclic changes and to maintain a more uniform operating temperature.

In spite of this, distortion was not entirely eliminated, and at one time we ran into a serious epidemic of the engine cutting out on take-off due to blowback, igniting the charge in the induction system. We then decided to take the bull by the horns and fit flame traps at the entry to the inlet ports. This cure was very effective and took care not only of any little valve-seat distortions but also of any preigniting plugs.

While on the subject of cylinder heads, it is of interest to note that after experimenting with many different types of combustion chambers, we have not been able to surpass the performance obtained from the four-valve Merlin shape. It seems to possess very good characteristics for

operating at high gas pressures and temperatures within the detonation limits of any given fuel, and gives peak power at a high piston speed.

Piston and ring development is another long story entailing many design changes in order to cope wtih the higher loads and heat stresses, also in the case of rings where blowby and heat transfer problems had to be solved. In the latter case, improved materials and treatment during manufacture contributed mainly to the present satisfactory rings.

Due to the well-balanced design of the crankshaft, we have experienced no trouble whatever with this unit in any of the scale-ups, and they are particularly free from any vibration trouble without the aid of dampers or dynamic balancers.

During the Kestrel development, we switched over to hardened nitrided shafts, which at that time helped us with the bearing problems. Main-bearing problems have been continuous through the Merlin engine development, and until recently all our efforts were concentrated on the research into the development of lead bronze alloys with additives of tin, nickel, and small percentages of other elements. At the same time, improved means of lubricating bearings have been employed.

It will, of course, be realized that in obtaining greatly increased powers from the same engine, the bearing areas cannot be changed and the loads, therefore, increase in direct proportion to the power. Here, again, the greatest advance has been contributed by improved bearing material and by the use of lead-lined silver-backed shells, and we are able to cope with very high loadings per unit area.

As regards frictionless bearings, the increased loadings reflect mostly on the reduction gear bearings, especially the two sets of roller races supporting the main gear. Since the size could not be increased, improvements had to be made in technique, and in this connection the Bower Roller Bearing Co. in Detroit has made valuable contributions.

■ Supercharger Design

With regard to progressive engine development, we conducted intensive supercharger investigations to supply higher charge pressures with increased efficiencies. These followed on the lines of increasing impeller tip speeds, improved impeller and diffuser characteristics, and more efficient air entry conditions to the supercharger. The latter improvement alone accounted for 3000-ft altitude.

You would, perhaps, like to ask the question of why the Merlin engine is fitted exclusively with a mechanically driven supercharger. The answer is that we decided to concentrate our efforts on getting the most out of this type in view of the fact that the engine was compact and complete as a power unit and offered a more simplified installation compared with the exhaust turbo-driven system. This applies particularly in the case of pursuit aircraft.

We have shown that charge pressures can be maintained to very high altitudes with the mechanically driven supercharger at high efficiencies. Also, effective use is made of the exhaust energy in the ejector-type manifolds, which are a development of the Rolls-Royce company. Instead of using the exhaust energy to drive the turbosupercharger, it is employed in the form of jet propulsion.

There are two other advantages, in that the shaft horsepower of the engine is increased owing to the lower exhaust back pressure, and the problem of flame damping is facilitated.

The Merlin XX engine has a two-speed supercharger and was a development of the Merlin X.

The continued demand for higher military powers from the Merlin at all altitudes, and the fact that experience has shown that the economic limits have been reached on a supercharger having a single impeller, led to the development of the two-speed, two-stage, supercharged Merlin engine introducing a charge cooler.

The author now proposes to give a condensed version of a Rolls-Royce publication which deals with the evolution and development of the two-stage, charge cooled Merlin engine, introduced as the Merlin 61 series.

It is well known that at the beginning of the war and during the battle of Britain, every RAF first-line fighter aircraft was fitted with a Rolls-Royce Merlin III engine, and the complete defeat of the German air force in August and September of 1940 definitely established the technical superiority of our engines. This superiority was not obtained by chance, but every move of the enemy had been anticipated and a definite counter worked out.

For example, early in the war German aircraft had resorted to low flying tactics and on account of this we immediately increased the sea level power of the Merlin engine by 40% by an increase in the supercharger pressure. This move so improved the performance of the Spitfire at low altitude, that German aircraft were forced to fly higher, and throughout the battle of Britain there was a noticeable tendency for the German Me 109's to go higher and higher into the substratosphere in order to try and escape from our fighters.

It seemed at this stage that the German aircraft had an advantage owing to their smaller dimensions and lighter weight, but, fortunately, we had ready for production a new supercharger giving more power at higher altitudes and we were able to introduce the Merlin 45 and the Merlin XX engines into the Spitfire and Hurricane, respectively.

These increases in engine power output were achieved without any radical change to the aircraft and the flow of fighters from the factories was not affected in the slightest by these modifications. A continuous supply to the RAF of improved fighters was maintained as the basic engine remained unaltered, except that new superchargers had to be manufactured in large quantities.

The war demands the performance of all types of military aircraft, and particularly that of fighter aircraft, shall continually improve. Ranking above the need for more and more aircraft of all types, is the overriding necessity that our aircraft shall have technical superiority over those of the enemy.

It soon became apparent that in the Spitfire we had a supreme aircraft from the aerodynamic and military points of view, and all that was required to keep this airplane on the top of the list was a steady improvement in the performance of the Merlin engine with which it was fitted. The advantage to the RAF was that any improvement obtained could be immediately applied to existing aircraft and could be put into service in the minimum of time.

With these facts in mind, Rolls-Royce has continually striven to increase the power output of the Merlin engine by improvement to the supercharger and carburetor, and

the course of this development has led to the adoption of the two-stage supercharger.

Factors Affecting Power Output

It will be appreciated that the horsepower which an engine can develop is determined by the amount of air and gasoline which it can be made to consume, and not necessarily by the size or number of the cylinders which make up its physical shape. Summarizing, we can say that the power output of an engine depends upon three main factors:

(a) The ability of the engine to withstand with reliability the stresses involved.

(b) The ability of the supercharger to force into the engine the weight of charge necessary to produce the horsepower.

(c) The limiting mep's imposed by the detonation characteristics of the fuel.

In the case of the Merlin, condition (a) has been established as the result of long years of experimentation and of actual service under wartime conditions with the RAF. There remains, therefore, the problem of producing a supercharger of high efficiency having unusual compression ratios and a large capacity to deal with increased quantities of air. This consideration of high compression with large quantities of air introduces the further problem of cooling the charge before it enters the cylinders, as this is necessary, firstly, in order to maintain the weight of charge, and secondly in order to reduce the charge temperature to furnish the additional advantage of raising the detonation limit with available fuels, thereby permitting higher boost pressures to be used.

In order to carry out this requirement a water-cooled aftercooler has been designed for the Merlin engine.

Until quite recently all aero engines were fitted with superchargers of the single-stage variety, that is to say, the superchargers are of the rotor or fan type rotated at very high speed either by the engine or by an exhaust turbine.

To increase compression by the supercharger, it is necessary to drive the rotor faster, but such increase is limited because of mechanical considerations. On the Merlin XX supercharger, rotors had already reached the tremendous speed of 29,000 revolutions a minute, and it is neither a practical nor an economical proposition to increase the speed further.

In order to overcome this limitation, a new system of supercharging had to be evolved if further progress was to be made with the Spitfire and the Merlin engine. This was accomplished by using two rotors or stages, placed in series, in which the charge is first compressed and then passes on and is compressed again in the second stage. The resulting high temperature had to be reduced by means of a water-cooled aftercooler. The task of designing this complex system onto a Merlin engine so that the combination would still fit onto the small space available on Spitfire aircraft was a big one, but it was successfully overcome and resulted in the Merlin 61 series of engines.

Let us review the advantages of this system by comparing aero engines powering the various first-line aircraft of the nations engaged in this war. German engines, without exception, were then fitted with single-stage superchargers capable of maintaining ground level pressure in the engine induction system up to 20,000 ft. Merlin engines fitted with single-stage superchargers maintained the same pressure up to 30,000 ft, and until the advent of the Merlin 61 series, this increase in altitude had been the main means of our achieving technical superiority.

Certain American engines and installations are fitted with turbosuperchargers which also maintain sea level pressure up to 30,000 ft, and although this system is excellent when applied to bomber aircraft, there are reasons, however, which make it less suitable for fighter aircraft.

Finally, the two-stage supercharger maintains sea level induction pipe pressure up to 40,000 ft, at which point the charge is being compressed to six times the surrounding atmospheric pressure and the power of the original Merlin engine is about doubled.

The results when interpreted in the performance of the improved Spitfire more than fulfilled the hopes and expectations of all who helped in this work. Every aspect of the performance of this outstanding aircraft was tremendously improved by the introduction of the Merlin 61.

We have seen no indication that the Germans were following a similar course and in fact, the advent of the Focke-Wulf 190 showed that they had adopted the alternative of a larger machine powered by a larger engine.

So much for the high-altitude performance, but the opposite end of the scale must not be forgotten. The former offers by far the most difficult problem, but once achieved by the Rolls-Royce two-stage supercharger and intercooler, the advantages can be interpreted into performance at lower altitude by suitable adjustment of the supercharger gear ratios.

Details of Merlin 61

The following details of the Merlin 61 may be given:

The two-speed, two-stage supercharger has two rotors driven on a common shaft and is really two separate superchargers in series. The charge is drawn through the carburetor and is then compressed by the first stage supercharger and then delivered to the inlet of the second-stage supercharger. Here it is still further compressed and is finally delivered to the main induction pipe feeding the 12 cylinders which are still arranged in two banks of six as on all Merlin engines.

The manner in which this intercooling is carried out is worthy of note. A square, box-like structure is mounted in the induction system between the outlet of the second-stage supercharger and the rear of the cylinder blocks. This box contains a radiator through the tubes of which water is circulated. The hot charge from the outlet of the supercharger is passed, through the matrix of this radiator, so that its temperature is considerably reduced.

In addition to this charge cooling radiator, there is a water jacket between the two stages of the supercharger which also contributes to the cooling of the charge. The intercooler system is entirely separate from the main engine cooling system, and one advantage of this method is that the actual radiator from which the excess charge temperature is dissipated to the atmosphere may be placed at any convenient point on the powerplant or airplane.

In the case of the improved Spitfire it is mounted under the wing, which contains also one of the main engine coolant radiators. The other engine cooling radiator is mounted in a similar duct under the opposite wing. This second duct contains, in addition, the engine oil cooler.

Another advantage of the liquid-cooled intercooler is that it can be made considerably smaller than would be the case if the heat interchange was made direct to the atmos-

phere. Since it is important that the intercooler should be mounted in such a position that the induction system is kept as short as possible, space limitations would probably mean that a sacrifice in intercooling efficiency would have to be made if direct aircooling of the charge was employed. This progressive development of the basic Merlin engine should do much to counteract any tendency to belittle the qualities of our military equipment and to exaggerate the good points of our adversaries' equipment.

Finally, I would like to say that the Merlin engine is the direct result of coordinated team work on the part of all sections of the Rolls-Royce company and to the wonderful energy, foresight, and understanding characteristics of E. Hives, the general works director. It is also due in a large measure to the great help and close cooperation given by the British Air Ministry.

I would also like to pay tribute to the Packard Motor Car Co., to George T. Christopher, its president, to Col. J. G. Vincent, and to the AAF for their wonderful work in connection with the Packard produced Merlin. Their labors have definitely contributed in effecting valuable improvements and I wish them every success in their continued efforts to satisfy the demands for still greater powers from the Merlin.

Particulars of the GERMAN VDM ELECTRIC PROPELLER

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has shown that very few projectile perforations of the pitch-change mechanisms occur. It would be easier to form the dome shell of the Hamilton hydromatic out of hardenable steel than enclose it in a heavy steel spinner possessing gyroscopic tendencies of a highly disagreeable nature.

No de-icing equipment was on the spinner examined and no locking key was furnished, a thin screwdriver worked easily.

■ Weights

Units	Weight of Assembly lb	Weight of Assembly oz	Weight of Propeller lb	Weight of Propeller oz
Hub	46	6	46	6
Blade Mounting	21	12	65	4
Blade	39	0	117	0
Gear Box	21	4	21	4
Spinner	8	8	8	8
Motor, Governor, Relay System	12	3	12	3
Total	....	....	270	7

Note: Missing installation parts would probably not weigh over 10-20 lb more. Total weight runs approximately 0.33 lb per horsepower.

■ Conclusion

The VDM propeller is versatile, well designed, and well constructed; however, it is not superior to comparable British and American units.

A pitch-change rate of 1 deg per sec is quite inadequate for a propeller installed on a high-performance aircraft. The Germans, realizing this, have installed a more rapid pitch-change mechanism on their latest model fitted to the Focke-Wulf 190 fighter. An inspection of the latest Focke-Wulf installation left the writer impressed by the ability of the VDM works to modify considerably their basic design when production is sorely needed.[2]

Service requirements are easily met in this propeller. A propeller bar, side cutters, hammer and drift, and a blade beam will enable a mechanic to remove the blades from the hub and the hub from the shaft. A screwdriver will suffice to make all adjustments to the motor and control system.

Lack of provision for adding grease to the blade bearings without disassembly is not such a good feature for a propeller whose makers claim 1000 hr of operation between major overhauls. Absence of any balancing means may indicate good manufacturing practices, but does not ensure the best of balance in subsequent overhaul without reducing weights of blades by filing.

Vibration characteristics of an early VDM propeller may be found in a paper by Karl Lürenbaum of the Deutschen Versuchsanstalt für Luftfahrt.[3] Here the VDM is referred to as the Hedder Copper Co. propeller and the HKW (Heddernheimer Kupferwerke) propeller.

The electric governor is greatly inferior to the finely built American governors which have been in use for almost ten years, with constant improvement. The relay is complicated and evidently not intended to be adjusted in the field. Pitch indicators are believed to be superfluous.

Production of the VDM must be in the high numbers because no part is particularly difficult to manufacture.[4] The great number of small parts are easily made, but are a nuisance for inspection and assembly. This was found in studying the propeller. Although magnaflux inspection of every steel part did not find a single bad indication, the process took three times as long as the same inspection of a Curtiss electric.

The VDM spinner is superior to American spinners, at least from the removal, installation standpoint. It is only a matter of seconds to remove or install the VDM, whereas American spinners take an infinitely longer time requiring wrench extensions and considerable effort on the part of the mechanic. Such mounting accomplishes nothing more in the way of support and loses a great deal in the matter of weight. That the VDM spinner has merit is demonstrated by the RAF's application of its principles to their own aircraft.

■ Acknowledgment

Sincere appreciation is extended to the British Air Ministry, the Enemy Aircraft Section of the Royal Aircraft Establishment, the Director of Scientific Research, and the propeller firms of Rotol and DeHavilland for supplying units and data for study.

The rest of the work was accomplished through the splendid cooperation of the engineering, photo, and propeller departments of Lockheed Overseas Corp.

[2] This change in design was necessary because of the addition of an engine-driven cooling fan to certain engine installations.

[3] See *SAE Journal*, Vol. 39, December, 1936, p. 475: "Vibration of Crankshaft-Propeller Systems," by Karl Lürenbaum.

[4] Since the preparation of this study on the VDM propeller, the VDM factory at Frankfurt, the whole industrial area of Hamburg, and the blade factory at Düren have been severely handled by bombers of the RAF and the Eighth Air Force. It is reasonable to suppose that production at these factories must be considerably disorganized or completely stopped.

STRUCTURAL FLIGHT RESEARCH

by **W. LAVERN HOWLAND**
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STRUCTURAL flight research is the one branch of aeronautical research that has really been neglected in the past. Aerodynamic research, both experimental and theoretical, has been carried on in a comprehensive manner for a good many years. Numerous wind tunnels throughout the world are engaged in the never-ending process of aerodynamic research and testing. Aeronautical literature is full of aerodynamic theory and elaborate calculations on the airplane's aerodynamic performance. Powerplant testing and research has been pursued on a large scale since World War I. Structural research, however, has been confined almost entirely to theory and two types of tests:

1. Static tests to determine whether or not the completed airplane could meet certain empirical or sometimes arbitrary loading conditions.

2. Static tests of individual parts of the airplane in a testing machine or a testing jig.

Upon surveying aeronautical literature one is impressed by the absence of articles on structural flight research. Undoubtedly, the primary reason for these lack of data are the difficulties involved in making structural measurements in flight.

The general purpose of structural flight research is to determine or measure the magnitude and frequency of various stresses or loads which the airplane structure is subjected to under flight or dynamic conditions. Many of the loads imposed on the airplane structure are only known by their overall effect on the airplane and have never been measured directly. A large percentage of all structural flight research performed to date has been done with accelerometers and not with strain gages. Gust loads, which govern the design strength of many parts on nearly every airplane, have been determined entirely from calculations based on acceleration measurements at or near the center of gravity of the airplane.

Unfortunately, it has never been proved experimentally or theoretically that there is a perfect correlation between the acceleration at the center of gravity of the plane and the stresses in the structure. As a matter of fact, the reverse seems to be the case, it can and has been proved theoretically[1, 2] that under dynamic conditions such as gusts or sudden pullups the stresses set up in the structure may be greater than or less than those which satisfy equilibrium conditions. That is to say a gust which produces an acceleration at the center of gravity of 2g does not necessarily produce a stress in an elastic wing supporting the airplane of exactly twice that of one load factor. The reason for the above can be easily visualized by realizing that the air forces and structure constitute an elastic system in which dynamic conditions are set up. Wing deflections as well as the rate at which aerodynamic forces can change cannot be neglected.

Many of the existing design load requirements for modern airplanes are based to a large extent on results obtained from acceleration measurements on airplanes of 10 years ago. These airplanes were not only considerably slower than those being designed today but much smaller. Other factors which could also produce important differences in the dynamic characteristics of the airplane's structure are: lower wing loadings, fewer powerplants, less power, lower allowable stresses, and a smaller amount of distributed load in the wing. How these factors will affect the modern airplane remain to be seen or determined by flight tests. Some of the loads imposed on the airplane structure are so complicated as to render an analytical analysis at the present time impossible. Such loads as maneuvering loads are very difficult to measure in a wind tunnel, if possible at all. This means that there are some uncertainties in basic design loads and particularly in those for large airplanes until accurate flight measurements are made.

For several years the author was engaged in testing and developing higher strength and more efficient compression panels. During a period of only six years the average al-

THE general history and object of structural flight research is discussed here by Mr. Howland, who compares the importance and future application of this type of testing with static testing and stress analysis.

The author also describes the adaptation of wire resistance strain gages and the equipment used in making flight measurements. He gives a report on a few of the many quantities that have been successfully measured, and discusses some of the difficulties and pitfalls to be encountered.

A stress warning device that was developed for use in flight testing is also described in detail.

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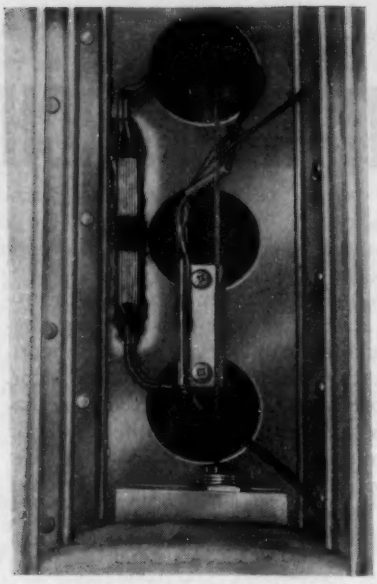
[1] See *Journal of the Aeronautical Sciences*, Vol. 9, December, 1941, pp. 64-67: "On the Reaction of an Elastic Wing to Vertical Gusts," by William R. Sears and Brian O. Sparks.

[2] See NACA Technical Memorandum No. 654, January, 1932: "Stresses Produced in Airplane Wings by Gusts," by Hans Georg Kussner.



■ Fig. 1 – Strain gages installed on front spar of P-38 stabilizer

lowable compression stress of stiffened aircraft panels doubled, that is, went from below 20,000 psi to over 40,000 psi. This increase can be attributed mainly to two factors. First, increase in material properties, and second and more important, intensified research by numerous investigators through the world. During the past few years, however, there has only been a relatively small increase due to diminishing returns caused by an approach to an upper limit.

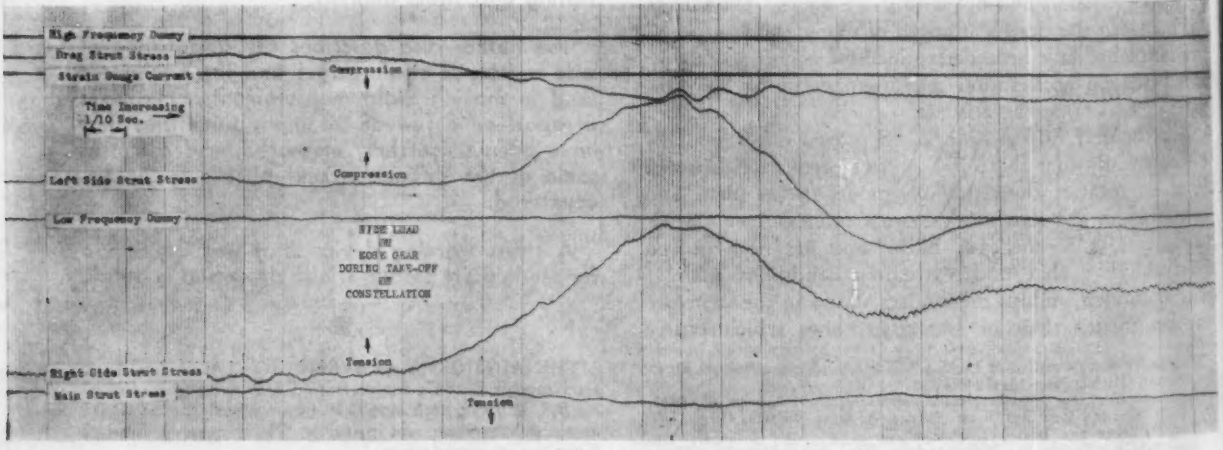
The question now is, how much effort will it take to reduce the structural weight of the airplane another one per cent through increasing the allowable compression or tension stress compared with accomplishing the same thing through accurate determination of the loads applied to the structure. Or stated in another way, is it sensible to devote a lot of effort to obtain another thousand psi allowable stress or another one per cent increase in accuracy in the allowable stress of a structural member when the aeronautical engineer does not know the loads applied to the structure to within, let us say 10, 20, and sometimes 50 or more per cent. It seems to the author that greater returns will be gained in the next few years toward improving the art of airplane design through structural flight research than the same amount of effort and cost devoted to structural testing of a static nature – thus the reason for this paper.

■ Strain Gages

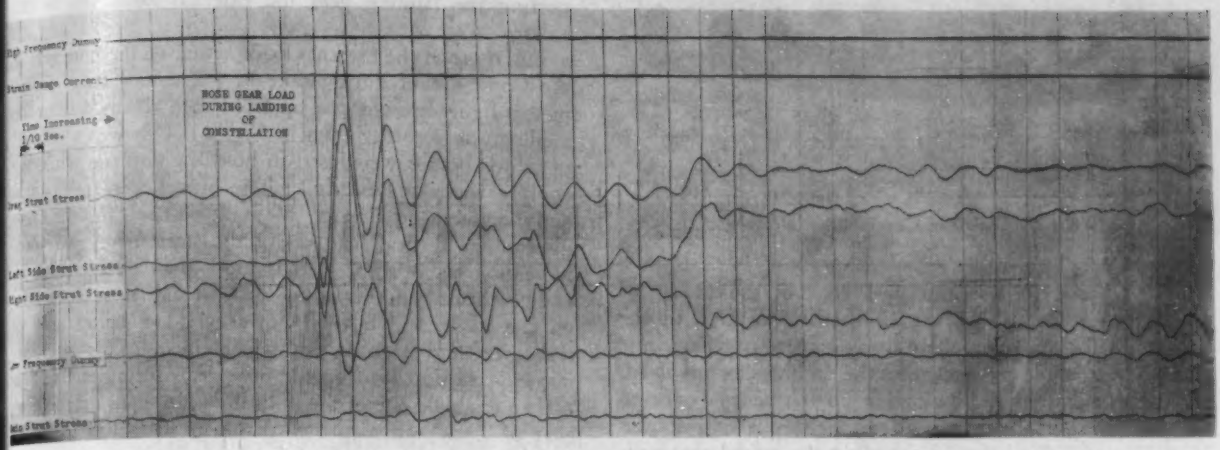
One of the chief reasons why structural flight research has not been more actively pursued has been due to the lack of suitable instruments. Remote reading strain gages which were accurate, stable, easy to install, fast responding, and suitable for use in an airplane have been nonexistent until the appearance of the wire resistance strain gages. At Lockheed Aircraft Corp., considerable research has been done on the wire resistance strain gage first commercially introduced by Baldwin-Southwark Division.

The wire resistance strain gage is a perfect solution to many of the instrumentation problems connected with structural flight research. See Fig. 1. First of all, they are by their very nature remote reading. Gages can easily be mounted on the structure almost anywhere, once the technique of attaching has been developed. The speed of response of these gages is apparently faster than any investigator has been able to measure to date. Tests conducted up to frequencies as high as 30,000 cps show no indication of an upper limit to the frequency of response to these gages. This feature makes these gages ideal for studying dynamic strains under such conditions as gust and landing. Another nice feature of these gages is that strains of different members can be integrated electrically, that is to say, tension and compression stresses can either be added or the difference taken. This is very useful in measuring the loads in a column in which there may be bending or in a member in which the direct stresses are desired and there may be bending. In such a case, gages can be put on both sides of the member and the output be added such that any bending strain is eliminated from the reading. With the development of the wire resistant strain gage now at a point where it is a reliable and accurate instrument, there is no reason for not carrying on structural flight research.

The equipment used by the Lockheed flight-test group for strain-gage work is very simple basically. It consists of just four units: batteries, gage circuit, switching unit, and an oscillograph. For accurate tests, a separate battery or batteries are installed in the airplane to ensure constant current to the gage circuit; however, the ship's battery may



■ Fig. 2 – Oscillograph record obtained from strain gages on nose gear of Constellation airplane during take-off



■ Fig. 3 – Oscillograph record obtained from strain gages on nose gear of Constellation airplane during landing

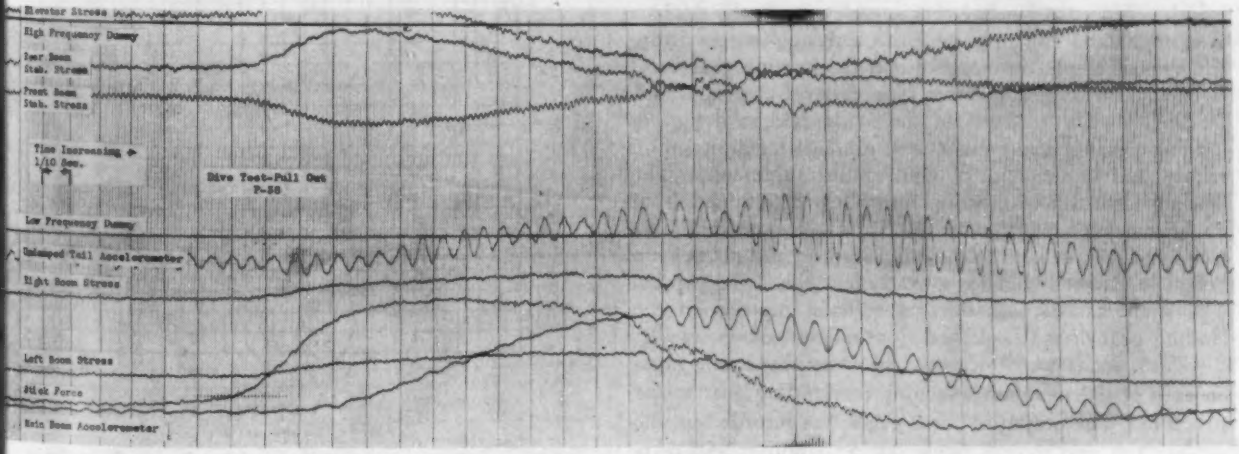
be used on tests where the accuracy required is not too high. It is important to maintain constant current since the output of a strain-gage circuit is directly proportional to the current flowing in the circuit. The strain-gage circuit consists of four or more gages in the form of a wheatstone bridge. As was previously mentioned, the gages in this circuit can be made either to add or subtract, thereby averaging or summing various stresses. More complete information about strain-gage bridges will be the subject of a future paper.

The details on the technique of making accurate strain-gage bridge installations are too elaborate to be incorporated in this paper. It is obvious to anyone familiar with making electrical measurements that a bridge circuit which is sensitive to a resistance change of less than 1/100 of an ohm requires considerable attention to details: effect of moisture, thermoelectric effects, and the effects of sunlight must all be considered.

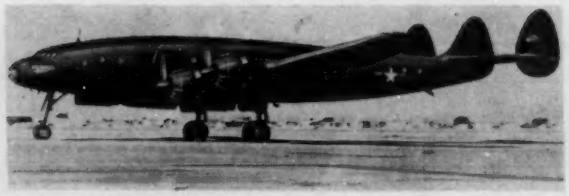
As in all strain-gage work the detail location of the gages is of utmost importance. Stress concentrations, local bending, and effect of combined stresses must all be watched for in installing strain-gage bridges. At Lockheed, a great deal of effort has been devoted to getting more and more power output out of wire strain gages, and several marked developments have been made toward accomplishing this end. This development had advanced to the point where many loads are measured on meters directly with no amplifiers or oscillator equipment at all. That is to say, a strain-gage bridge drives a meter directly. The switching arrangement used consists of a number of relays. These relays are hooked up such that when a switch is turned on a certain quantity is recorded on the oscillograph. For example, if the flap load, nose gear landing load, and the main gear load all need to be recorded, three switches are turned on – one for each quantity to be measured. If fuselage stresses or tail stresses are to be measured another switch is thrown. The reason for this is that we are definitely limited to the number of quantities that we can record at one time. The oscillograph used is a standard Miller oscillograph with high sensitivity elements. These highly sensitive elements have a flat response characteristic up to approximately 40 cps.

In general, we have found that the stresses measured on the airplane have a frequency less than 40 cps. This is undoubtedly contrary to what many men think but, nevertheless, to date we have measured few stresses which change with a frequency greater than 30 or 40 cps. One of our only exceptions was obtained in a recording of recoil loads during the firing of a gun. The type of records obtained from the oscillograph are shown in Figs. 2, 3, and 4.

Fig. 2 shows a part of the oscillograph record obtained during the start of a take-off of the Constellation. This record gives one an idea of how much deflection can be obtained on the traces and the general nature of the traces.



■ Fig. 4 – Oscillograph record obtained from strain gages on a P-38 during dive test pullout



■ Fig. 5 – Lockheed Constellation airplane

Fig. 3 shows an oscillograph record of the nose gear loads during landing of the Constellation. Due to the abrupt nature in which these loads come on, a harmonic analysis was made to determine the frequencies involved. The harmonic analysis shows that no frequencies over 30 cps showed up in the record. Thus the frequency response of oscillograph elements did not affect the record obtained.

Fig. 4 shows a more complicated oscillograph record. It is the record of a dive test pullout of a P-38. It should be mentioned that both the tail accelerometer and the main beam accelerometer were of an undamped type, thus the oscillatory nature of their record.

The low-frequency dummy, so called, is installed in order to determine the effect of the pullout on the oscillograph unit itself.

During the last two years the recording end of strain-gage work has given us considerable more trouble than the strain gages themselves. At least 80% of our problems involved in carrying on structural flight research during the last two years has been connected with the oscillograph. Position of the trace on the record is not only affected by temperature but may be seriously affected by acceleration if the element is not balanced. At the present time our limit of accuracy is governed almost entirely by the oscillograph and not by the strain gages themselves. Note in Fig. 4 the sudden increase in stick force and also note the lag between the change in stick force and acceleration in the main beam of the airplane.

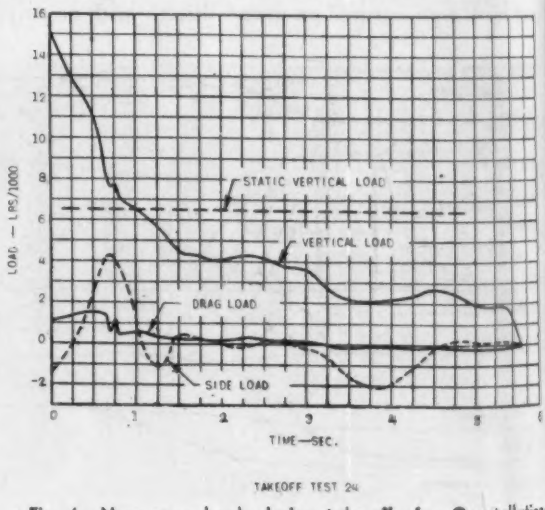
■ Landing Gear Load Measurements

One of the most important structural items of an airplane is the landing gear, it is also one of the most difficult to design. During the year of 1940, 47.6% of all structural failures on commercial airplanes were in the landing gear structure. In a modern airplane the landing gear will constitute (nose and main gears) 12 to 14% of the structural weight. Since the landing gear is useless while the airplane is in flight and still must be carried throughout its entire lift, it is of utmost importance that the landing gear be as light as possible and still free from failures (barring crash landings). The majority of the loads used in designing landing gears at the present time are entirely arbitrary in nature due to the lack of accurate measurements. The landing load factors used in design are determined from an arbitrary equation which is a function of the gross weight of the airplane or the wing loading and a free-drop height is required which is a function of the stalling speed.

A study has been made of the loads imposed on the landing gear of the Lockheed Constellation shown in Fig. 5. These loads were measured by installing strain gages on each of the members attaching the landing gear to the structure. The output of these gages was recorded on the oscillograph and records made during landings and take-offs. From the record of the stresses in each member supporting the landing gear the external loads can be calculated.

Fig. 6 shows the external loads acting on the nose gear of the Constellation during take-off. The side loads shown in this figure are worked up from data obtained from the oscillograph record shown in Fig. 2. This was a take-off in which brakes were used to hold the airplane while the engines were "revved up." The brakes were released at approximately zero time. Thus, the reason for the large increase in vertical load above the static nose gear load. Note that the vertical load is more than doubled due to engine thrust. It might be mentioned that at zero time on the curve, full take-off power was not being used so that even higher loads can be obtained. This large increase in vertical load on the nose gear during engine run up can easily be one of the critical design loads on the nose gear of a modern airplane. From a structural flight research standpoint this load is of passing interest since it can be accurately calculated by knowing the thrust developed by the propellers and the geometry of the airplanes. However, the load that is of considerable interest is the side load of more than 4000 lb, which occurred 7/10 of a sec after the brakes were released. This load was due to either the two brakes not releasing at the same instant or else the pilot applying one brake momentarily in order to line the airplane up with the runway. Loads of this type cannot be calculated without making a lot of assumptions, and arbitrary assumptions can be costly in terms of weight or failures.

Fig. 7 shows loads which occurred on the nose gear during a landing of the Constellation. Some of these loads are definitely not external loads, but are "apparent" loads. By "apparent" is meant that the stresses on the struts are converted into vertical, side, and drag loads which would produce these stresses. Undoubtedly, some of the stresses are caused by inertia forces and not external loads, but the net effect on the structure is the same, so they can be called apparent loads. This is quite an important point since analytical men may say certain measured loads aren't possible because of limiting friction coefficient or for some other reason, whereas, actually it has not been taken into account that these apparent loads are not equal to the external loads. The dynamic load reversal of the drag load in Fig. 7 is of particular interest. This is obviously the influence of the natural frequency of the gear itself.



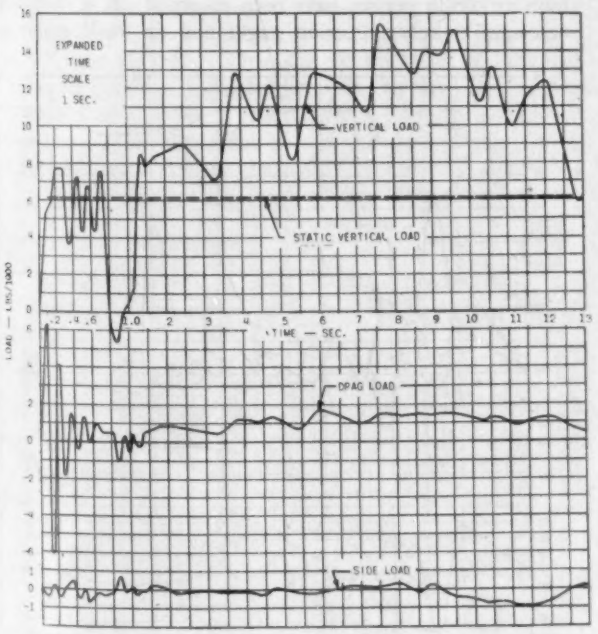
■ Fig. 6 – Nose gear loads during take-off of a Constellation airplane

The general characteristics of these nose gear loads are surprisingly similar in nature to those recently published by W. B. Bergen, of the Glenn L. Martin Co.[3] Since the airplanes on which the measurements were made are widely different, a ray of hope is cast in obtaining some rational design loads from analysis of the nose gear loads obtained from flight measurements. The dynamic nature of the vertical load in Fig. 7 is believed to be due mainly to two things. One is the binding of the shock strut, which builds up a force which in turn deflects the tire and then the strut moves and the force decreases; and second, the airplane has a certain natural frequency in pitch about its center of gravity.

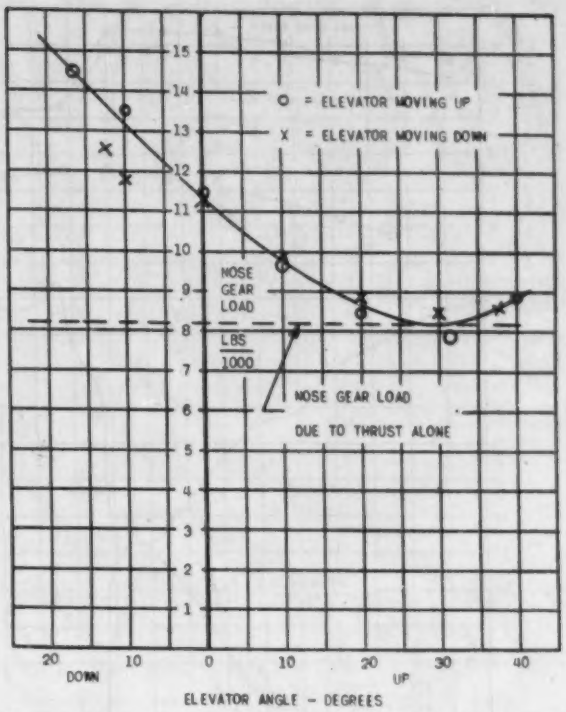
While making engine runs on the ground in the Constellation it was noticed that elevator deflection affected the nose gear load. In order to investigate this effect of elevator angle on the nose gear loads during engine run up and prior to take-off, a test was conducted with all four engines running at full power, the brakes locked, and the elevator moved throughout its full range of travel. The results of this test are shown in Fig. 8. The nose gear load values on this curve are those in excess of the static vertical load and are thus applicable for any cg condition. The calculations for the nose gear load due to thrust were made by using a thrust value obtained from the measured drag load on the main gears. Fig. 8 shows that under full-power conditions, movement of the elevator from one extreme to the other can cause a 6000-lb load change on the nose gear. This result has a special importance in this paper, since this type of load is liable to be overlooked by the stress analyst. Obviously, it is far from being negligible in magnitude and especially when it adds to one of the critical conditions.

Effects of prerotation of wheels were studied by making tests where the nose gear was set back on the ground at different time intervals after being lifted off of the ground.

[3] See *Journal of the Aeronautical Sciences*, Vol. 10, October, 1943, pp. 233-249: "Experimental Investigations in Aircraft Dynamics," by William B. Bergen.



■ Fig. 7 – Nose gear loads during landing of a Constellation airplane



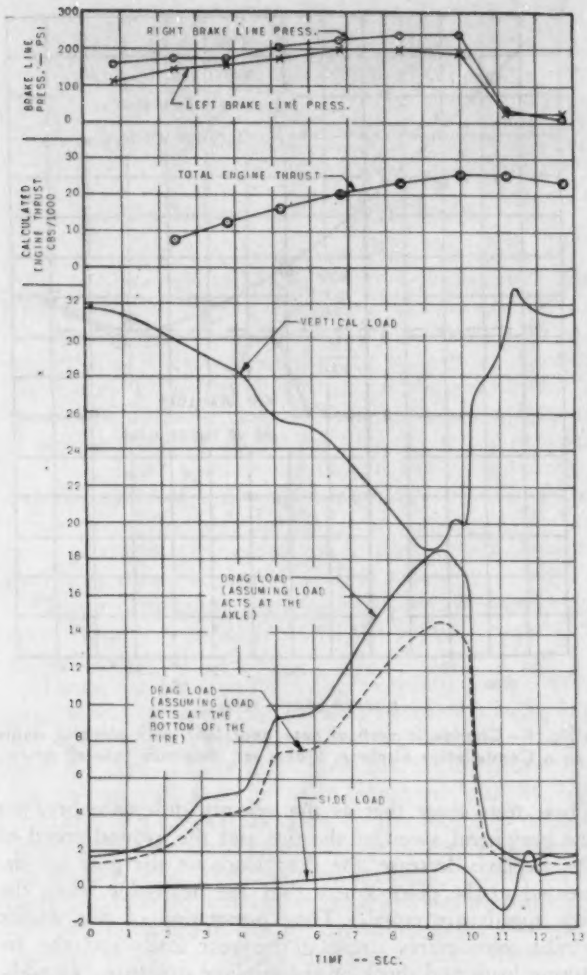
■ Fig. 8 – Change in vertical nose gear load with elevator angle on a Constellation airplane, brakes set, maximum take-off power

These tests show that as the velocity difference between the peripheral speed of the tire and the ground speed of the airplane decrease, the drag loads on the gear are decreased to the point where they are negligible when the two speeds are equal. Thus prerotation of the wheels would considerably decrease the gear loads and the resulting loads and shock on the airplane structure. In addition, the elimination of this drag load on the gear during landing would contribute greatly to smoother landings.

■ Main Landing Gear

Some measurements of the main gear loads during the start of a take-off of a Constellation are shown in Fig. 9. The loads were measured by means of electrical strain gages placed on the side and drag struts similar to the installation on the nose gear. The above figure shows the vertical, side, and drag load on the left gear as well as the brake line pressure and calculated engine thrust. Note how the vertical load decreases as the power increases, this means that some of the load is being shifted to the nose gear by thrust and some is being carried by lift on the wing produced by the slipstream of the four propellers. The explanation for the vertical load going back to nearly its static value immediately after releasing the brakes is believed to be due to the thrust force being resisted by inertia loads instead of by the moment produced by nose gear load.

The loads on a main gear during a landing are shown in Fig. 10. The oscillatory nature of these loads are admittedly impressive. Here the measurements show that there is an "apparent forward drag load" which is larger than the rearward one occurring at the instant of impact. The reason for this large forward load is that when the airplane is in landing attitude the main shock strut extends forward, thus the gear tends to go forward as the vertical load builds up. This effect plus the inertia of the


RIGHT BRAKE LINE PRESS.
LEFT BRAKE LINE PRESS.
BRAKE LINE PRESS. — PSI
TOTAL ENGINE THRUST
CALCULATED ENGINE THRUST LBS/1000
VERTICAL LOAD
DRAG LOAD (ASSUMING LOAD ACTS AT THE AXLE)
DRAG LOAD (ASSUMING LOAD ACTS AT THE BOTTOM OF THE TIRE)
SIDE LOAD
TIME — SEC.

■ Fig. 9 – Left main landing gear loads during the start of a take-off of the Constellation airplane

gear results in the large negative drag load. Since the main gear cannot caster like the nose gear, much larger side loads can be set up due to drift, thus the 4000-lb side load.

The structural research conducted to date on the Constellation's landing gears indicate the following results which are important to future design:

a. For landings there is no correlation between gross weight or center-of-gravity position, with the main and nose gear loads. The loads are mostly a function of the pilot technique in landing and the field.

b. At limit load conditions on the nose gear it is possible for the forward "apparent" drag load resulting from the dynamic action of the gear to be equal to the aft drag load and for both to be equal to the vertical load. Also, it is possible due to the action of the tire and shock strut for the vertical load to be sustained long enough so that both fore-and-aft drag loads will occur during the period which the vertical load is a maximum. Thus the ratio of drag load to vertical load for design conditions would be ± 1.0 for the nose gear.

c. At limit load conditions on the main gear it is possible for the forward "apparent" drag load resulting from dynamic action of the gear to be equal to the aft drag load and for both to reach values equal to 0.8 of the vertical load.

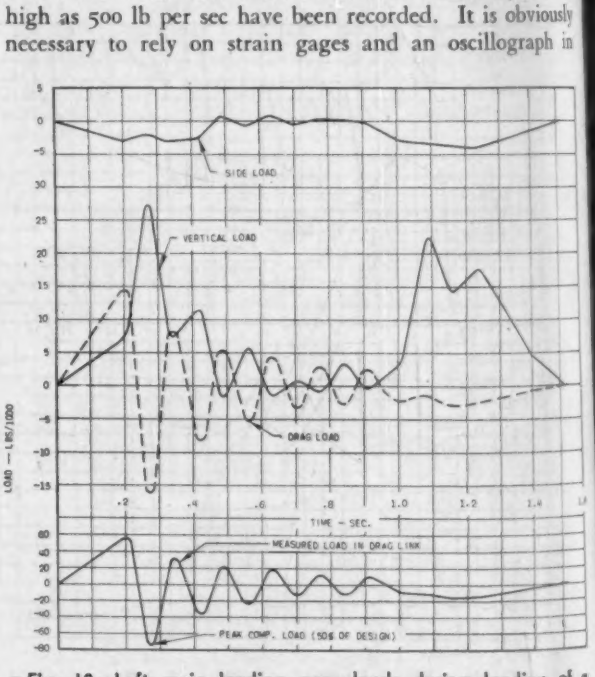
d. Drag loads on the gears during landing can be reduced to very low values by prerotating the wheels.

The above conclusions should not be interpreted as being universal since they may be and probably are influenced by the construction of the gear. It is to be expected, however, that the general dynamic characteristics will be present in all gears. Some of the many landing gear failures in the past may have been due to reversals in drag load which had never been considered in designing the gears.

■ Control Forces

The measurement of pilot control forces has become a standard item in flight-test requirements. This is one phase of flight testing where aerodynamics and structural research meet. Both branches of aeronautics are not only interested in the magnitude of control forces but also their rate of change. In flight testing a modern airplane it has become necessary to determine exactly the forces which the pilot must exert to control the airplane.

Several years ago a test pilot would come down after making some pullouts and say that he had to pull several hundred pounds in order to get a certain acceleration. This method of testing was very unsatisfactory since it is very difficult if not impossible to estimate with any degree of accuracy forces required under high accelerations. When one considers the fact that during a pullout the pilot's arms and hands weigh four or five times their normal weight it is easy to see how difficult it is to estimate the pull which is being exerted on the control wheel. Fig. 11 shows an example of a record of elevator control force against time during a pullout. On the same plot the acceleration at the center of gravity is also shown. Note the time lag between the peak elevator control force and the maximum acceleration. This lag is, of course, natural and easy to understand. The chief object in showing this figure is to indicate the rapidity at which the control forces can be changed; in less than a second the load on the control column changed over 100 lb. This is by no means the maximum rate that has been recorded, rates as high as 500 lb per sec have been recorded. It is obviously necessary to rely on strain gages and an oscillograph in



■ Fig. 10 – Left main landing gear loads during landing of a Constellation airplane

order to obtain accurate measurements at rates this high.

For several years in flight test we have measured all the control forces (such as elevator, aileron, rudder, and brakes) by the use of strain gages which were placed on standard ship members. The output of the gages was put on meters directly without amplifiers or oscillators. The meters were recorded by the automatic observer, which makes a photographic record of all pertinent data connected with flight testing of the airplane.

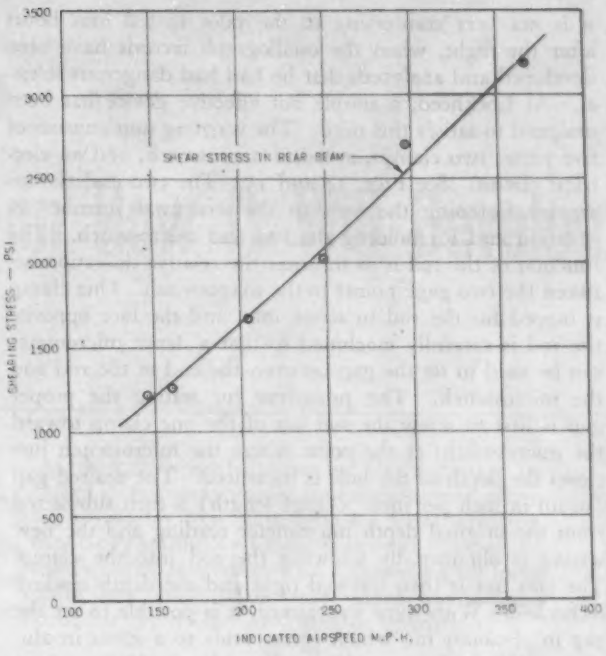
■ Maneuvering Loads

Recently, a series of roll tests was conducted. These tests were conducted on an airplane for which the rate of roll had been doubled by use of a new device. Consequently, there was some concern over how high the stresses in the airplane would be under high-speed rolls. This interest resulted in a structural flight-test program along with the aerodynamic program of determining how fast the ship would roll. A theoretical analysis of some of the stresses produced under rolling conditions requires a knowledge of the flight path of the center of gravity of the airplane.

Strain gages and an oscillograph were installed in the airplane. The strain gages were mounted on all of the members thought to be critical. After several flight tests in which mild rolls were made it was found from an analysis of the oscillograph records that few if any of the members in the airplane were highly loaded. One of the highest stressed components in the airplane was found to be the rear shear beam of the wing just inboard of the aileron.

For these tests the shear stress in the rear beam was considered to be a function of two parameters. The one parameter was the airplane speed and the second was the maneuvering load. To isolate these two parameters the shear stress was measured for various level speeds. A plot of the measured shear stress against indicated air speed is shown in Fig. 12.

The ship was gradually rolled faster and faster, with a check of maximum measured stresses with each increment of increase in rate of roll until finally the maximum rate of roll was reached. This was repeated for a range of indicated speeds. A reduction of the stress data obtained is shown in Fig. 13. This figure shows the change in rear beam shear stress due to just the maneuvering loads. There is some scatter in the points, but when it is realized that these points are the result of two different pilots rolling the ship as fast as they could at each speed, the scatter is certainly not excessive. The curves in Fig. 13 show that higher maneuvering stresses are set up in stopping the roll than starting it. This is probably to be expected since in stopping the roll the aileron helps the pilot return the surface to neutral, whereas it opposed when starting. The maximum or minimum stress occurring during the maneuver can be obtained by correcting Fig. 12 for the load factor developed and adding the maneuvering stress given in Fig. 13.

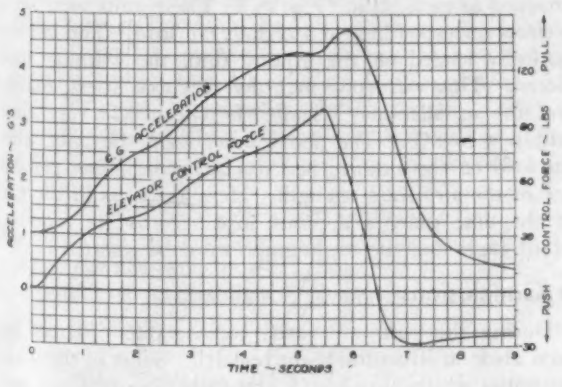


■ Fig. 12 – Shear stress in wing rear beam for level flight against indicated air speed

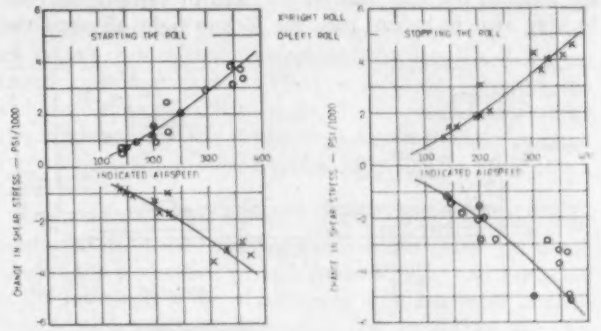
The general result of the structural measurements made during this roll investigation was that the stresses produced in flight were not as high as calculations indicated. Also, the engine mount was believed to be critical; however, it was found that engine cowling which has normally been considered as purely nonstructural, carried a large percentage of the engine loads.

■ Stress Warning Device

It is one problem to measure the stresses occurring during various maneuvers, but it is quite another to warn the pilot of dangerous structural conditions. In other words,



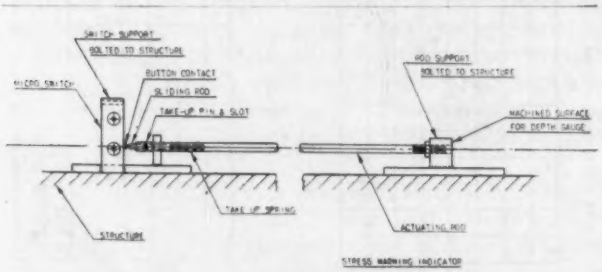
■ Fig. 11 – Elevator control force and acceleration against time for a dive test pullout



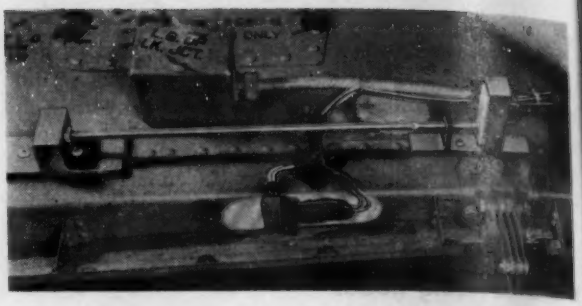
■ Fig. 13 – Maximum change in rear beam shear stress due to rolling the airplane at different speeds

it is not very comforting to the pilot to tell him hours after the flight, when the oscillograph records have been developed and analyzed, that he had had dangerous stresses. At Lockheed, a simple but effective device has been designed to satisfy this need. The warning unit consists of five parts: two clamps, a rod, a microswitch, and an electrical circuit. See Figs. 14 and 15. The two end clamps are for fastening the unit to the structural member in question and for holding the rod and microswitch. The function of the rod is to transmit the relative deflection between the two gage points to the microswitch. One clamp is tapped for the rod to screw into, and the face opposite the rod is carefully machined so that a depth micrometer can be used to set the gap between the end of the rod and the microswitch. The procedure for setting the proper gap is first to screw the rod out of the one clamp toward the microswitch; at the point where the microswitch just closes the depth of the hole is measured. The desired gap (strain in inch per inch $\times$ gage length) is then subtracted from the original depth micrometer reading and the new setting is obtained by screwing the rod into the clamp. The jam nut is then screwed tight and the depth reading rechecked. With some experience, it is possible to set the gap to $\pm$0.0001 in., which corresponds to a stress in aluminum alloys of $\pm$100 psi or 300 psi for steel (for a gage length of 10 in.).

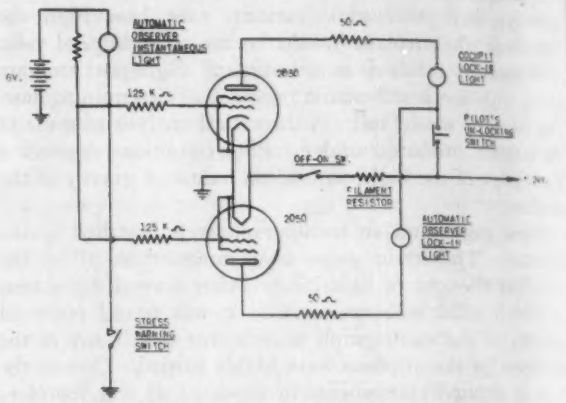
In order that the stress warning device will operate even if the stress for which the unit is set occurs for only a very short interval of time, a special electrical circuit was designed. See Fig. 16. This circuit is designed with two types of warning lights; one light burns whenever the stress warning switch is closed and goes out when it opens; the second light, the so-called "lock-in light," burns whenever the stress warning switch is closed and continues to burn after the switch is opened. The feature of locking one light on can be accomplished by the use of multiple point relays; however, normal relays require a time interval of approximately one or two hundredths of a second in order to lock in. This would mean that if there was a vibrating stress or a peak stress whose duration was less than one or two hundredths of a second, no indication would be obtained. Since many of the natural frequencies of airplane structural components are of the order of ten cycles per second, relays could not be safely used in a lock-in circuit (20% error if sinusoidal). By employing thyrotrons, a circuit can be built which will have a very fast "speed of response." The circuit in Fig. 16 will lock in if the stress warning switch closes for longer than 4 microseconds (0.000004 sec). From the lock-in light, the pilot knows that the structure has been stressed more than the amount the unit was set for, and by pushing the unlocking switch, he can turn the lock-in light off until the stress is again exceeded.



■ Fig. 14 – Diagram of stress warning indicator



■ Fig. 15 – Photograph of stress warning unit on a channel in a P-38 wheel well – note strain gages below warning unit



■ Fig. 16 – Stress warning unit electrical circuit

Stress warning units have been used successfully in several applications. On the maiden taxi tests of the Constellation with its tricycle landing gear, it was thought desirable not only to measure the dynamic stress in the nose gear on the oscillograph but also to install warning units. Units were installed on each strut and set at approximately 40% of limit load for the first tests. The warning lights were placed in the cockpit so that the pilot could watch them while maneuvering the airplane around on the ground. By watching the stress warning lights, the pilot was able to determine just what type of handling produced high stresses in the side struts and drag struts. During landings, through the use of the lock-in feature, the same units would indicate whether or not certain stresses were exceeded.

In dive tests on a P-38, stress warning units have been attached to each boom (Fig. 15). These units were set at a stress corresponding to a certain tail load. The tail was statically loaded on the ground until the warning units closed. This was done in order to check stress analysis and the installation. Usually the check was very close, within 5 to 8%. The above units warn the pilot when the load on the airplane had reached some preset percentage of the airplane's strength. Here again, as in the case of the nose gear, if accurate load records are required, strain gages and the oscillograph must be resorted to.

■ Conclusions

During the past few years, considerable progress has been made in structural flight research. Some of the many quantities which have been successfully measured are: control forces, landing gear loads, gear door loads, engine

concluded on page 444

TRUCK DESIGN—

Guesses on How It Will Develop

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The Autocar Co.

THESE down-to-earth speculations by Mr. Bachman should help to clarify for all their picture of the truck of the future.

Mr. Bachman considers such a detailed but important item as increasing the overall width up to the rear fender top line to 102 in. from the 96-in. figure now adhered to in all but two states for the maximum width at any point, but keeping the width of the cab at 96 in., because in traveling along the highway, the cab actually takes up several more inches, anyway. These extra inches across the fenders would allow improvements to be made in tire mounting, spring suspension, riding qualities, stability, and braking.

Other changes foreseen by Mr. Bachman include: increases in weight and horsepower, allowing for faster and more efficient transportation, especially on the first-class highways planned by the Federal government for connecting all large cities and most small ones; more comfortable cabs for the driver; and quieter operation.

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THE AUTHOR: B. B. BACHMAN (LM '10) vice-president of engineering, Autocar Co., since 1929, joined the company in 1905 after previous affiliations with several Philadelphia manufacturing companies and with the Frankford Arsenal. President of the SAE in 1922, Mr. Bachman is now chairman of the SAE War Activity Council, under whose direction the entire SAE War Program is coordinated. He is president of CRC board of directors, Society Treasurer, and active member of the War Engineering Board.

SOME day, we will enter an era of repairing the ravages of war and rebuilding a new world.

Everyone, in thinking ahead to that time, cannot help but speculate on the form which the new things will have. We know that through destruction and the interruption of normal processes forced on us by the war, opportunity will exist for starting almost with a clean slate. In some parts of the world, fortunately not here in our own land, whole districts have been wiped out. This destruction, as ghastly as it is, has nevertheless created the opportunity for correcting the errors of earlier days and eliminating the handicaps of archaic structures which could probably never have been accomplished in any other way. This removal of restriction cannot help but influence general thinking which will affect the whole world. In view of these things, it is small wonder that we who are concerned with the field of highway transportation should feel the surge of questions as to what the future holds in store for us in the way of opportunities for doing things in different and better ways.

The first thing to be considered in trying to determine what the future developments may be is the kind of roads on which our new vehicles will operate. In 1918, the highways outside of metropolitan areas and a relatively small mileage of state highways were just coming out of the mud. While the Lincoln Highway in Pennsylvania and the National Road in Maryland were hard-surfaced throughout, the grades and curves were severe and the general condition and contours of the road held speeds down. These roads, while not the best examples of existing highways of that date, were much better than the average for the northeastern part of the country and infinitely better than other sections excepting the West Coast.

Federal aid to states for highway construction was begun in 1916 and has continued since that time. Between 1920 and 1941 more than 1,500,000 miles of highways have been surfaced and over $40,000,000,000 spent to accomplish this purpose.

The development of this program has provided a highway system which has made almost every section of the country accessible on hard-surfaced roads. The activities of the state highway departments have kept these roads open to travel through all seasons, and these facts have contributed largely to the phenomenal growth of highway transportation both for passengers and freight. The highest point to which this development has progressed is illustrated by such examples as the Pennsylvania Turnpike, the Pulaski Skyway in New Jersey, the several tunnels under the Hudson and East Rivers, together with their connecting roadways, and the Express Highway in New York City.

Unfortunately and inevitably, the growth of the traffic promoted by these facilities has created other problems. One is the congestion in and around cities, and the other is the mounting score of accidents with attendant fatality, injury, and property damage. The engineers who are

[This paper was presented at a meeting of the Pittsburgh Section, Pittsburgh, Pa., April 25, 1944.]

engaged in the highway field throughly appreciate their share of responsibility for these conditions and have pertinent and practical ideas for improvement. It behooves those who are engaged in the manufacture and operation of the vehicles to exercise their abilities in a similar manner.

■ Interregional Highway

Probably the most definite information as to the character of thinking being done relative to the future highways, is contained in the report of the National Interregional Highway Committee transmitted to Congress by a message from the President in House Document 379. This report submits a proposal for a road system embracing a total of 33,920 miles. Of this total, 29,450 miles are in rural sections and 4470 miles are in urban sections. The total mileage is 1.04% of the 3,267,717 miles of rural roads and urban streets in the United States. While this, at first reading, may seem to be an insignificant figure, the supporting data given in the report reveal that it will provide a surprising service. Without attempting to duplicate this information, it is of interest to note that it will connect all cities containing 300,000 and over and will connect 587 cities having a total population of 51,805,000 out of 1077 cities with a total population of 62,716,000. These same 1077 cities had a total value for manufacture of $18,127,000,000, and of this total the 587 cities on the system accounted for 83%, or $15,045,000,000. The counties traversed by the system contributed 43% of the agricultural products of the country and contained 22,113,607 of the 32,210,562 vehicles registered.

The report proposes that provision shall be made in the design, or allowance made for subsequent provision for handling estimated traffic to be expected 20 years from the date of construction. The dimensional factors given in Table 1 are proposed for the vehicles.

Table 1 – Proposed Dimensional Factors for Vehicles

Width, in.	96
Height, ft	12½
Length (overall including bumpers and load)	
Single Vehicle, ft	35
Tractor and semitrailer combinations, ft	50
Other combinations, ft	60
Axle load* on pneumatic tires, lb	18,000

* Defined as the total load on all wheels whose centers may be included between two parallel transverse vertical planes 40 in. apart.

Gross weight on any vehicle or combination of vehicles can be determined according to the formula:

$$W = C\ (L \text{ plus } 40)$$

where:

W = Gross weight of vehicle, lb

L = Length between the forward and rear axles of the vehicle or combination of vehicles, or any group of axles thereof, ft

C = Coefficient with the following values:

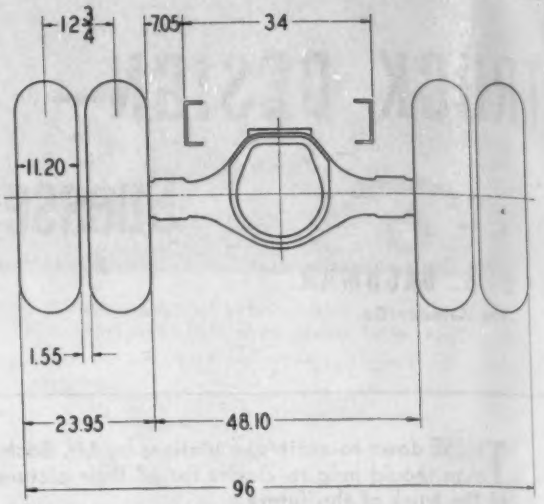
For values of L less than 18 ft, $C = 650$

For values of L equal to or greater than 18 ft, $C = 750$

All rural sections of the system are to be designed for a speed of 75 mph for passenger vehicles and 60 mph for trucks and tractor combinations in flat topography. In more difficult terrain, the design shall contemplate lower speeds, but in no case less than 55 mph for passenger vehicles and 35 mph for trucks in mountainous country.

The width of lanes provided is 12 ft and generous dimensions are stipulated for shoulders and median strips. Minimum sight distances, limits of curvature, and banking of curves are provided which contribute to the capacity for safe operation.

Gradients range from 3 to 6% dependent on topography and the traffic density anticipated. Separation of grades at intersections and regulating access to the road promotes speed and safety.



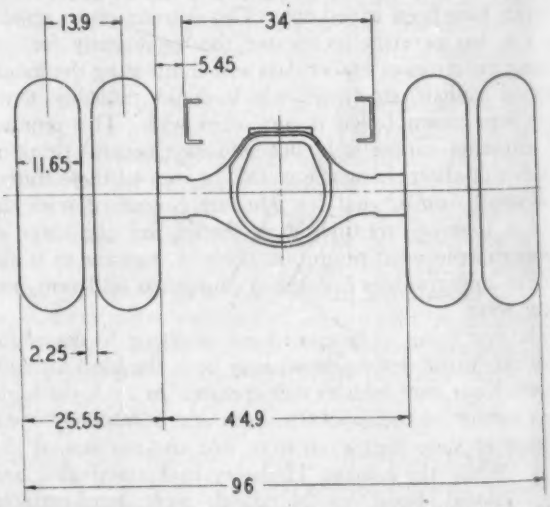
■ Fig. 1 – Layout with dual tires when overall truck width is 96 in. maximum – present tire base

■ Vehicle Width

With these road characteristics available for consideration, it is of interest to study the vehicle.

While the width of traffic lane is 12 ft, the width of the vehicle is limited to 8 ft. In view of the fact that a program involving billions of dollars and many years is under consideration, this relation should receive careful study.

The 96-in. width limit on vehicles has been in existence for a long time. By 1940 all states had adopted this dimension, while Connecticut and Rhode Island had adopted 102 in. Such a preponderance of opinion would seem to



■ Fig. 2 – Layout with dual tires when overall truck width is 96 in. maximum – proposed wide tire base

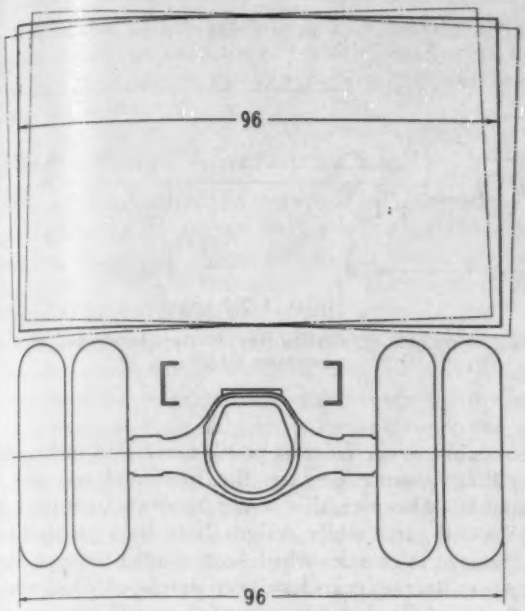


Fig. 3 – A body 96 in. wide takes up several inches more, when in motion

establish the validity of the 96-in. width on very solid ground. However, an analysis of the subject develops some very important questions. In the early days of the truck, the solid tire was used universally and maximum cross-sections did not run much over 14 in., which left ample space between the wheels for brakes, springs, and frame. With the advent of dual pneumatic tires, the picture has changed materially. The smallest tire which can be used to match the 18,000-lb axle loading permitted, is the 10.00-24 with a rating of 4550. The use of the 24-in. base diameter tire is undesirable in highway work due to the extra height of center of gravity which it produces.

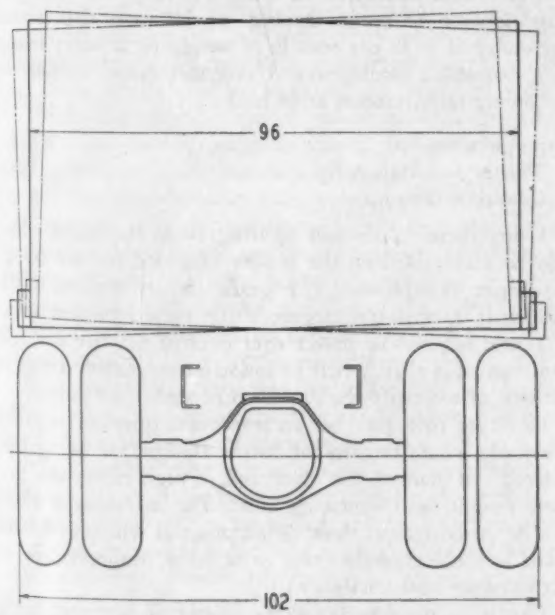


Fig. 4 – Layout using proposed dimensions: 96-in. body width and 102-in. width up to the rear fender top line

A reduction in diameter of base makes an increase in cross-section to 11.00 necessary to reach the desired rating. The 11.00 tire mounted on the 7.33V (9-10) rim with 12¾-in. spacing requires a space of 23.95 in. with 1.55 in. between tires, leaving a space between the inner tires of 48.10 in., as shown in Fig. 1. Recent proposals by the tire manufacturers based on experiments made by them indicate that desirable improvement in life of the tire can be obtained by increasing the width of the tire base. At present, the width of the base on the 11.00 size is 7.33 in.

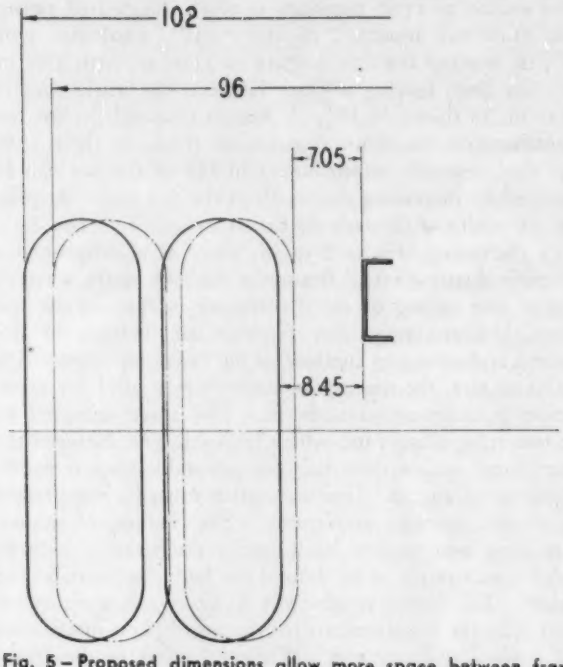
By increasing this to 8.37 in., there is accomplished a desirable distribution of flexing in the side walls, a reduction of side rolling of the tire and an increase in air volume. Unfortunately, this requires an increase in tire spacing and causes an increase in the tire cross-section. On the 11.00 tire, the spacing is made 13.9 in. and the cross-section is increased to 11.65 in. The space occupied by the two tires is 25.55 in., which leaves 44.9 in. between the inner tires, or less than half the allowable overall width, as shown in Fig. 2. Into this space must be compressed the brakes, springs, and frame. The brakes, of course, were long ago pushed back under the tires, a location which leaves much to be desired for both the tires and the brakes. The frame is generally 34 in. wide, a figure arrived at by the requirements for the powerplant installation and a straight frame rail. While an offset in the frame rail in the horizontal plane can be made without causing any production troubles, it introduces elements of weakness which are undesirable. Finally, we have left 10.9 in. into which the springs, clips, and clearance must be located. What is done to a decent spring design "shouldn't happen to a dog."

All of this is based on the assumption that chains will be provided for on only the outside tire and that the chains can extend beyond the width limit without incurring the wrath of the gendarmes.

The width limit is in one important particular placed to prevent excessive and unreasonable width for vehicles that produce a hazard to passing vehicles and interfere with the vision of following vehicles. However, no distinction is made as to the wheels and the body. It is the width of the body which limits vision and the width of the body plus side sway which determines the clearance requirements. It seems obvious that the space occupied by a body 96 in. wide in motion will exceed 96 in. by an appreciable amount. See Fig. 3. Now, if this fact is recognized, then it would seem practical to increase the width over the tires to a greater dimension than the body without incurring any bad effects and at the same time making it possible to improve some conditions which at present are not so good.

These factors were recognized by the SAE Automotive Transport Code Committee appointed in 1931 to study this problem, and the report approved by the SAE Council in 1935 proposes that the body width be set at 96 in., but that up to the rear fender top line 102 in. be permitted, as shown in Fig. 4. This additional 6 in. would make possible important design changes in trucks and buses which would favorably affect both the safety and performance.

Half of this increase allotted to the tire width would provide the desirable improvements in tire mounting which wider rim bases will produce, affecting both tire life and vehicle stability and safety. The remainder used to provide longer and wider springs with greater spring centers, will improve riding, reduce shocks loading on the vehicle and the road, and improve stability, reducing sway of



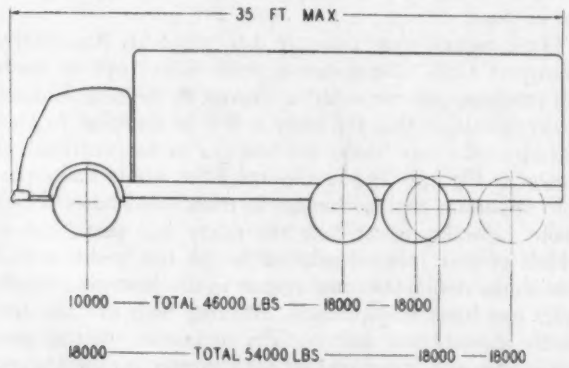
■ Fig. 5 – Proposed dimensions allow more space between frame and tires making improvements in tire mounting, spring suspension, and brake cooling possible

body and load. The available space for brakes will also be increased and even though this space is still inside the wheel, important improvements can be made by changing proportions which should increase the space between the drum and the rim, thus making better airflow around the brake possible. See Fig. 5.

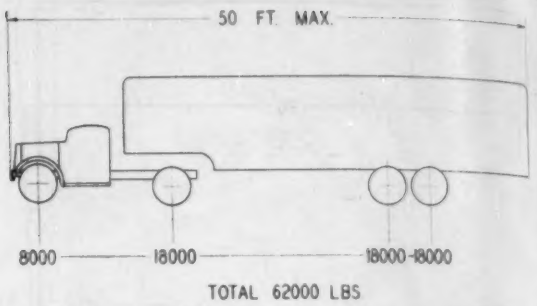
■ Vehicle Weight

In light of the limitation on gross weights provided by the formula which makes the weight a function of length, it is of interest to consider the possibilities for the three classes of equipment recognized in the length provisions.

A single vehicle is limited to 35-ft length and on the basis of normal design practice, a six-wheeler could be set up which would have a rating of 46,000 lb, as shown in Fig. 6. It would be possible to increase this to a maximum of 54,000 lb. This would be accomplished by pushing the rear axles as close to the rear end as possible and the front



■ Fig. 6 – Weight distribution for single vehicle of 35-ft maximum length



■ Fig. 7 – Weight distribution for tractor semitrailer of 50-ft maximum length

axles as close to the front as possible, and thus making the controlling distance between the front and rear axles a maximum. However, this would involve a front-axle load of 18,000 lb, and while designs have been proposed for dual tires on front axles which seem to offer some promise, and power steering gears have been developed which would handle an axle load of this amount, there still remain many problems to be solved to make such an arrangement practicable.

The tractor semitrailer, limited to 50 ft in length, could be rated at 62,000 lb. With the limitation of 18,000 lb per axle, this would require four axles, as shown in Fig. 7.

For other combinations, a maximum of 70,000 lb is indicated, and this would probably require five axles, although as with the single unit and subject to the same problems, a four-axle design can be visualized. See Fig. 8.

■ Vehicle Performance

From the speed limits which are proposed, an approximation can be made of the horsepower which should be provided to realize the advantages obtainable from operation on a highway of this character.

From data contained in a report by C. C. Saal[1], an extrapolation of curves in Fig. 11 indicates that a total resistance of 30 lb per 1000 lb of weight for a heavy vehicle is a reasonable assumption. Using this value, we find the following requirements at 60 mph:

Single unit, hp	245
Tractor semitrailer, hp	330
Combinations, hp	373

Using these values and figuring back, the speed which can be maintained on the grades provided for are for all categories 35 mph on a 3% grade and 21 mph on a 6% grade. It is, of course, apparent that these represent a considerable increase in power over current practice and it is quite possible that it will be found impracticable either to provide or to justify the provision of such high power.

However this may be, an important question is raised when the characteristics of future design are being considered. In general, the objections to high power are: first cost, weight, and operating cost. The advantages which can be offset against these objections and which can make them less objectionable are, principally, improvement in performance and reliability.

This is no place to argue the matter of first cost, but it

[1] See *Public Roads*, Vol. 23, May, 1942, pp. 33-54: "Hill-Climbing Ability of Motor Trucks," by C. C. Saal.

is believed that an excellent case can be made for the contention that of two units the one which initially costs more will be less expensive through its life and will have a longer useful life.

Engine Power Affected by Fuel

The weight question presents scope for many interesting speculations. In the engine there are two basic means for reducing the weight in relation to the power produced: (a) by increasing the horsepower per cubic inch displacement; (b) by decreasing the weight.

The first of these involves fuel, design factors, and materials; the second, design and materials.

An important consideration with respect to fuel is the octane rating and the resultant effect on compression ratio. A great deal of interest has been aroused by the volume production of 100-octane fuel for airplanes and the performance which this fuel has made possible. In some quarters, the idea seems to be that after the war this kind of fuel will be so plentiful it will be available for every use. Of course, this sort of thing is as fantastic as some of the pictures of post-war automobiles which adorn the advertising pages of every magazine. A paper by Bruce K. Brown and D. P. Barnard[2] presents a very capable and realistic discussion of this subject. The six conclusions with which the authors conclude their paper are so pertinent that they are reproduced here:

"1. 100 octane-number aviation gasoline is now being made to the maximum capacity of the industry without regard to the economics of operating cost, transportation, or crude utilization.

"2. Under peacetime conditions, the amount of such gasoline to be required must, of course, be but a fraction of the wartime requirements.

"3. Under such conditions, the amounts which can be made within acceptable limits of economics of crude utilization and using the equipment of the war emergency program are probably not over half the present schedule.

"4. The net result is that immediately after the war a substantial portion of our emergency aviation gasoline manufacturing facilities will be discharged from war duty, and those which can be economically operated will be turned to the manufacture of motor gasoline for which they will run under conditions chosen for economy rather than maximum octane number.

"5. These and the other factors concerned in this discussion seem to indicate that immediately after the war we can expect that the average octane-number level of motor fuel may increase as much as three units over immediate pre-war values and that the majority of the marketers will be able to offer house-brand products around 80 and premium products of, say, 85 octane number.

"6. It seems fairly probable that values higher than these would result in extravagances in crude usage which will not be tolerable in the post-war period."

It will probably be just as well if this superfuel is not immediately available because the problem of adapting the engine to use it will be more than just shaving some metal from the cylinder head. Parts which will need detailed study and development include valves, pistons, rings, and bearings, as a minimum. There is a reasonable doubt as to whether merely changing the compression ratio will produce the results that are to be anticipated from the use of such a fuel if it becomes available. In all probability, the best results would be obtained by the use of a supercharger with little or no modification of the compression ratio. There seems little probability that improvement in power developed per cubic inch will be available from this source in the near future.

[2] See SAE Transactions, Vol. 52, March, 1944, pp. 87-93: "How Will the 100-Octane Aviation Gasoline Program Affect Post-War Motor Gasoline?" by Bruce K. Brown and D. P. Barnard.

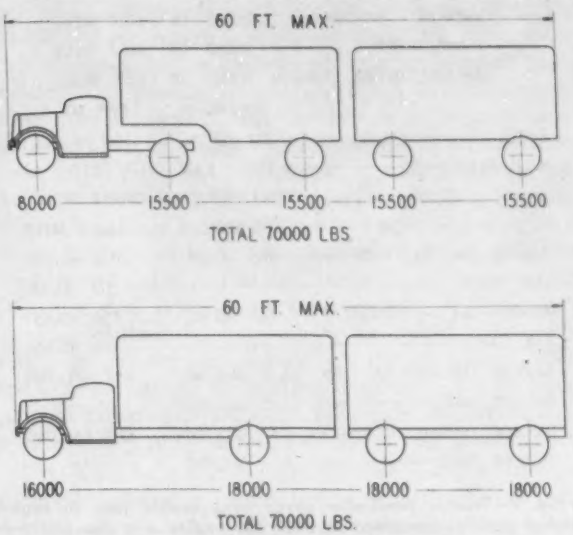


Fig. 8 – Weight distribution for other vehicles having four and five axles

Engine Power Affected by Speed

Because horsepower is related to speed, it is, of course, possible to increase the output by raising the speed. This has been done in passenger-car engines and to a lesser degree in truck engines. However, from one point of view, the engine speed is determined very largely by the axle gear ratio and the tire diameter. In consequence, while in passenger cars maximum horsepower is obtained at 4000 rpm, or better, the average cruising speed is probably not much over 2500 rpm. On a truck, the load factor being higher, the percentage of time that full load and full speed are reached is greater. For this reason and because the axle ratio is numerically higher, the engine speed in rpm is greater than in passenger cars for equal vehicle speed. From this angle, therefore, truck engines are higher speed engines than passenger-car engines.

Another item which is important is the life expectancy of truck engines as compared with passenger-car engines. Truck engines must deliver many times the mileage between overhaul and also before being worn out than is acceptable for passenger-car engines. Such details as bearings, rings, valves, valve gear, and lubrication are the limiting factors in increasing the peak horsepower speed. With some of these, material limitations present the principal problem in other design details. Generally, both are combined in varying degrees.

Specialists in all lines are carrying on research and development which have been productive of the results made evident in current practice. It can be confidently expected that progress will continue along these lines.

For the reason that smaller bore dimensions reduce dynamic loading and improve heat conditions in valves

ARDMORE — PITTSBURGH	ROUND TRIP 570.7 MILES	
PENNA. TURNPIKE	ROUND TRIP 319.2 MILES	
BALANCE OUTSIDE TURNPIKE	ROUND TRIP 251.5 MILES	
	TRIP NO. 1	TRIP NO. 6
ENGINE	377 CU. IN.	501 CU. IN.
AXLE RATIO	6.53 — 8.53	6.77
	SINGLE REDUCTION 2 SPEED	DOUBLE REDUCTION SINGLE SPEED
RUNNING TIME FULL TRIP	18.15 HOURS	15.37 HOURS
TIME SAVED		2.78 HOURS
RUNNING TIME — TURNPIKE	9.00 HOURS	7.50 HOURS
TIME SAVED		1.50 HOURS
GASOLINE CONSUMED FULL TRIP	115.15 GALLONS	115.7 GALLONS
GASOLINE SAVED	.55 GALLONS	—
GASOLINE CONSUMED TURNPIKE	61.7 GALLONS	63.9 GALLONS
GASOLINE SAVED	2.2 GALLONS	—

■ **Fig. 9 — Tractor semitrailer saved considerable time at slightly higher gasoline consumption when the engine was changed from one of 377 cu in. displacement to one of 501 cu in. displacement**

and pistons, it can be anticipated that increasing the number of cylinders either on one engine or by mounting multiple engines will receive consideration.

In any event and by whatever of these means or combinations of them the result is obtained, it can be confidently expected that peak horsepower and the speed at which it is produced will be increased.

■ Weight Affected by Materials

The second alternative for decreasing the weight penalty incurred by providing more power lies in the possibility of using lighter materials. There are a number of parts which can be made from aluminum and which already have been, successfully. These include the cylinder head, inlet manifold, cylinder block, crankcase either integral with the cylinder block or not, bearing caps, timing gear case and cover, pan, flywheel housing if separate from the crankcase, and connecting rods and pistons. In addition, there are a number of accessory items, such as the fan, water and oil pumps, and valve covers, which could be made from aluminum. Naturally, any weight which can be saved in the chassis elements outside the engine will contribute an important share in keeping gross weight down. Parts which seem to offer the most promise are the clutch housing, transmission case and covers, power take-off casing, axle gear carrier, axle housing, brake spider, brake shoes, wheels or wheel hubs, and front-axle eye beam. Hoods, fenders, and cab frame and panels also offer fields for study.

It should be stated right here that these items are not listed as either complete, on the one hand, or developed to a point that promises early adoption, on the other hand. While all of these and others have undoubtedly been made from aluminum at one time or another, all of them present problems to produce strength or stiffness or some other characteristic comparable to that existing in materials from which they have been generally made in the past. This statement is not made without recognition of the fact that in such parts as pistons, cylinder heads, and brake shoes, the superior conductivity of aluminum provides superior properties for the dissipation of heat.

However, the basic problem in connection with the use of light metals lies in the question of cost. In pre-war days, the difference in cost was so great that these materials could not be used on a competitive basis. Some information would indicate that this condition may be changed after the war, but the situation is still not sufficiently clear to hazard a prediction on what will happen.

■ Weight Affected by Operation

Another factor in decreasing the weight of the unit lies in balancing the various units in the chassis to the work to be done. To do this will require an understanding attitude on the part of the operator.

It is fairly obvious that the design must take into consideration stresses developed by the weight of the vehicle and the load for the operating conditions that will be encountered. These stresses vary in different members. The frame is principally affected as a load-carrying member. The front axle is subject to the load and to brake reaction; the rear axle to the load, to brake reactions, and must deliver engine power to the wheels. It is equally obvious that, since the value of all of these stresses is affected by shock loading, the character of the road influences the design provisions which must be made. On roads of the type proposed in the Interregional Plan, the type of road surface and the value proposed for grades, as well as the relative freedom from frequent stops in urban areas, all point to the possibility of reducing the weight of many of these units. While this does not mean the vehicles would fall apart if taken off the highway, it does mean that the operator will have to modify speeds when operating on lower-type roads to reach the desired destination not immediately located on the highway.

■ Engine Power and Operating Costs

The effect on operating costs of high horsepower is probably too frequently supposed to be all bad. Undoubtedly, this opinion is responsible to a large degree for the use of underpowered vehicles.

Several years ago, an attempt was made to obtain some information on this subject. A tractor semitrailer of 40,000 lb gvw was driven between Ardmore and Pittsburgh over the Lincoln Highway and the Pennsylvania Turnpike. Six round trips were made and the average of the odometer readings for the six trips was 570.7 miles. The distance on the Turnpike was 319.12 and on the rest of the road 251.5, or 55.9% and 44.1%, respectively.

On the first trip, an engine of 377 cu in. displacement and a two-speed rear axle having gear ratios of 6.53 and 8.53 were used. See Fig. 9. On the succeeding runs, larger engines and different axles with different gear ratios were used. For purposes of this discussion, the comparison of trip No. 1 with equipment as noted above and trip No. 6 with an engine of 501 cu in. displacement and a single-speed double-reduction axle with a 6.77 ratio, will be made.

The running time on trip No. 1 was 18.15 hr, on trip No. 6, 15.37 hr, a saving of 2.78 hr. The running time on the Turnpike was 9.00 hr on trip No. 1 and 7.50 hr on trip No. 6. Thus, 54% of the time saved was on the Turnpike, which represented 55.9% of the distance. The actual time on the Turnpike was 49.5% on trip No. 1, and 48.8% on trip No. 6, which, in view of the variables

involved, is a negligible difference. If the average speed attained on the Turnpike could have been maintained over the rest of the distance, the running time on trip No. 1 would have been 16.1 hr, and on trip No. 6, 13.4 hr. On the other hand, if the whole distance had been the same kind of a road as the portion outside the Turnpike, trip No. 1 would have required 20.8 hr, and trip No. 6, 17.85 hr.

Therefore, the high-type road would show a saving of 4.7 and 4.45 hr with the two different pieces of equipment, while the more powerful equipment would save 2.95 hr on the conventional road, and 2.7 hr on the high-type road.

The gasoline used on trip No. 1 was 115.15 gal, and on trip No. 6, 115.7 gal. Of these quantities, 61.7 gal on trip No. 1 and 63.9 gal on trip No. 6 were used on the Turnpike. If the rate of consumption for the Turnpike had been maintained for the whole trip, 110.5 gal would have been used on trip No. 1 and 113.2 on trip No. 6. If the rate needed for the ordinary road had been required for the whole trip, then 121.5 gal would have been used on trip No. 1 and 117.8 on trip No. 6. These figures seem to show that the high-type road has less effect in producing economies in fuel consumption with a powerful vehicle than with one less powerful. However, the differences between the two vehicles are so small that speculation of this nature is probably not warranted. The fact that under the conditions encountered there was practically no difference, is significant.

Without attempting to make a detailed analysis, it seems to be clear that saving of time would more than compensate for any difference in fuel consumption between the two types of equipment.

The Future

It is certain that the demand for fast, efficient transportation will continue and increase. To meet this demand, air transport will undoubtedly find a useful and distinctive place in the transportation field. To make this new service most effective will in all probability call for an alliance with highway transportation. In any event, the facts forecast a growing demand for highway transport.

Radical innovations in equipment, such as the pneumatic tire, do not seem to be in anticipation. On the other hand, changes in road structure as previously discussed would create greatly different operating conditions, and as has been discussed would form the basis for considerable change in the vehicle.

Trucks have conventionally been made as a chassis in which to mount the machinery and on which to mount a separate body. The structural advantages to be obtained from making the body the main element of the vehicle have been illustrated in the bus field. Probably all of these advantages could be secured in truck construction if it were not for the fact that there is such a large variety in the body types required. Certain of these, such as tank bodies, would seem to be readily applicable, but consideration of cost and production problems bar the way.

Clear space for loading at the rear is an important requirement in many lines of business. At the same time, the driver undoubtedly must be in front. As the engine and the driver can be associated more easily than the engine and the loading doors, rear-engine mounting does not appear attractive except, again, on a limited number of specialized vehicles.

Provisions for the comfort of the driver have an important effect on his efficiency, on the performance of the vehicle and on the safety of the vehicle on the road. Carefully designed and manufactured cabs made from metal will make it possible to improve vision by reducing the dimensions of essential structural members. In many cases, the dimensions of cabs must be increased to provide more room in all directions. Of course, a truck is designed to carry a load and with limitations on dimension and weight, there is a definite tendency to reduce sizes and weights of the vehicle in order that a maximum space and load capacity will remain. The time has come when there must be a new deal in this matter, particularly on trucks used for long distance hauling, where the driver is forced to stay in the cab for long periods. On the other hand, where the driver must get in and out frequently, the details of steps and door clearances are of greater importance, but even in these cases, cramped quarters are not good business.

Undoubtedly, passenger-car practice has had an unfavorable influence on truck design in this regard as it has had on accessibility. There seems to be a reaction against this tendency setting in on the part of those who have the responsibility of keeping "'em rolling." Most designers will sympathize with this attitude, but unfortunately it frequently happens that the man who runs the equipment is not the man who buys it, and the latter shares the common trait of being impressed by superficial appearances. It is not intended to criticize the so-called stylist and to make a case for the car designer. Rather, the purpose is to point out that these matters require the close coordination of the two skills which will produce a design that provides for the practical necessities as well as a good, common-sense appearance.

Noise is another item which calls for intensive study. Present vehicles make entirely too much racket, affecting both the occupant and the person outside. The results which have been obtained in this particular on modern streetcars should provide a powerful stimulus for work on this problem.

When Will These Things Be?

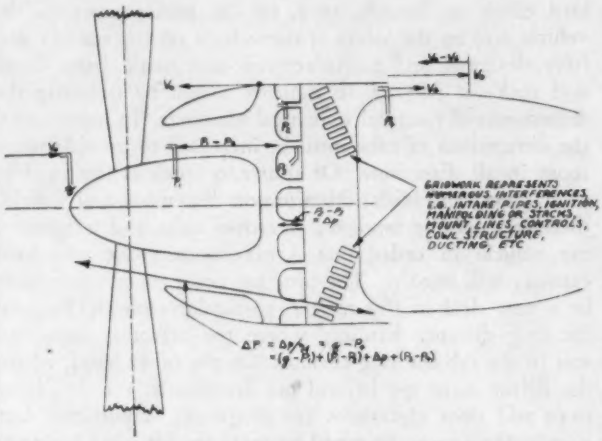
When will these things be, is a common and natural question to all of these speculations. A statement made so frequently in recent months that it is becoming as tiresome as some of the radio commercials, is, "Evolution not revolution." Probably no one is capable or willing to answer this question. Some people don't know and those who do or think they do, are not ready to commit themselves. There is a distinct possibility that someone may make a very radical departure from current practice because, undoubtedly, when the present barriers are removed, there will be keen competition to obtain an advantage in the market which will exist in the truck field. If such a novelty finds acceptance with the buyer, then others must follow or be left behind, no matter how sensible it may seem to progress slowly and certainly.

What has been attempted has been to sketch a broad picture, partly factual and partly speculative, with no effort to fill in details. From these considerations, it can be said that what has been provides a pattern for what will be. However, this does not mean that there will not be something "new under the sun," and if there is, it is to be hoped it will not be "vanity."

ENGINE COOLING FAN

by KENNETH CAMPBELL

Senior Project Engineer,
Wright Aeronautical Corp.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■ Fig. 1 – Diagrammatic sketch of cooling airflow mechanism – no fan

IT was pointed out in 1938[1] that the cooling problem, like many aircraft and engine development problems, constituted a continuous limitation to engine and airplane performance. The statement is as true today as it was then, despite great progress made in the cooling research field. This may be partly because some results have been capitalized on in terms of increased specific power output or airplane weight loading until marginal cooling is again reached; but in considerable measure it is believed that important implications of long published knowledge have been overlooked in design.

Cooling fans, to which this paper is devoted, comprise only one component of performance improvement to be had through cooling development. Table 1 shows the place that cooling fans occupy in the overall picture of cooling improvement opportunities. Research is taking place as actively as ever in each of the components of Table 1, and further substantial gain in airplane performance will undoubtedly result from each, as has occurred in the past.

But cooling fans, unlike the other components of Table 1, offer much greater and more immediate returns in airplane usefulness than is forthcoming from any of the other efforts individually. Worth-while, and in many cases very large, improvements, in rate of climb, in top speed at high altitude, in ceiling (top or single engine), and in cruising economy or, alternatively to these, in useful load are to be had on short notice from fan application.

Referring to Table 1, outlining cooling research components, Objective I is strictly an engine-development portion. Much research has been done on subitems a and b in the past 10 years,[1] but further important gains are forthcoming from current work. Particularly beneficial is the redesign of fins to recognize the rise of air temperature along the path. It has been asked, why not reduce the volume, instead of the baffle pressure drop? This is covered briefly on p. 517 of reference.[1] Generally speaking, reducing the volume requires very disproportionately increased baffle pressure drop or, with fins redesigned to avoid this, causes severe temperature rise in the cooling passages, which defeats the objective of cooling the hotter rear portion of the cylinder.

Regarding subitem I-c, the importance of keeping up the allowable head limits was published in 1940 in a fairly elaborate study.[2] This work was begun in 1934 and published, in preliminary form, in 1935.[3] It was emphasized and explained at the end of the 1940 article that an increase of 50 deg (F) in allowable head temperature limit reduces the baffle pressure drop required almost one-half for a given cylinder and baffle design. The airplane designer is well aware of this from a drag cost standpoint; the engine does not always agree, that is, when asked to endure satisfactorily 150 type-test hr. If the pilot wishes to raise temperature limits for combat periods he may, and, we know, does, and comes home.

A very great research expenditure on subitem I-d has been made by the engine companies during the past three years, particularly on the larger engines. Fuel injection into the cylinders appears still to offer the best solution. Uniform mixture strength distribution on carburetor engines is a most difficult problem. Fundamentally, the difficulty is due to the airstream's making a 70- to 90-deg bend approaching the supercharger impeller entrance. The greater the aerodynamic perfection achieved in this passage while still respecting the current axial length and bend limitations, the greater is the deviation of the heavier fuel streams from uniformity.

Objective II of Table 1 has benefited most from extensive fundamental analysis long since completed and pub-

[This paper was presented at the SAE National Aeronautic Meeting, New York City, April 5, 1944.]

[1] See SAE Transactions, Vol. 33, December, 1938, pp. 515-528: "Cylinder Cooling and Drag of Radial Engine Installations," by Kenneth Campbell.

[2] See *Journal of the Aeronautical Sciences*, Vol. 7, February, 1940, pp. 141-147: "Predetermination of the Aircraft-Engine Cooling Requirements for Specific Flight Conditions," by Kenneth Campbell.

[3] See SAE Transactions, Vol. 30, November, 1935, pp. 401-411: "Evaluation of Variables Influencing Aircooling of Engines," by Kenneth Campbell.

Table 1 – The Field of Development Opportunity for the Reduction of Thrust Horsepower Consumed in Aircooling of Engines (Exclusive of Internal Coolants)

Objective I

Reduce the baffle pressure differential across the engine required to cool:

a. By better heat transfer design of internal cylinder structure, fins, and baffles, particularly in the region of the highest temperatures or stresses.

b. By reducing any restrictions to external airflow across the engine which happen to be disposed in series with the fin passages.

c. By developing metallurgical, structural, and combustion chamber design to permit generally higher head temperatures.

d. By improving the uniformity of mixture strength and total charge distribution among the cylinders, reducing somewhat the hottest temperatures.

Objective II

Given a required pressure differential across the engine, reduce the large additional losses in the rest of the system used to obtain it:

a. By reducing the pressure drops entering and leaving the cowl or duct system and approaching and leaving the engine.

b. By adding energy to the cooling airstream within the system, discharging it rearwardly more nearly at flight velocity:

1. By a cooling fan.
2. By exhaust stack ejector action.

FAN
Theory and Practice

WHEN operating the larger engine installations under marginal cooling conditions, considerable horsepower per nacelle is frequently expended solely on cooling instead of on improved airplane performance. The numerous development opportunities for reducing cooling drag are discussed briefly by Mr. Campbell.

Cooling fans offer performance gains which, in many cases, are startling to the uninitiated, yet which can be had today for the asking.

Determination of the degree of fan boost required, which is relatively small, and methods of calculating the net thrust horsepower gain to be had from fan application are provided by the author, who also discusses briefly the question of fan controllability requirements.

Finally, Mr. Campbell analyzes the published Schicht and Wattendorf contributions to blade design for higher pressure coefficient fans, and the application of these principles to a number of cooling fan projects.

■ ■

THE AUTHOR: KENNETH CAMPBELL (M '34) senior project engineer of Wright Aeronautical Corp., is in charge of applied research and design of his company's several engine models with respect to the specialized fields of supercharging, induction, combustion and cooling. Before joining Wright in 1933, he was connected with Bethlehem Steel Co., Trent Anthracite Corp., and Sanderson & Porter, consulting engineers. Mr. Campbell is chairman of the NACA Special Subcommittee on Supercharger Compressors and a member of the Subcommittee on Heat Exchangers.

lished by the NACA.[4] In spite of this valuable analysis, however, the number of installations which place a high premium on subitem a, keeping internal cowl-loss power and resultant enormous induced pumping-loss power to a minimum, are markedly few. These losses are usually tolerated in the interest of a space-saving or a center-of-gravity compromise. But it is a compromise which is costing, under marginal cooling regimes with large engines, *several hundred thrust horsepower per nacelle*, if, indeed, this can be called a "compromise." The subject happens to be intimately bound up with cooling fans, as will be seen shortly.

Energizing the cooling airstream from within (subitem II-b) gives performance gains due to cooling drag reduction which in many regimes are startling to the uninitiated and gives some increase in performance in nearly all useful regimes. This can include cruising, according to flight-test data on one of our recent applications.

Having first taken this "snapshot" of the whole cooling field from a very high altitude, as it were, let us now come down to look at the specific target of cooling fans in more detail. We will cover theory first, then practice. The latter dates fairly continuously from 1932.

■ Cooling Fan Theory

A straightforward and simple principle of vital importance to airplane performance has been confused for years with all its many influencing variables and mathematical niceties. Rigorous and elegant expositions have perhaps left some responsible designers inappreciative of the real performance dividends from applying this simple principle to engine installations. This, we believe, has been the cause of many serious deficiencies in performance from the expected, even on recent important designs. Apologies to those steeped in the subject, to whom the above remarks do not apply. The analysis which follows is open to criticism from the meticulous on many counts; but its purpose is simplicity: to promote important performance improvement, which can still be had for the asking.

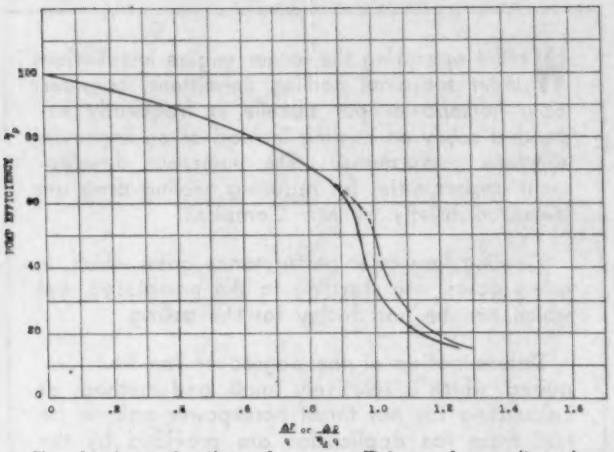
The simplest and best physical conception of the cost of cooling airflow is the overall measure of thrust horsepower consumed. The free air at zero absolute velocity is first picked up by the advancing entrance opening and is accelerated to absolute flight speed in front of the engine. Then, after using a little or a lot of the pressure energy thus created to overcome the several required pressure drops built into the cowling passages, including the engine proper, the remainder of the pressure is used to accelerate, a lot or a little, respectively, the air out in a rearward direction. But since the relative exit velocity from the system ordinarily does not equal the speed of flight (because it is generated by a pressure less than the initial flight dynamic pressure, q), the exhausted air still retains a forward component of absolute velocity after it is discharged. (Also, forget the heat or Meredith effect for the present.) This exit velocity deficiency, or forward component of absolute velocity, times the mass flow per second is an mV value, or thrust. That is, the airplane has been subjected to a net rearward thrust, or drag, in order to put forward velocity energy into the discharged cooling air which originally was at zero velocity. This thrust, acting on the plane at flight velocity, results in a certain thrust horsepower consumption, all chargeable to cooling. It is always greater, and much of the time is several times greater, than the power required merely to force the cooling air through the engine baffle passages. It can be reduced enormously in several ways.

Referring to the nomenclature of Fig. 1, which diagrams the situation just discussed, the velocities being in feet per second, this thrust horsepower is represented by the following expression:

$$\text{Cooling Thp} = (V_0 - V_4)\, mV_0/550 \qquad (1)$$

Let us next examine the thrust horsepower losses within the cowling itself. In Fig. 1, Δp is the baffle pressure drop across the engine (in pounds per square foot) required to cool for the particular atmospheric and engine conditions

[4] See NACA Report No. 592, 1937: "Full-Scale Tests of NACA Cowlings," by T. Theodorsen, M. J. Brevoort, and G. W. Stickle.



■ Fig. 2 – Approximation of pump efficiency for cooling drag evaluation

prevailing. This Δp value across a given engine model in turn meters a definite volume, Q, in cubic feet per second, determined in the engine manufacturer's laboratory and corrected for density. The power required to force the air through the engine is, then:

$$\text{Engine Cooling Thp} = Q\Delta p/550$$

As discussed following Table 1 (Objective I), the engine manufacturer's principal cooling improvement activity is devoted to reducing the Δp required and thereby the component of cooling thrust horsepower for the engine passages alone. *But this opportunity for improvement is only the beginning. It is ΔP, not Δp,* which determines the internal thp consumed, that is, the *drop in total pressure,* ΔP, or $(q - P_4)$ in Fig. 1, *across the entire internal cooling system.* Thus, the engine Δp must be divided by the first of our efficiency factors, duct efficiency, η_{id}, to convert baffle drop which the engine manufacturer finds necessary, to the drop which the airplane manufacturer must provide by virtue of his own propeller shank losses, ducting restrictions, and other turbulence losses. The total thrust horsepower devoted to internal cooling then becomes:

$$\text{Internal Cooling Thp} = Q\Delta p/\eta_d 550 = Q\Delta P/550 \qquad (2)$$

Lest this be considered a minor item, let it be pointed out clearly here that in some relatively recent airplanes today, ΔP is about double Δp; that is, duct efficiency, η_{id}, is about 50%.

Now, given, for better or for worse, a fixed engine design and a fixed installation design, what is the ratio between the internal cooling horsepower and the overall total cost of cooling as first described? The loss in relative velocity through the cowl from flight speed V_0 to V_4 gives us another way of expressing the energy expended in the internal system, which is in terms of velocity (kinetic energy loss or $\frac{1}{2}\ mV^2$) and can, therefore, be used in the comparison we are seeking:

$$\text{Internal Thp} = \frac{\frac{1}{2}m(V_0^2 - V_4^2)}{550} \qquad (3)$$

To make our comparison, we now take the ratio of this internal thrust horsepower to the total cooling thrust horsepower first determined (equation No. 1), and this turns out to be the long-famous pump efficiency:[4]

$$\eta_p = \frac{\text{Internal Cooling Thp}}{\text{Total Cooling Thp}} = \tfrac{1}{2}\left(1 + \frac{V_4}{V_0}\right) \qquad (4)$$

It is shortly going to be evident that the ratio of drop through the cowl passages and engine to dynamic head available, $\Delta P/q$, is of enormous importance to performance, and we will need to express pump efficiency in these terms instead of velocity. Since velocity is nearly proportional to the square root of the pressure differential creating it, pump efficiency as expressed in equation (4) becomes:

$$\eta_p = \frac{\text{Internal Cooling Thp}}{\text{Total Cooling Thp}} = \tfrac{1}{2}\left(1 + \frac{\sqrt{q - \Delta P}}{\sqrt{q}}\right)$$

$$= \tfrac{1}{2}\left(1 + \sqrt{1 - \frac{\Delta P}{q}}\right) \qquad (5)$$

This ends the chase for something we can use to approximate cooling drag. The important objective is still to understand the vast implications of this expression. It explains many a disappointing shortage in performance and offers many a partial fix for built-in trouble.

It is readily seen from either of these expressions for pump efficiency that if $\Delta P/q$ could ever equal 1, that is, if all of the dynamic head, q, available were dissipated in overcoming the cowl ΔP required, the efficiency of the cowl would be only 50%. This would mean that for such a condition the airplane would be deprived of *twice* the thrust horsepower actually consumed in the cowling and engine passages. If, now, we deliberately assume a number of values of the ratio $\Delta P/q$, we can plot pump efficiency as a curve against $\Delta P/q$ or $\Delta p/\eta_{id}/q$ (Fig. 2). Note the trend of rapidly decreasing efficiency or multiplying penalty in total thrust horsepower robbed from the airplane as the ΔP required by the cowling and engine becomes more and more nearly equal to the available head, q.

The above expressions for cowling efficiency become imaginary for values of $\Delta P/q$ greater than 1, which condition represents the use of induced cooling (cowl flaps) to obtain more than the available dynamic head across the cooling system. But we all know there is nothing imaginary about the increased drag encountered in the wind tunnel with the flaps open. Extending the cowl flaps into the passing airstream to obtain some depression beneath them increases the total drag even more disproportionately for the reward in increased differential across the cowling. Since our results are now being influenced by parasite drag as well as by further pumping loss energy added to the discharged air, all in the interest of cooling, and all nacelle designs differ, no accurate universal curve of pump efficiency above values of 1.0 for $\Delta P/q$ can be computed. But there is experimental evidence[4, 5] that it continues to decline at the same rapid rate for further increase in $\Delta P/q$, then probably flattening off, as indicated by the dashed line of Fig. 2, to values at best as low as 15 or 20% at 1.3 $\Delta P/q$.

Furthermore, contrary to the simplified algebra above, the cowl flaps *actually* must be opened *before* the $\Delta P/q$ required is increased to unity. The reason for the existence of a drop through the system in the first place is that a predetermined cooling airflow is necessary. Without additional drop across the cowl induced by the flaps, no flow would exist when $\Delta P = q$, which implies zero velocity out of the exit, contrary to the principal objective of achieving a flow. The flaps must begin to be used when $q - \Delta P$ leaves just sufficient total pressure, P_4, before the exit to cause the specified volume to flow out of the chosen size of exit opening (with flaps streamlined). This means that the value of ΔP at which flaps must begin to be used

[5] See "New Research on the Cowling and Cooling of Radial Engines," by James H. Brewster, III, and Richard C. Molloy, United Aircraft Corp., April, 1942.

varies with choice of normal-position exit opening area allowed. The lower line of Fig. 2 shows the effect of making an approximate allowance in the calculations for this additional pressure drop, as $\Delta P/q$ approaches unity. Some designers defeat their purpose of reducing the drag by designing the exit area so low that flaps must be opened in the more frequently encountered regimes. This applies the penalty of induced cooling when otherwise it could have been avoided.

Now, examine again the disastrous effect on cooling thrust horsepower consumption caused by excessive overall drop through cowl and engine passages. Assume, as an extreme case, that the total cowl drop, ΔP (or $\Delta p/\eta_d$), is twice what it might have been, due to unsatisfactory engine design, or cowl and ducting design, or propeller shank effects, or all three. If by this increase we find ourselves operating with the flaps open at a value of $\Delta P/q$ of 1.2 when we might have operated with the flaps closed at a $\Delta P/q$ of 0.6, the curve of Fig. 2 tells us we are saddled with a pump efficiency of only about 20%, roughly, instead of about 80%. Having previously seen that the internal thrust horsepower consumption is doubled also (proportional to the $Q\Delta P$, and Q is a prescribed constant), then the total cooling thrust horsepower with the higher ΔP is

$2 \times \frac{80}{20}$ *or eight times what it might have been with half the drop,* ΔP, *through the cowl and engine.*

This is not just another minor variable of interest to those specializing on cooling problems; the horsepower and performance costs can be extremely large. The more favorable values above are moderate enough, between 50 and 100 thrust hp per nacelle for the larger engines, depending on altitude. *That means the unfavorable case just assumed is costing 400 to 800 thrust hp per nacelle.* The gains from some recent external airplane refinements look small indeed by comparison with the possibilities being missed here. If these horsepowers are extreme for representing actual practice, it is only because the airplane manufacturer has decided not to permit the flaps to be opened enough to pay this terrific penalty. However, with far less pressure-drop contribution from the flaps, he runs into the cooling difficulties to be expected.

So much for the simple principle of built-in cooling drag and its effect. The price of large drop across engine or cowl passages alike is high. Both the engine and the airplane manufacturers are important contributors to it. To what extent can cooling fans redeem us?

First, a fan can eliminate the very disproportionate fall-off in pump efficiency accompanying high values of $\Delta P/q$, in these marginal cooling regimes cutting the cooling drag perhaps to one-third. Fig. 3 is intended to represent the same flight conditions as Fig. 1, and the same installation except for addition of a fan. Let $P_1 - P_2$ be that portion of ducting loss now included as a part of the fan structure and $\Delta P'$ be the boost added by the fan at the cylinders. $\Delta P''$ is the remainder of the original total ΔP which must then still be provided by the cowl system after subtracting the fan contribution. Further to simplify this first illustration, assume that the fan provides the *entire* ΔP required, that is, $\Delta P'' = \Delta P - (P_1 - P_2) - \Delta P' = 0$. In this simplified case, the total pressure, P_4, before the cowl exit available to accelerate the leaving air out to atmospheric pressure outside is restored to full q. Thus the leaving relative velocity, V_4, is restored to equal flight velocity, V_0, and the forward absolute velocity energy remaining in the air after it has been discharged is reduced to zero.

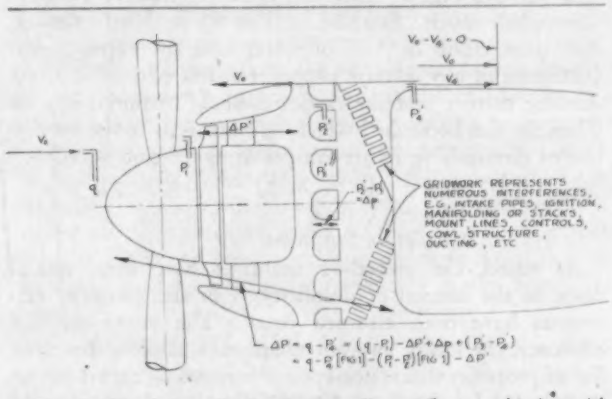
This is the physical condition representing 100% cowl pumping efficiency. (It is also evident from equation No. 4, when $V_4/V_0 = 1$, or equation No. 5, when $\Delta P''/q = 0$.) The power, $Q\Delta P$, to overcome the internal cowl and engine resistance is in this case all provided by the fan, which deprives the propeller shaft of somewhat more brake horsepower (depending on fan efficiency), this in turn depriving the propeller of somewhat less thrust horsepower (depending on propeller efficiency). Further assuming, for the moment, that the fan efficiency and propeller efficiency are equal, the propeller thrust horsepower is, by these exchanges, deprived of exactly the internal cooling thrust horsepower of the cowl and engine.

Again removing the fan, the internal cooling thrust horsepower value is restored to the propeller thrust horsepower, and the cooling air is pumped through the ducting and engine by the cowl, *but not at 100% efficiency;* far from it, at from 80% down to 20% efficiency. Thus by removing the fan we may be paying five times as much for our cooling, in this highly simplified and special case.

This probably represents the irreducible minimum of cooling drag obtainable by the addition of a fan to a given engine and installation design. Further reduction of ΔP through reduced cowl losses or reduced baffle drop would give further reward in direct ratio to this pressure-drop reduction, but *whether this is determined feasible or not,* it is seen that under marginal cooling conditions *the gain from the fan is very worth while.*

Second, in the particular case of a fan located at the cowl inlet, the pressure drop, ΔP, which must be charged against many current cowls may be greatly reduced, with the attendant large rewards described earlier. This is because pressure losses as high as $0.3q$ to $0.4q$ are occurring before the engine in a number of modern installations. The uniform-flow conditions achieved with an engine nose fan may or may not cut down these unfortunate losses; a certain portion perhaps will remain, caused by propeller shank effects. But in any event, in computation of performance with a fan installed, the loss actually occurring within the entrance is eliminated entirely from the cowl ΔP requirement, since this region becomes a part of the fan, and the loss is already included and discounted in the figure of static fan boost designed for, which boost is to be delivered at the front of the engine, not at the cowl inlet. (This explains the deduction of $(P_1 - P_2)$ from ΔP, in addition to fan boost, $\Delta P'$, in the preceding example.)

Since a fan helps most when $\Delta P/q$ is high, that is, when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■ Fig. 3 – Diagrammatic sketch of cooling airflow mechanism with fan, idealized

cooling conditions are marginal, the benefit is greatest in climb, or in level flight near ceiling. Fig. 4 represents a typical pair of airplane power-required and power-available curves. The ordinate B, as is well known, determines the rate of climb at any indicated air speed selected, since it is the power left over to raise the weight of the plane after deducting ordinate A, the power required to fly it level at the air speed. Application of the fan reduces the power available by a certain small increment; but at a high value of $\Delta P/q$, application of the fan also reduces the power-required curve by *several times this increment*, resulting in a net gain in power, B', available for climbing, with proportionately increased rate of climb. Like earnings on common stock after heavy fixed charges, a moderate improvement on a total-power basis gives a large percentage gain for climbing dividends. Frequently, a fan should improve rate of climb 20% – in extreme cases, more.

In 1942 we were all reminded quite emphatically what a fine rate of climb the German Focke-Wulf 190 had. See Fig. 5. Was it coincidence that this ship completely eliminated induced cooling and energized the cooling air by a fan? No indeed. The fan and sliding gill contributed in all probability a very substantial percentage of the rate of climb of the FW-190, and still do.

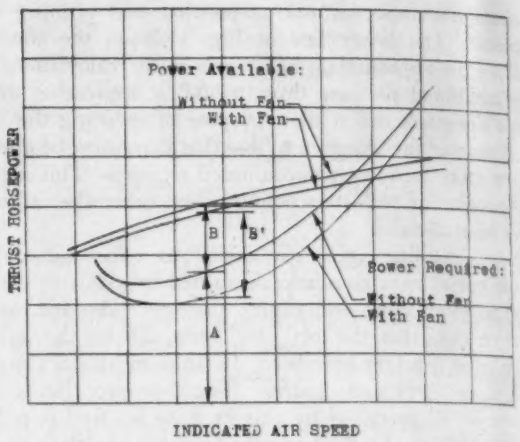
Ceiling, likewise, benefits materially from a little boost of thrust horsepower. Since ceiling also is one of the high $\Delta P/q$ operations, involving marginal cooling and reduced indicated speed, it stands to gain much, with rate of climb, by fan application. The two-engine transport, whose single-engine ceiling leaves a good deal to be desired, or the over-loaded bomber, offer interesting possibilities for application of this principle in improvement of ceiling.

It should be emphasized that it is unnecessary to overcome the *entire* cowl ΔP with the fan, leaving zero energy wasted in the discharged air. This will usually be found impracticable. When improvement of an existing installation is the goal, the lion's share of the gain from a fan is to be had in raising cowl pump efficiency from values of 0.20-0.40 up to values of 0.70-0.80. Diminishing returns in cooling power reduction are achieved at efficiencies above the latter range. This being the case, reference to Fig. 2 will assist in approximating the proper fan boost.

The proper fan boost for a nose installation is that which, after deducting both the no-fan entrance loss $(P_1 - P_2)$ (Fig. 1) and the proposed fan boost $\Delta P'$ from the total ΔP required without a fan, leaves a value of $\Delta P''$ to be provided by the cowling such that $\Delta P''/q$ lies between 0.5 and 0.6. See Fig. 2. This asks relatively little of a fan, to accomplish much. Suppose, still as a simplified example, that q in climb is 7 in. of water and the entrance loss (exclusive of any adverse propeller shank effects) is $0.15q$. Assume further that the no-fan cowl ΔP required is 9 in. Then the fan boost $\Delta P'$ which will bring us to the borderline of diminishing returns in performance gain is only:

$$\Delta P' = 9 - (0.15 \times 7) - (0.6 \times 7)$$
$$= 9 - 1 - 4$$
$$= 4 \text{ in. of fan boost}$$

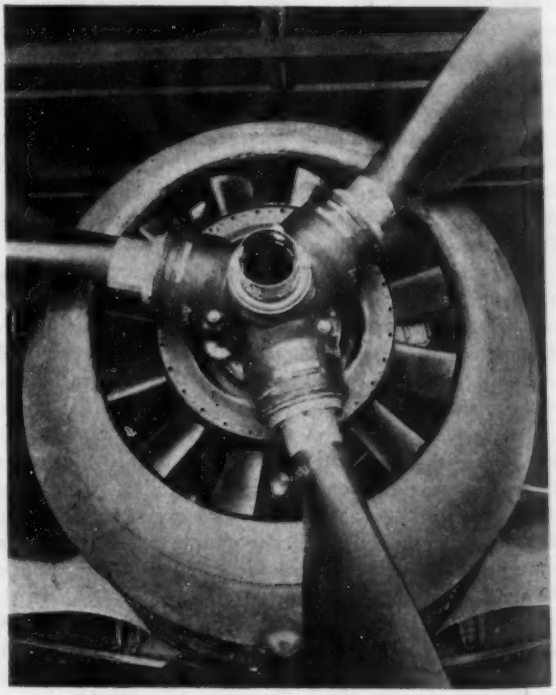
As stated, the preceding examples have been special cases, in the interest of simplicity. Fan and propeller efficiencies have been assumed equal. The lower the fan efficiency, the less of an even swap takes place in the sacrifice of propeller thrust horsepower for cooling thrust power contributed by the fan. Also, the smaller the portion of the required pressure drop the fan is asked to provide, the greater the contribution the cowl must make in relation



■ Fig. 4 – Variation of horsepower with air speed – illustration of the contribution of a cooling fan to rate of climb

to dynamic head, q, resulting in lower cowl pump efficiency, per Fig. 2. However, given the assumed values of fan and propeller efficiency, fan boost, drop across the engine and adjacent system passages, and volume of airflow, together with an approximation of cowl pump efficiency plotted against $\Delta P/q$, per Fig. 2, a logical approximation of performance gain from a fan is mathematically straightforward and easy.

First, a word of disillusionment and caution. Logically, yes, it is very easy. Numerically, and therefore factually, it is difficult. Assumptions involve guesses. Only those who have not tried it can happily assemble all the values assumed for these variables and not expect to be 20% off on one or several of them. Compensating errors, or cumulative? Who knows? Therefore, be not deceived by the following formulas. They are honest enough, but the real task is to find the numbers to put into them. Some fan evaluation studies are going to be made in the early stages



Courtesy, British Air Ministry

■ Fig. 5 – Cooling fan on Focke Wulf 190

which are going to miss the mark, but at least not by several hundred per cent.

Q = Volume of airflow, cfs

$\Delta P = \Delta p/\eta_d$ = Required drop in total pressure across entire ducting system, including engine and propeller-shank losses, lb per sq ft

Δp = Engine baffle drop required, total pressure entering baffle passage to static behind, lb per sq ft

η_d = Cowl duct efficiency, that is, ratio of drop across engine alone to total pressure drop through entire cowling or ducting system

$(P_1 - P_2)$ = Pressure drop incurred between cowl entrance and engine face, lb per sq ft

$\Delta P'$ = Fan boost (static pressure only), lb per sq ft

$\Delta P'' = \Delta P - (P_1 - P_2) - \Delta P'$ = Portion of cowl drop, ΔP, which must still be provided by cowling when fan is helping, lb per sq ft

η = Propulsive efficiency

η_f = Fan efficiency

η_{cp} = Cowl pump efficiency when providing cowl drop ΔP

η''_p = Cowl pump efficiency when providing only cowl drop $\Delta P''$

q = Dynamic head from forward flight speed, lb per sq ft

With the cowling and no fan, the thrust horsepower charged to cooling is

$$\text{Thp}_c = \frac{Q \times \Delta p/\eta_d}{\eta_p \times 550}$$

With the fan, the Thp charged to cooling is in two components:

a. Thp taken from the propeller by the fan.

b. Thp due to deficiency of cowl exit velocity, as in no-fan case, but reduced in magnitude.

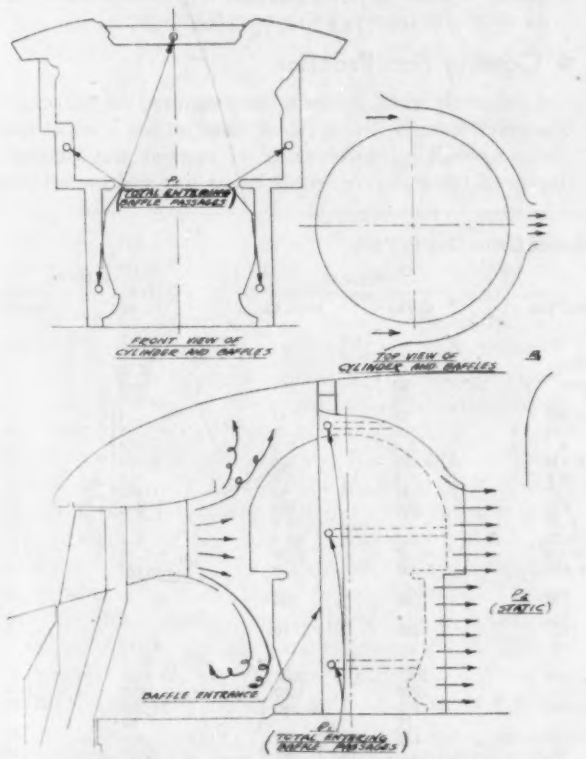


Fig. 6 – Sketches illustrating the necessity for basing performance gains on static pressure boost

$$\text{Thp}_{fan} = \frac{Q \times \Delta P'}{550} \times \frac{\eta}{\eta_f}$$

$$\text{Thp}_{cowl} = \frac{Q \times \Delta P''}{\eta_p'' \times 550}$$

The difference between the cooling Thp for the no-fan and the fan case is, then:

$$\text{Thp Gain from Fan} = \frac{Q}{550}\left[\frac{\Delta p/\eta d}{\eta_p} - \left(\Delta P' \times \frac{\eta}{\eta_f} + \frac{\Delta P''}{\eta_p''}\right)\right]$$

The gain may be made to come out a loss if low values of $\frac{\Delta p/\eta_d}{q}$ required are selected, combined with adverse values of fan efficiency.

In estimating the gain from a fan, *the static-pressure boost should usually be used,* not the total-pressure boost. Fig. 6 represents a typical nose fan installation. Bearing in mind that the engine fin and baffle passages constitute the air meter, the pressure differential which causes the required amount of cooling air to flow is total entering the baffle passages (at all points) minus static pressure of the chamber into which the baffle orifices discharge. Cooling fan blades are necessarily short radially. Referring to Fig. 6, it is possible to achieve a high velocity impinging on the baffle entrances directly opposite the fan, and the flow in this region will benefit. But the lower half of the cylinders and the crown fins of the head, cannot simultaneously be subjected to this velocity of approach. Unless there is conversion to static pressure after the fan, any gain in pressure differential across at least half of the baffle passage area will be very small. Not only will cooling suffer, but airflow will be below expectations, and below that for which the fan was designed.

With the pusher or exit discharging fan, the cooling air discharges at atmospheric static pressure. If, without a fan or induced cooling, baffle pressure differential is sufficient for cooling, then the fan may be largely a velocity producer and still be fully effective. Otherwise, an increase in static depression behind the baffle passages is necessary, and the attainment of this is dependent on the *static* boost of the fan.

There are a number of additional refinements which, if allowed to, can confuse the answer. One is the effect of heat energy, or Meredith effect. Just as the fan adds energy to the airstream, so does heat rejected from the cylinders, exhaust manifolding, or stacks. Rejection from cylinders is known over the mass flow, density, power, and mixture range.[6] Not so the rejection from exhaust systems, which are all different and disposed differently in each installation with respect to the air flowing over them.

The mechanism of the gain from heat is simple enough: For a prescribed mass flow, the temperature before leaving the exit opening is increased and, therefore, density decreased. With the given pressure, P_4, available to force the air out of the exit to atmospheric pressure, acceleration of the lighter air is higher, and velocity, V_4, is higher. Since mass flow is, by cooling requirement, the same, mV_4 is higher and the thrust deficiency $m(V_0 - V_4)$ at the exit is reduced. The exit area to achieve the same mass flow at the same q and P_4 is, of course, larger in the heated-air case.

The problem can be further complicated by recognition of the fact that mass flow for a given pressure drop across the engine declines materially with increased temperature

[6] See Wright Aeronautical Serial Report No. 831 (unpublished).

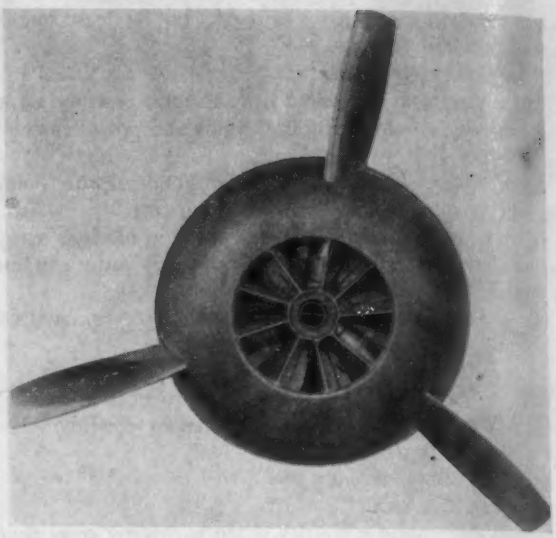
rise of the air along the path through the fins, and this occurs with variation in either altitude, power, or mixture variation to a varying degree depending on the initial mass flow considered. The effect has been investigated theoretically and on a Wright engine but is a subject in itself.[6]

Facing all the above variables and their influence on the performance result, we make a practice of calculating gain from a fan on a step-by-step basis. That is, taking Fig. 1 and Fig. 3, the conditions of pressure, temperature, flow, and velocity at each of the strategic points in the system are calculated as needed, and the final cooling thrust horsepower is arrived at arithmetically. However, in cases where $\Delta P/q$ is greater than that permitting streamlined flaps, the pump efficiency approximation of Fig. 2 still has to be used, although this is unnecessary when $\Delta P/q$ required is less than this value.

Table 2 demonstrates this procedure for a hypothetical case, the last figures in the columns representing the brake horsepower devoted to cooling in climb with and without a fan at sea level, 20,000 ft, and 30,000 ft. For reasons of military secrecy, these examples have deliberately been invented and do not represent any airplane or engine whose characteristics we know.

An important finding influencing fan and/or engine design policy springs from many actual studies similar to the above example. There is a range of sea-level fan boost, say, between 4 and 10 in., over which at normal values of cowl ΔP the net horsepower devoted to cooling does not change to an important degree. With the lower values of this boost range contributing large gains in performance up to intermediate altitudes from 25,000 to 30,000 ft, the higher values of boost encountered with the same fan at sea level also give performance reasonably near optimum. This means that for this altitude range a controllable fan usually is not needed.

The above is not true for a useful 40,000-ft fan, as indicated later. It should be obvious by now that the range of requirements and variables is too vast to establish any specific altitude above which the engine manufacturer should provide a controllable fan. This is a matter requiring good judgment.



■ Fig. 7 – 1932 "spinner cowl" fan

In analyzing a specific airplane-engine cooling fan installation, the added weight of the fan installation, and therefore a slight increase in airplane gross weight or horsepower required to fly, must be taken into consideration. Propeller-speed fan installations will average an additional 40 lb per engine and may be estimated to require approximately an additional 16 bhp per engine in level flight to carry this weight. For engine geared fan installations, the added weight per engine over a conventional standard configuration will average approximately 60 to 80 lb on a typical high-horsepower low- or medium-altitude installation (30,000 ft) and 120 to 140 lb for a high-altitude installation (40,000 ft).

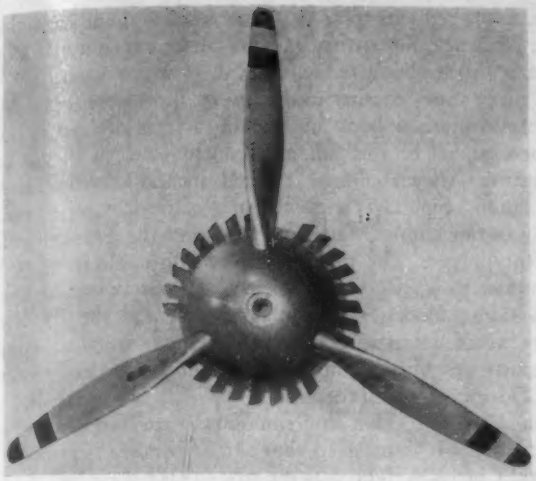
So much for theory; let us turn to practice.

■ Cooling Fan Practice

A relatively small fan boost was required for the original objectives, namely 3 to 4 in. of water at sea level, at rated rpm. As cooling performance of engines was constantly improved through more and deeper fins and more techni-

Table 2 – Analysis of Performance Gains Due to Fans

Altitude	Sea Level			20,000 Ft		30,000 Ft	
Configuration	No Fan	4-In. Fan	10-In. Fan	No Fan	10-In. Fan	No Fan	10-In. Fan
Ias, mph	150			150		150	
q, in. of water	11.2			11.2		11.2	
P_0, in. of mercury	29.92	.....	.....	13.75	.....	8.88	.....
Oat (Free Stream), F	100	.....	.....	28	.....	−8	.....
V_0 (True), fps	228.5	.....	.....	314	.....	376.5	.....
t_2 (In Front of Engine), F	104	106	109	38	41	3.4	8.4
q Recovery Assumed, %	75	85	85	75	85	75	85
P Entering Engine or Fan, in. of water	8.4	9.5	9.5	8.4	9.5	8.4	9.5
σ Corresponding	0.941	0.942	0.944	0.503	0.506	0.356	0.359
Bpd (Δp), in. of water	6.5	6.7	6.9	7.2	7.6	8.5	9.0
Temperature Rise through Engine, F	79	78	77	94	92	102	99
Temperature at Exit, F absolute	643	644	646	590	593	565	567
Fan Boost, Actual, in. of water	.....	3.77	9.28	.....	4.9	.....	3.42
P_2 Entering Engine, in. of water	8.4	13.27	18.78	8.4	14.4	8.4	12.92
P_4 at Exit, in. of water	0.92	5.56	10.84	0.12	5.66	−1.38	+2.57
σ_4	0.807	0.819	0.826	0.405	0.415	0.270	0.278
$V_4 = \sqrt{\frac{P_4 \text{ in. of } H_2O}{\sigma_4} \times \frac{3970}{60}}$	71	172	240	36	244	.....	201
$V_0 - V_4$ (Velocity Deficiency, Minus Denotes Excess)	158	56	−11.5	278	70	.....	175
$\Delta P/q$	.....	.....	.....	.....	.....	1.112	.....
Pump Efficiency, %	.....	.....	.....	.....	.....	21.5	.....
$\sigma_4 \Delta p$	5.25	5.40	5.56	2.92	3.08	2.36	2.50
Weight Flow, lb per min	1705	1735	1760	1220	1300	1080	1125
Volume Entering Engine or Fan, cfm	23,700	24,100	24,400	31,700	32,600	39,700	41,000
Propeller Efficiency Assumed	70	.....	.....	75	.....	80	.....
Cowl Drag Bhp (Minus Sign Denotes Thrust)	83	30	−6.2	134	36	456	93
Fan Efficiency	.....	60	60	.....	60	.....	60
Fan Bhp	.....	24	60	.....	42	.....	37
Total Cooling Drag Bhp	83	54	54	134	78	456	120



■ Fig. 8 – 1934 spinner fan

cally correct baffling, thus greatly reducing the baffle pressure drop required to cool a given power output, engine cooling kept pace with the increased outputs of the successive engine models, and the expected demand for fan assistance never materialized. Nevertheless, down the years considerable development work on fans was consummated.

In 1932 a propeller-speed internal-flow "spinner cowl" was built and completed a 50-hr endurance test on a 650 hp two-row engine, for which it was designed. Fig. 7 is a photograph of this spinner-type fan, which, it will be observed, took in air through a hole in the nose, discharging it axially from the rear of a spinner to a substantially larger diameter. Internal vanes were, of course, provided. The design turned out to be quite a severe structural problem, and considerable development, ultimately successful, had to be carried on to make it stay together.

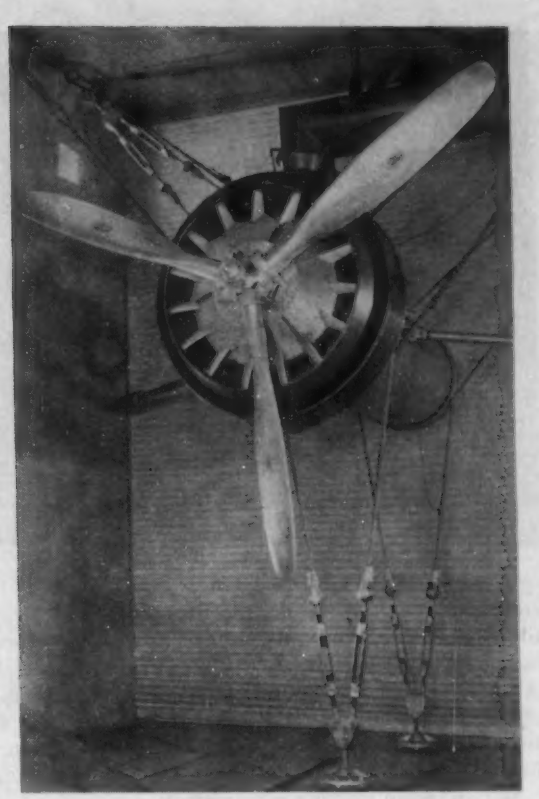
This fan was followed in 1934 by a second design for the same engine, in which the blades were put at the periphery of a somewhat smaller spinner, photographed in Fig. 8. In many ways it resembles modern spinner fans of today and was probably the first of this type; unfortunately, the blades were not designed aerodynamically.

In 1935 design studies were made of the pure centrifugal-blower type for a 750 hp single-row radial engine, but the large inlet, structural complexity, and weight required for a strong enough blower of high capacity, coupled with desirability of a vaned diffuser outside of the blower, appeared discouraging, and the layouts were not released.

In 1936 a propeller-speed fan for a 9-cyl 800 hp radial engine was released, and in 1937 was tested with the engine on a test stand. Fig. 9, previously published,[1] is a photograph of this fan installed for test. The fan was designed to give 3 in. of water boost at sea level and 2300 engine rpm, attached to a propeller running at 11/16 of engine speed. Actually, the fan gave 4 in. under this condition. The blades were made of micarta molded with an integral, oblique palm, which was bolted to two dished aluminum cones. The blade design was by the method of O'Brien and Folson[7] except that the annulus swept by the blades was designed to decrease in area along the air path by virtue of the spinner's having a larger O. D. at the trailing

[7] See Transactions of the ASME, Vol. 57, October, 1935, pp. 383-388; "Design and Performance of a High-Pressure Axial-Flow Fan," by L. S. Marks and T. Flint.

[8] See NACA Technical Memorandum No. 927, 1940, "Constant-Pressure Blowers," by E. Sorensen.



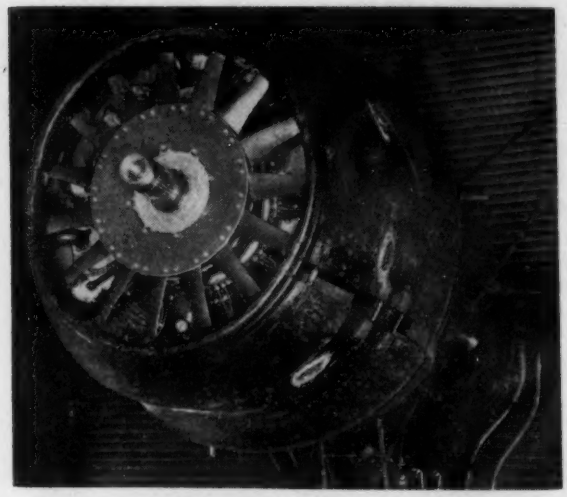
■ Fig. 9 – 1936 propeller-speed fan

edge of the blades than at the leading edge. (This has subsequently been called the Schicht principle of design.[8]) The fan had 15 blades, 5 in. high, of RAF 6 section, 12% thickness ratio, and was 45 in. tip diameter. The low solidity of the fan was by present standards a mistake, although adequate for the requirement. The new cylinders for which this fan was designed turned out to cool at very favorable baffle pressure drops, and there was no demand for the fan to meet the then current objectives.

The next development was a fan for an early model 2000 hp engine, anticipating that a fan might be of interest in connection with protracted water take-offs or slow, single-engine climb. Calculations similar to those for the preceding fan for the 9-cyl engine showed that a propeller-driven fan would be a much more difficult assignment, even for obtaining 4 in. of fan boost. The methods of blade design then in use, based on Keller,[9] supposedly limited the boost which could be obtained for a given speed (pressure coefficient) and the existence of other methods for obtaining high pressure coefficients (discussed later in the paper) was not yet appreciated. The propeller speed was only 7/16 of engine speed, as compared with 11/16 in the case of the 9-cyl engine. Further, the volume of flow was to be nearly doubled, with diameter remaining unchanged, thus further necessitating higher pitch angles and more critical fan performance. Speeding up the fan would overcome all these difficulties. Thus, in 1939 an early gear-driven fan was designed, which was tested on a running engine in 1940. Fig. 10 is a photograph of the installation.

This fan had several novel structural features of interest in connection with today's gear-driven fans. The fan ran at crankshaft speed by means of a planetary step-up gearing

[9] See "The Theory and Performance of Axial-Flow Fans," by Curt Keller and L. S. Marks, McGraw-Hill, 1937.



■ Fig. 10 – First gear-driven fan, installed for test – designed in 1939

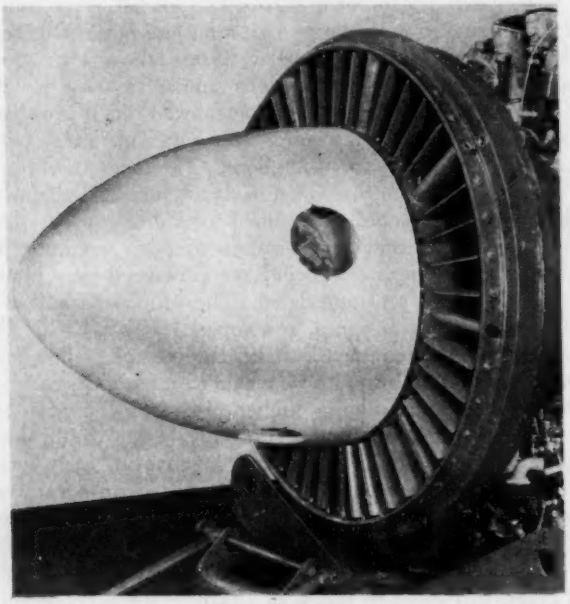
surrounding the propeller shaft in the nose. As before, the blades were made of micarta formed to the desired aerodynamic dimensions, but each blade terminating at the root in an integral projecting micarta block, suspended centrifugally within two forged steel rings at the front and rear of the fan blades at the root. The fan gave 4 in. of water additional boost on the front of the engine at 2400 engine rpm, again meeting requirements.

The next fan built was for 40,000-ft altitude performance, in preparation for an airplane project which was subsequently canceled. Design was initiated in 1941 and fabrication completed in 1942. Recognizing that the entire ΔP need not be provided by the fan, a 40,000-ft boost of 5 in. of water at 80,000 cfm was chosen, which, without controllable features, reflects over 300 hp at sea level. An extensive fan and drag evaluation study[10] at varying boosts from sea level to 40,000 ft showed that if the fan boost could be reduced to 50% at sea level without impairment of efficiency, no further gain in performance would accrue from further boost reduction at the lower altitudes.

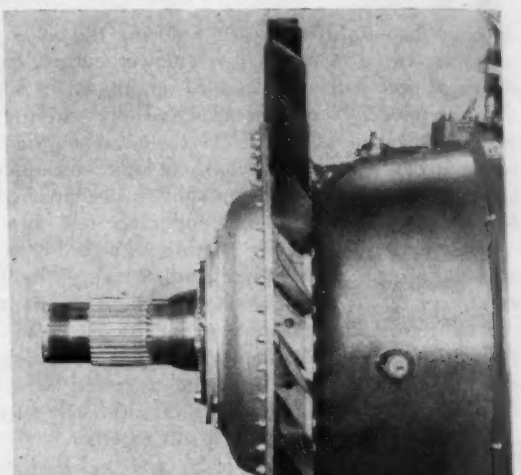
Since vanes counter to the fan rotation were found desirable to increase boost at altitude, it was decided to make these ahead of the fan and controllable, so that at low altitudes they could be twisted in the direction of fan rotation. Fig. 11 is a photograph of the completed assembly on an engine, and Fig. 12 shows the gear-driven fan rotor proper. The pretwisters are adjustable from 45 deg counter to 45 deg with rotation. The pretwister assembly, including blades, is cast magnesium, and the fan blades are again micarta, per Keller method, installed as previously described, though shorter and more closely spaced.

During this last-mentioned project two published[8, 11] fan blading design contributions were brought to our attention. It is believed that the St. Louis Airplane Division, Curtiss-Wright, was the first to take these two theories seriously enough to go ahead with design and construction

[10] Wright Aeronautical Project Report S-809 No. 9 (unpublished).



■ Fig. 11 – Adjustable pretwister assembly – high altitude gear-driven fan

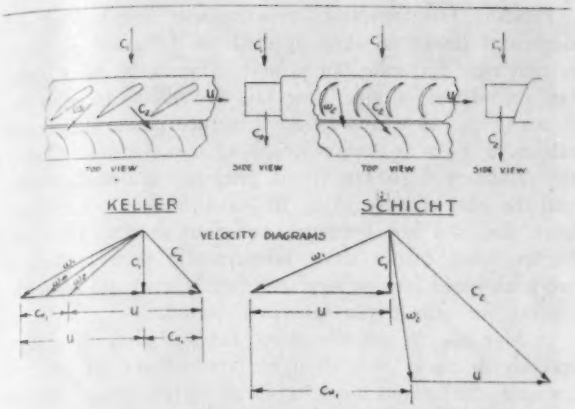


■ Fig. 12 – High-altitude gear-driven fan rotor

of an aircraft fan model incorporating them. This fan model was tested in a wind tunnel.

The two published theories mentioned are those of Schicht and Wattendorf. The Schicht theory[8] claims that the breakaway from the blades of a Keller-method designed airfoil fan at high attempted pressure coefficients can be eliminated by not permitting the relative velocity ω (Fig. 13), through the blades to slow down as it normally must do according to the geometry. Instead, Schicht holds that ω_2 leaving the blades should be maintained equal to the entrance value ω_1 by reducing the radial height of the blades along the path. Then, says Schicht, the lift coefficient or curvature of the blades can be enormously increased and more energy put into the air without separation. As the velocity diagram indicates, this means that virtually all the energy appears as velocity C_2, and if static

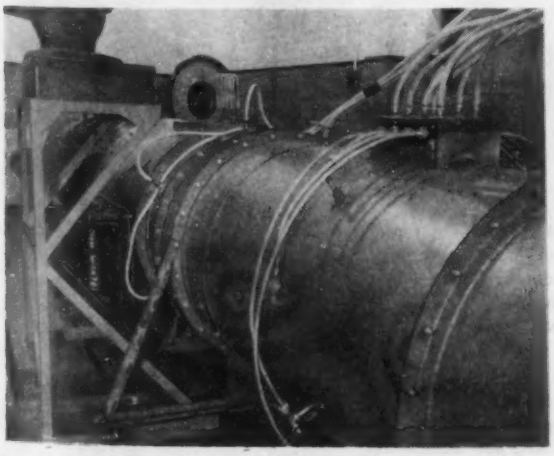
[11] See "Theodore von Kármán Anniversary Volume – Applied Mechanics," California Institute of Technology, 1941, pp. 285-292: "The Ideal Performance of Curved-Lattice Fans," by Frank L. Wattendorf.



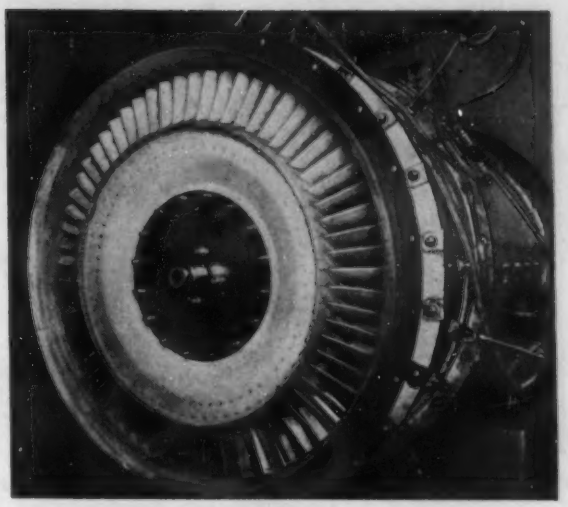
■ Fig. 13 – Sketches illustrating the Schicht principle of maintaining relative velocity magnitude, permitting increased lift without breakaway

pressure boost is desired, efficient diffusion must follow, as in supercharging. For our purposes, then, if some "Schichting" is found necessary, its degree should be compromised with the pressure coefficient (boost for a given speed) and static pressure boost planned. We have since found that a small degree of "Schichting" goes a long way, if done properly.

In 1941 Dr. Wattendorf published[11] a short treatise on fan-blade design in which, even without "Schichting," he demonstrated actual test points which reflected, again on a total-pressure basis, pressure coefficients substantially higher than the Keller limitations imply can be obtained in a single, axial fan stage. Keller[9] finds that the minimum spacing of blades must not reflect a chord/pitch ratio of over 1.1 and, simultaneously, that the maximum lift coefficient expected of airfoils must not be over 1.0 on an infinite-aspect-ratio basis. Wattendorf found he could go to chord/pitch ratios of at least 2.0 (we have since found that this depends a good deal on the entering blade angle and pressure coefficient) by lining up the leading and trailing edges of the blades with the entering and leaving angles of relative velocity, ω_1 and ω_2, calculated, and then joining these by a curve. This procedure is similar to Hagen's (Sturtevant) practice employing a simple arc, which he has used a long time and published.[12] Both designers calculated boost merely by applying velocity diagrams, together with assumed efficiencies, to the simple Euler theorem, on which all axial-fan designs are based.



■ Fig. 14 – Paterson fan laboratory – tunnel at fan station



■ Fig. 15 – Three-quarter-scale model (30 in.) mounted for test – Paterson fan laboratory

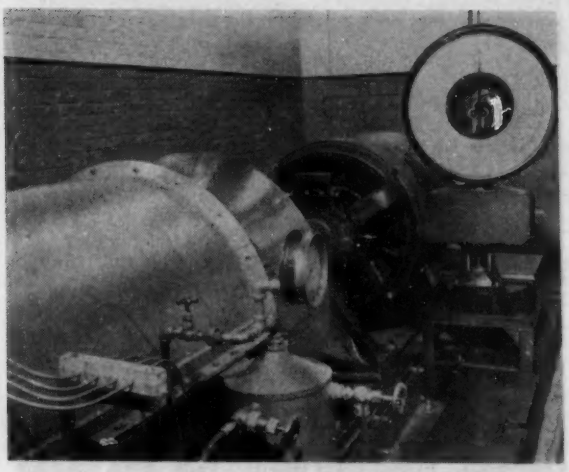
Reference Wattendorf publication also demonstrated mathematically that the shape of the pressure coefficient curve (again on a total-pressure basis) plotted against volume is declining, flat, or rising, depending simply on the magnitude of the design pressure coefficient; those fans with very high pressure coefficients having rising, while low pressure coefficient fans have declining, characteristics with increasing volume.

On receipt of design information concerning the fan built by Curtiss-Wright at St. Louis, it was decided essential immediately to determine how far the curvature of this so-called "stream filament" blading can be carried without breakaway for high total-pressure boost for various conditions and, equally important, what are the limitations of this approach in terms of *static-pressure conversion achieved within a reasonable axial travel.* These objectives involve research on control of tangential acceleration of the air, and expansion between the blades, as influenced by blade curvature, blade shape, and degree of "Schichting," both for rotor and stator vanes, for fans of low and high capacity (entering blade angle).

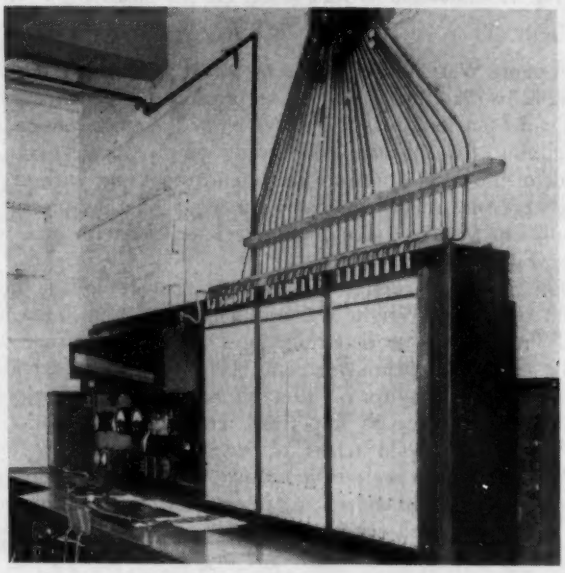
We have had operating at Paterson a fan test laboratory since the late summer of 1943. Reference is made to photos, Figs. 14, 15, 16, and 17, of the laboratory. Three-quarter-scale fan models (30 in. diameter) are driven by a 75-hp dynamometer motor at tip speeds up to 500 fps. Both low pressure coefficient blading (for gear-driven fans) and high pressure coefficient blading (for propeller-speed fans) are studied, as well as stationary aftervaning.

Unlike our earlier fan designs, we believed that vanes of constant thickness (sheet metal), making for rugged but light design, should be equally as effective as streamlined shapes, the rate of expansion of path between blades or the rate of tangential acceleration along the path being taken care of by technically correct blade curvatures, not necessarily circular arcs, and by "Schichting." These ideas are taken directly from similar design research on supercharger entrance blades, impeller passages, and diffuser vanes, conducted partly by ourselves but also by a number of other companies and the NACA over the past several years. All fan models procured thus far for laboratory testing have

[12] See "Mechanical Engineers' Handbook," by L. S. Marks, McGraw-Hill, 1941, pp. 1945-1960: "Centrifugal and Propeller Fans," by H. F. Hagen.

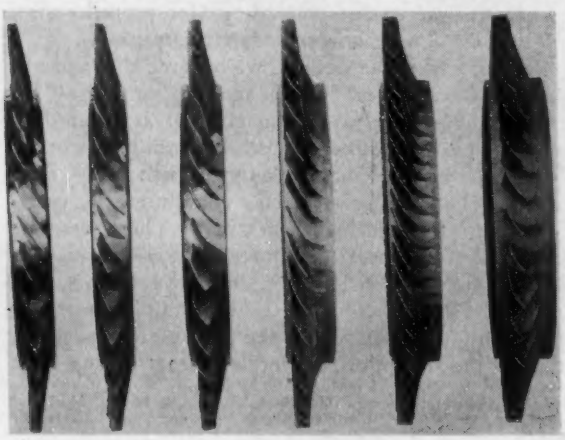


■ Fig. 16 – Throttling cone and dynamometer – Paterson fan laboratory



■ Fig. 17 – Control panel – Paterson fan laboratory

been designed on this basis. Fig. 18, which shows six of the twenty-two 30-in. fan models tested thus far, illustrates the progressive increase of blade curvature or fan boost; and Fig. 19 shows two fans designed to the same requirements, one "Schichted" and the other not "Schichted."



■ Fig. 18 (left) – Rotors showing increasing degrees of boost of 30-in. models

Pending fan laboratory procurement and testing, our unproved thoughts were applied in the construction of several new full-scale fan projects. First, the rotor of the last gear-driven engine nose fan described earlier in the paper (Fig. 12) was redesigned to incorporate sheet-metal blades. Fig. 20 is a photograph of this revision. Due to the relatively high fan speed (engine crankshaft speed) and the goal of only 10 in. of static-pressure boost at sea level, this is a low pressure coefficient design. However, higher boost blades have subsequently been applied to propeller-speed fans on airplanes (see below) and are forthcoming for gear-driven fans as required.

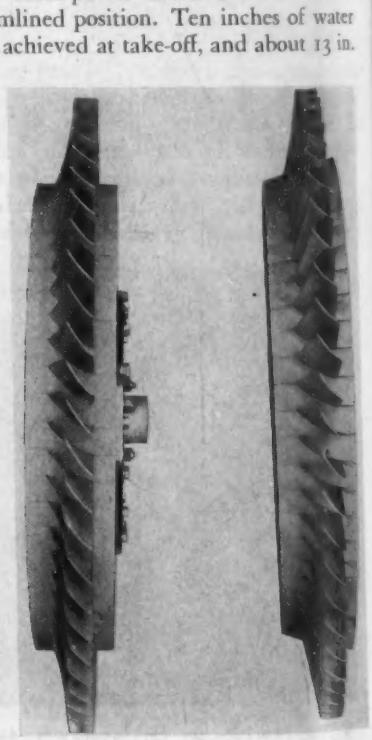
In May 1943 a propeller-speed fan and aftervane design embodying these new thoughts was offered to improve cruising performance of a large flying boat (Martin Mariner PBM-3) powered with two Cyclone 14's. With the ship flying at far greater load than its original design value, severe induced cooling penalties such as those explained earlier were being encountered, resulting in unsatisfactory cruising performance.

It was planned that the fan should give 4 in. of additional boost in cruising operation at 2100 engine rpm at 4000 ft on a hot day, with propeller turning at 0.4375 of engine speed. It will be noted that this is considerably more difficult performance than it was desirable to try for in the early days with a propeller-speed fan.

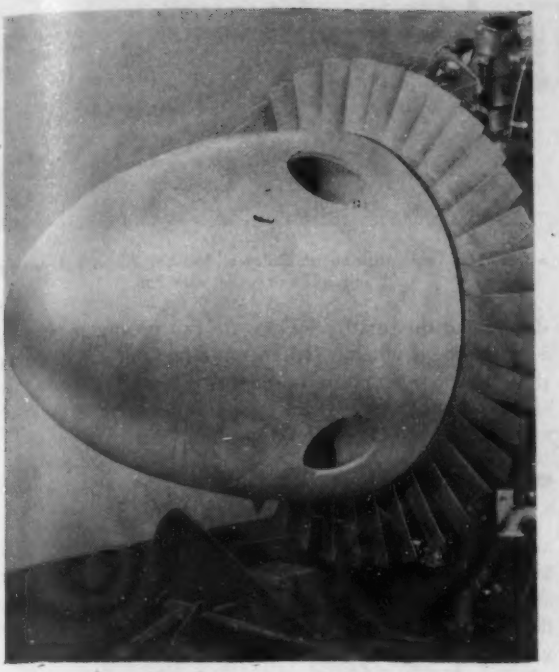
Complete cooling tests, as well as tests to determine the effects of two fan designs on various items of the airplane's performance, were made.

The first fan blades fell short of giving the required boost, because the helical component of approach due to the propeller was not allowed for sufficiently. The first fan was also tested with and without a large spinner (32 in. O. D.), and as no difference in performance could be detected, the production version was built without it.

Cooling and air speed in cruising are both distinctly improved with the 4+ in. of boost obtained. In addition, taxiing, take-off, and rated-power climb are now made with flaps in the streamlined position. Ten inches of water baffle pressure drop is achieved at take-off, and about 13 in.



■ Fig. 19 (right) – Two 30-in. models – right: "Schichted;" left: not "Schichted"



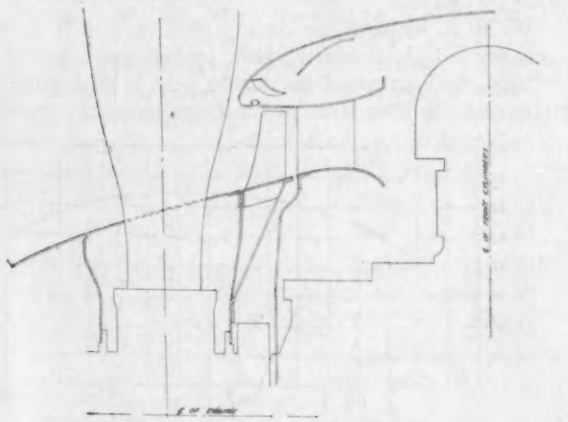
■ Fig. 20 – Sheet-metal gear-driven fan

of water baffle pressure drop is achieved in rated-power climb. Of the latter, about half accrues from the fan. Pressures around the engine are quite uniform.

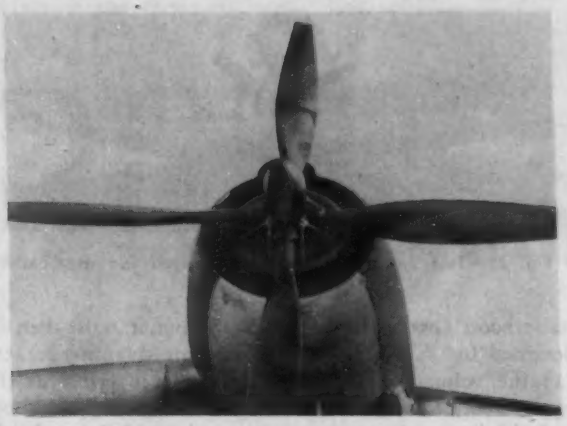
Fig. 21 is a sectional view of the installation. Note the small degree of "Schichting," all at the root end of the blade. Also note the very high rotor blade entrance angles necessary and the trailing angle of the blade at the root, which curves around to discharge the air slightly in the direction of rotation. The diffuser is built along the lines of supercharger diffusers, that is, a gradual increase of area at first, as influenced importantly by the sine of the vane angle, gradually increasing along the path. Figs. 22, 23, and 24 are photographs showing fan and installation itself. This fan is currently in production, and squadrons have been using it in active service for some months.

Just prior to the above project, a propeller-speed fan for a Cyclone 9 was installed in a test ship. Figs. 25 and 26 are photos of the installation, and Fig. 27 shows the rotor alone.

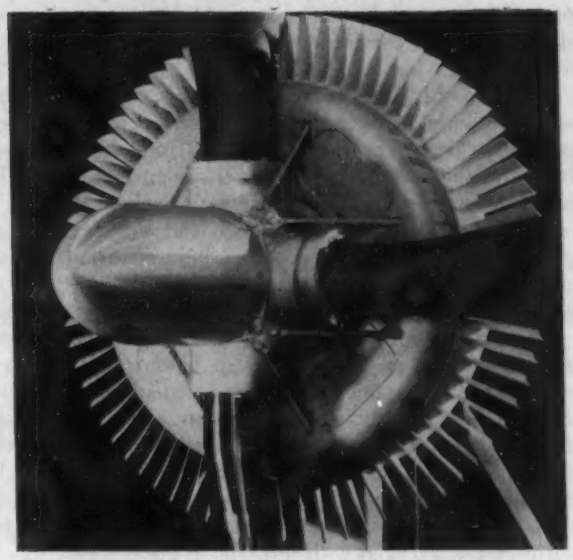
The fan, designed before the one for the flying boat,



■ Fig. 21 – Layout of propeller-speed engine cooling fan on Martin PBM-3 flying boat



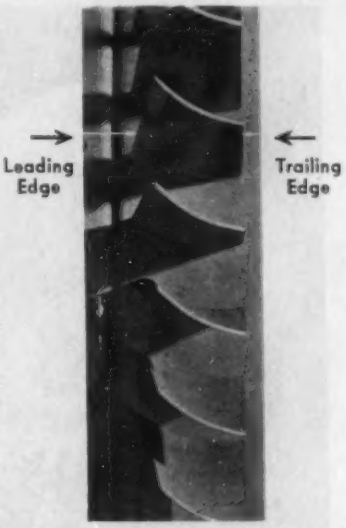
■ Fig. 22 – Propeller-speed fan installed on Martin PBM-3



■ Fig. 23 – Close-up of propeller-speed fan for Martin PBM-3

was built to an easier assignment in spite of being for a higher boost. The propeller runs at 0.667 of engine speed, instead of 0.437, and the volume of cooling airflow is reduced 35%. It should be pointed out, however, that this is not truly a high-altitude fan. It was designed for low and intermediate altitude work to give 10 in. of boost at rated power, sea level, and about 5 in. at 20,000 ft. This fan should be of particular interest in transport operation.

A higher-altitude propeller-speed fan for an engine of this size and gear ratio is perfectly feasible – say, one which would give 4 or 5 in. of



■ Fig. 24 – Martin PBM-3 blade runner

■ Fig. 25 – Test airplane with propeller-speed fan at Caldwell-Wright Airport

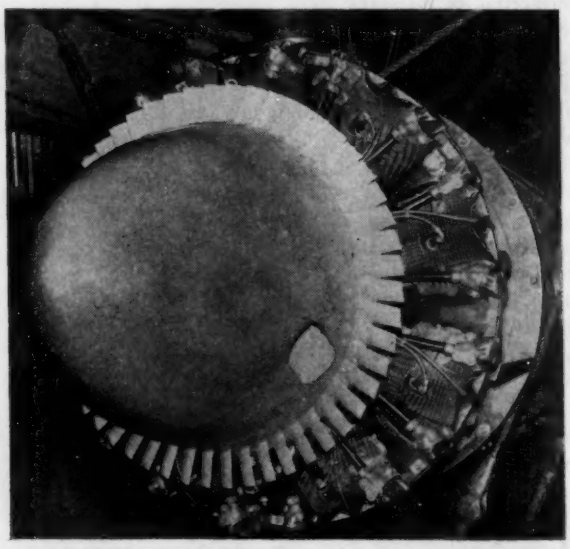
■ Fig. 26 – Test airplane at Caldwell-Wright Airport showing sliding gill exit used with fan

static boost (not total) at 30,000 ft – but it must then be designed for a very much increased air quantity because (a) the volume increases with altitude inversely as the square root of the density for a given baffle pressure drop, and (b) intercooler cooling air must be added (or should be) to the fan capacity. This would call for higher, steeper, and more curved blades and would be a more limiting design case, like the fan described above for application to a flying boat.

Tests on this fan are still proceeding; an identical fan structure without blades has been provided for flight comparison to determine the actual gain from the fan boost. It is also intended to compare performance of the entire fan installation with that of a similar airplane having the original installation but using this later engine. Fig. 28 gives pressures thus far achieved in flight at various sliding gill openings. Examination of these indicates that the fan has a satisfactory noncritical pressure-volume characteristic.

Structurally and mechanically the sliding gill exit opening, as opposed to the hinged flap type, is working out optimistically with this specific fan installation. It appears that the full-open sliding gill does not impose any additional penalties on airplane performance, while at the same time it allows full use of fan performance to achieve additional engine cooling. Here, again, more flight testing must be done and is planned to determine optimum rewards and deficiencies with this type of installation.

Many questions have been raised with regard to the icing problems which may be encountered with cooling-fan installations in combat or service conditions. All actual fan installations achieved to date have no provision for anti- or de-icing equipment, and much thought is being devoted to this matter. Tests are proceeding to determine what happens when a fan encounters icing conditions, and the development of anti- or de-icing devices must necessarily follow this. It is an important subject and covers a very extensive field of development.



■ Fig. 27 – Propeller-speed cooling fan rotor on engine for test

■ Conclusion

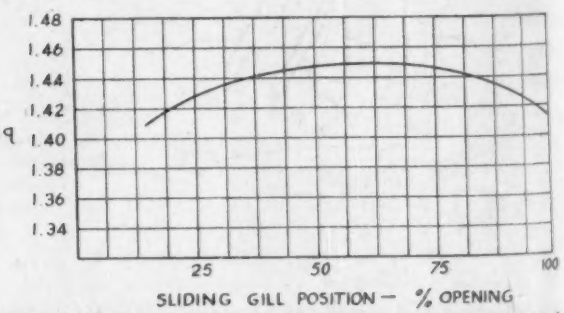
Contrary to popular opinion of the past few years, the aircraft-engine cooling fan is *not* an unfortunate cooling augmentation device which penalizes airplane performance. It is an *aerodynamic asset* which in many important flight regimes offers large increases in airplane performance. By virtue of its recent availability, careful study of its advantages in specific airplane applications is in order.

■ Acknowledgments

Important ideas on structural design of many of the fans described were provided by Roland and Allan Chilton of the Wright Aeronautical Corp. We believe that J. J. Jerger, of the Curtiss-Wright Airplane Division, St. Louis, was the first powerplant engineer to appreciate the importance of the Schicht and Wattendorf principles and to initiate their application to an aircraft cooling fan project.

Much of the reported recent progress in full-scale fan application to airplanes resulted from the cooperative efforts of the Glenn L. Martin Co., through L. T. Miller, its chief powerplant engineer, which provided the test ship and crew and fabricated the aftervanes for the flying boat fan project; the Curtiss Propeller Division, through George A. Dean, which assumed overall responsibility for that project; and the Benson Mfg. Co. of Kansas City, which contributed important structural details and fabricated rotors for this and other airplane projects during the year. The PBM-3 project was made possible through the active support of the U. S. Navy, in particular Capt. S. B. Spangler and Lt. C. T. Ray.

W. M. S. Richards, on cooling research, and B. J. Mezger, on fan development, both of the author's staff at Wright, have provided invaluable help in evaluation, design, and consummation of the more recent fan projects.



■ Fig. 28 – Boost obtained in flight test with propeller-speed fan and sliding gill exit

Rational DESIGN of FASTENINGS

by E. S. JENKINS

formerly with

Airplane Division—Research Laboratory

Curtiss-Wright Corp.

IN the design of joints and fastenings, the structural engineer has adopted a strictly pragmatic philosophy. In joints which are loaded concentrically, he uses the rule that the load is evenly distributed between the fasteners; for those eccentrically loaded, rules of the same order of simplicity have been formulated. Despite the fact that their bases are largely empirical, these design methods have the sanction of almost universal usage. Whatever success they achieve is due largely to a knowledge of strength test data, which is interpreted according to the same rules as are used in design. Recently, it has become generally recognized that extensive extrapolation of the methods will often suggest erroneous conclusions, since they do not constitute true solutions to the structural problems involved in the design of joints.

The growth of a gradual awareness of the limitations of the commonly used rules has prompted many individuals to seek more satisfactory solutions to the problems. Their search is motivated by more than mere intellectual curiosity, for it should be possible to make much more useful interpretations of the vast store of past design and service experience than present knowledge allows. Better prediction of the effects of repeated loads, of loads exceeding the material yield point and of the joint stiffness should be possible.

Probably the best reason of all for the development of rational methods for joint design is the economic one. The extremely high cost of making joints is inadequately appreciated. For example, it is not ordinarily realized that approximately 50% of the total cost of a present-day all-metal airframe is made up of the costs of fastening together the many bits and pieces which constitute the structure. The following table shows a typical breakdown of that part of the total airframe cost which concerns the joining of structural elements:

Total Cost of Airframe, %	100
Cost of Riveting (design, fabrication and installation of riveted connections), %	40
Cost of Spot Welding, %	5
Cost of Bolting, %	3
Gas and Arc Welding, %	2
Special Fasteners, %	less than 1
Total Cost of Fastenings, %	50+

Furthermore, a large part of the structural design time expended on an airplane is concerned with fastening problems. The substitution of successful analytic methods for the empirical calculations now used would result in greater design efficiency. The previous figures indicate that even a modest reduction in the number of fastenings utilized would provide a substantial reduction in cost.

If the skeptical designer is still unconvinced of the necessity for carefully studying joint design, it need only be mentioned that a present-day, large-sized craft has about

THE design of airplane joints and fastenings has become a very important problem, for approximately 50% of the total cost of a modern all-metal air frame is made up of the costs of fastening together the many bits and pieces that constitute the structure.

When the metal airplane was developed, manufacturers resorted to new methods of fastening, such as rivets, spot welding, and various types of removable fasteners, in place of the bolts, screws, and glue of earlier days. Later, cementing became feasible even for metal fabrication, with the aid of the new resin adhesives.

Mr. Jenkins gives here an analysis of the structural behavior of both continuous connections, such as cemented joints, and discontinuous connections, such as riveted and spot-welded joints.

■

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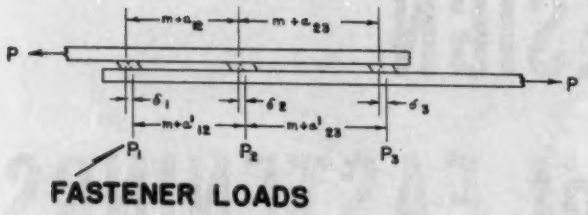
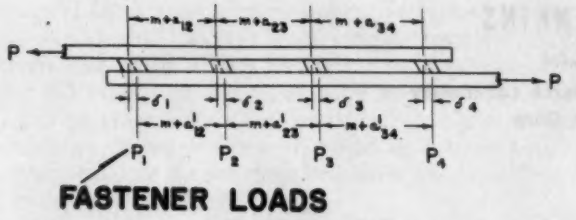
[This paper was presented at the SAE War Engineering-Annual Meeting, Detroit, Mich., Jan. 14, 1944.]



■ Fig. 1A – Three-row joint

■ Fig. 1B – Four-row joint

3,000,000 rivets distributed throughout its structure; this being only one of the many fastening methods which the complete craft must incorporate. An engineering item of such great importance certainly justifies an exhaustive design analysis.

Illustrative of the complexity of the problem are the results of a test program recently conducted in the laboratories of a leading aircraft manufacturer. The object was to determine the strength of a row of removable fasteners, placed parallel to the direction of the tensile load. (A removable fastener is one used in a joint which must be rapidly and repeatedly fastened and unfastened.) The somewhat disconcerting test data indicated that the use of an increased number of fasteners did not result in a corresponding increase in the strength of the connection. Stated in another way, it was found that the strength of the joint was not equal to the product of the number of fasteners present and the strength of an individual fastener.

These results contradict one of the most universally used rules of riveted joint design, namely, that each rivet of a noneccentric pattern carries an equal portion of the total load and that the strength of each rivet in the group can be obtained from tests of one or more rivets. While this rule was originated in connection with riveted joints only, its simplicity and clarity strongly suggested the possibility of its extension to cover other types of fasteners as well. Such an extension, in the case of the removable fasteners, is invalid. It is invalid for practically all fasteners since it oversimplifies the analysis of the joint. The engineer should realize that the design of fastened joints involves the study of a redundant structural system.

■ Early Methods of Fastening

In the earlier years of the aircraft industry, the problems of fastening were relatively simply decided. Wooden structures were bolted, screwed, or glued. With the inception of the metal airplane, riveting assumed a prominent part in the fabrication process. Spot welding, or any other form of welding, had not been sufficiently developed to allow its use to any great extent and the need for fasteners with specialized functions (removability, and so on) had not yet become pressing. In keeping with the limited knowledge of aerodynamics and material properties of the period, the design methods were sketchy and stressed simplicity rather than accuracy. Riveting, in the early years, was uncontested as the best permanent fastening for metal sheets. Bolting was accepted as the best available method for removable connections. Even in glues, so little was known about their chemistry that the use of casein glues, known to mankind for the past 3500 years, was almost universal. A powerful motivating force toward a careful survey of fastening techniques was absent during these formative years.

The impetus of the present war has changed this picture radically. Spot-welding performance has become comparable with that of rivets for all airplane structural applications; in addition, spot welding is found to offer economic advantages over riveting. The value of gas and arc welding has been proved beyond dispute. Cementing has gone beyond the field of wooden structures and has entered metal fabrication by the development of new resin adhesives; it is claimed by some that, with the aid of these adhesives, parts can be cemented together with a joint strength equal to that attainable with any other method. Special fastenings have been developed by the dozens for many different uses. Exemplary of the rushing progress of the field today is the latest proposed fastening method. Taking its cue from the common office stapler, metal sheets are stitched together, using high strength steel wire as the staple material. The stapled joint is extremely cheap to make and shows excellent static strength qualities and considerable strength in fatigue. Thus, the present-day engineer finds himself confronted with a host of fastening methods, each possessed with its own relative advantages and disadvantages. For each particular job, he must choose the most acceptable fastener. The choice must take into account both the economic and the engineering aspects of the application and must usually strike a proper compromise between the two. In addition to the structural requirements, due consideration must be given to the attainment of fabrication simplicity and, in many cases, to special factors such as aerodynamic smoothness.

It is not the purpose of this paper to give the ten commandments which govern the choice of the best fastener for a given job. In truth, no such rules or generalizations exist. For example, without fear of controversy and within the strict confines of logic, it could be stated that nonstructural parts should be fabricated by the most economical method. Unfortunately, no part of an airplane structure is completely nonstructural and hence the incontroversial nature of the "rule" vanishes. The choice of removable fasteners for access doors and hatches is another illustration of the complexity of the problem. Since practically all removable-type fasteners, save for screwed attachments, carry the load through a spring, their structural effectiveness is open to considerable question. It is customary, when using this type of connection, to provide means for diverting either all or part of the load from the cut-out region. Clearly, no rote rules could possibly specify the optimum amount and nature of the load diversion in every structure. Only the application of basic fastener principles, together with some facility in tracing the load flow in structures and coupled with experience and common sense, will allow the competent solution of such

problems.

With this in mind, a series of projects have been included in the present structural program of the Curtiss-Wright Research Laboratory, whose purpose is to attempt a comprehensive rational investigation of the entire field of aircraft fastenings. Studies of both specific fasteners and general problems are being conducted and due attention is being given to both experimental and analytical methods of attack. The results to date of two of these projects are given in this paper. It is hoped that the ideas presented will stimulate interest in fundamental fastener research and will also lead to a free exchange of information among the interested parties.

Structural Analysis of Fastenings

For ease of mathematical manipulation, an analysis of the structural behavior of fastenings is best divided into at least two parts. The first is a study of continuous connections, such as cemented joints; the second deals with discontinuous connections, as exemplified by multirivet and multispot-weld joints. Continuous connections utilize the entire contact area between the two parts for bonding, as compared with the partial bonding which exists in discontinuous connections. The two types of joints are naturally interdependent to some extent; for example, the action of a discontinuous joint with a large number of fasteners (in the direction of the joint length) is similar to that of a continuous joint. In most practical cases, however, the analogy is too obscure to eliminate the need for separate solutions.

Considering discontinuous connections, a discussion concerned specifically with riveted joints gives some initial insight to the problem. Thus, an understanding of the behavior of rivets under load leads to an appreciation of the important parameters in discontinuous joint analyses. It is of particular interest to discuss the load distribution in multirivet joints and to explain how load equalization on all rivets is brought about at high joint loads.

It is readily recognized that large, compressive bearing stresses exist in the frontal region of a loaded rivet. The effect of these high stresses is to cause early yielding of the sheet material surrounding the rivet, with the consequent distortion of the circular rivet hole to a somewhat elliptic shape. If the deformation across a single rivet of one of the joined sheets relative to the other is measured (such a deformation is shown as δ_1 in Figs. 1A and 1B), the sheet yielding around the rivet is evidenced by a reduction in the slope of the load-relative deflection curve; given load increments are accompanied by deformations of increasing magnitude as larger portions of the sheet become plastic. Expressed in another way, the rivet becomes less sensitive to deflection at high loads; its "stiffness" decreases.

Consider now two rivets, in line with the load and unequally stressed. Since the more highly loaded rivet has the least stiffness, it will pick up load at the slowest rate. The greater stiffness of the less highly loaded rivet causes it to pick up load more rapidly. This inequality of load gain will clearly persist until a joint load is reached at which both rivets achieve equal loading, or at which premature fracture occurs. The facility with which the load redistribution takes place is found to be dependent on the slope of the load-relative deflection curve for the fastener and on the stiffness of the sheets between fasteners (defined as Ewt/l, where E = Young's modulus of sheet material, w = transverse fastener spacing, t = sheet thickness, and l = longitudinal fastener spacing in direction of load). A large sheet stiffness and a marked yielding tendency of the fasteners leads to the most rapid equalization of load. These conclusions are based on the development given in Appendix I, where the problem of the load distribution in a multifastener joint is treated in detail.

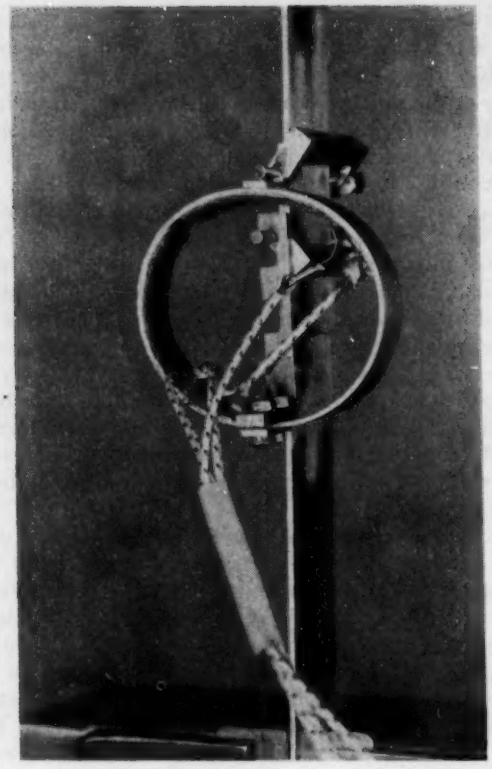
Since rivets do exhibit marked yielding tendencies at high load, each rivet will carry an approximately equal share of the joint load at fracture. Initial imperfections in the joint, such as an improperly upset rivet, will affect the load distribution only during the low load range; as soon as the initial clearance is taken up by plastic slippage of the remaining rivets, the unloaded rivet will rapidly pick up its share of the load. The fracture load distribution will be similarly unaffected by elastic stress concentrations introduced purely by virtue of the rivet pattern employed; they will be damped out in the high load range by the ductility of the fasteners.

The large bearing stresses, which account for the ductility of the rivets, are not usually present when other types of fasteners are used. Hence, there is no reason to believe that the loads in a multispot-weld joint, for example, are necessarily evenly distributed at fracture. To enable the study of this and allied questions, the method of load analysis given in Appendix I was developed and experimentally checked. The method is based on several easily visualized statements concerning the equalities of certain combinations of the extensions in the sheets between fasteners, and the relative deformations across the fasteners. Since the method is based on equalities of deformations, which are directly measurable quantities, the method is well adapted to experimental check.

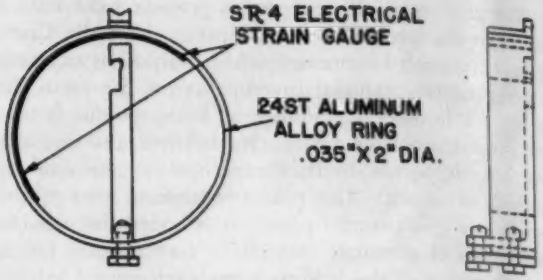
The object of the method is to enable rapid and accurate estimation of the load distributions at the yield and fracture of a multifastener joint. Accordingly, emphasis has been placed on the development of a simple rule which can be readily used in practical design. The present method satisfies this requirement. Furthermore, it presumes the knowledge of no empirical data save the load-relative deflection curve, obtained by test, for an individual fastener of the type under consideration.

Methods similar to those outlined in Appendix I have been suggested in the past by numerous investigators. One of the major difficulties encountered in their application was the accurate determination of the load-relative deformation data for the fasteners, especially in the plastic range just preceding fracture. The instrumentation problems are great; extremely small deflections must be accurately measured, using gages which are preferably not destroyed by fracture of the specimen. The possibility of using mechanical gages of the Moore, Tuckerman, or Huggenberger types is at once eliminated. In fact, no existing gage of requisite accuracy and design, capable of withstanding fracture, was known to us at the start of the test program.

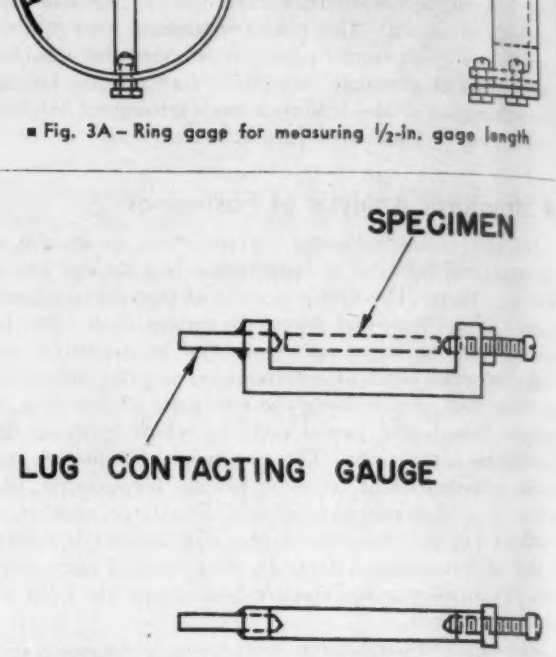
Consequently, a new type of gage was constructed. A brief description of it is of interest here. The gages are of the ring type and can be suitably modified to record overall deformations of the joint, deformations of the sheet between fasteners or load-relative deflection data across individual fasteners. Fig. 2 shows one of the gages mounted on a spot-weld specimen, arranged to record the sheet extension between two spot welds. Fig. 3A is a drawing of the gage, illustrating the details of its construction. To



■ Fig. 2 – Ring gage mounted on specimen



■ Fig. 3A – Ring gage for measuring ½-in. gage length



■ Fig. 3B – Steel clamp with tempered points (two clamps required)

use the gage, two lugs are first clamped to the specimen at the gage points, as shown in Fig. 2. The lug details are sketched in Fig. 3B. The ring gage is then positioned between the two lugs in a state of initial compression. As the specimen is loaded, the lugs move apart and relieve the compression of the ring. This in turn activates two electric resistance type strain gages cemented to the ring surface (Fig. 3A). The strain-gage indications are read with standard equipment and are interpretable directly in terms of specimen deformations. Since the outer ends of the gage grooves, in which the lugs are positioned, are built with large clearance, fracture of the specimen in a controlled-strain type testing machine does not harm the gages in any way. It is to be noted that the gages are positive acting throughout their range. The multiplying error of the gage and lug system used in the tests about to be described has been estimated as approximately $\pm 5\%$. A large portion of this error, it is to be noted, is due to the lug attachments; the gage alone possesses far superior accuracy.

Fig. 4 shows the load-relative deflection curves obtained across individual spot welds located in two- and three-row joints. Each curve is the average of many tests; the agreement between the curves for different samples is within the limits of the gage accuracy. (The excellent agreement was attained by having all of the samples welded on a single machine and by ensuring that the same machine setting was used at all times. All samples were made from the same lot of Alclad 24S-T sheet). It is to be noted that the load-deflection curves for all three spot welds are very nearly identical. The early bending over of the curve for the outside spot welds of the three-row joint is explained by the excessive yielding of the entire sheet outside the joint as the failure load is approached. This same condition does not occur in the two-row joint until much higher spot loads are reached. The agreement of the earlier parts of the curves fairly conclusively shows that the behavior of an individual spot weld is not dependent on the joint pattern, although premature yielding of the outside spots may sometimes be expected. Proof of this fact is essential if the methods of analysis employed in Appendix I are to be used successfully.

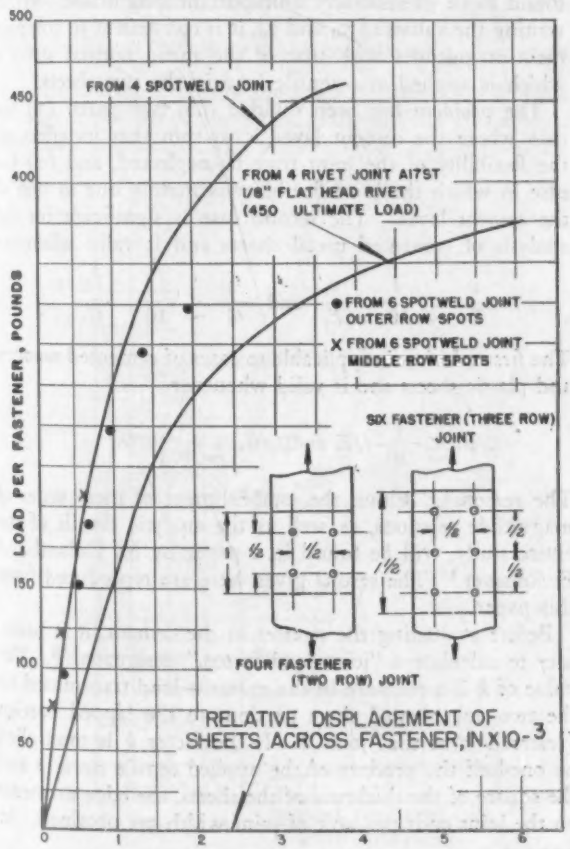
Also shown on Fig. 4 is a load-deflection curve for a rivet with an ultimate strength equal to those of the spot welds used in this study. The very much decreased stiffness of the riveted connections compared with the spot weld is apparent. Deformation data for the rivet were taken to the ultimate load, although it is not shown on Fig. 4 because of the large magnitude of the range of abscissa (deformation) values which would be required.

It should also be noted that data for one-row joints are not given. The excessive joint bending which occurs when one-row connections are loaded tends to make the action of this joint markedly different from those of multirow joints. Accordingly, the load-relative deflection data obtained for the spot welds in two-row joints are taken as representative of the action of all individual welds in multirow joints; the one-row data are not acceptable for this purpose.

Fig. 5 shows the agreement between the calculated and measured load distributions in spot-welded three-row joints. The calculated distributions are based on the load-deflection curve for an individual spot obtained from two-row

joints and on the methods of Appendix I; the manner of computing the experimental points is also outlined in the Appendix. The agreement between the two is excellent, save in the load range just preceding fracture where the measured values show a greater redistribution of load (toward uniformity) than do the calculated. At a joint load of 45% of the ultimate load, for example, the measured and calculated load distributions are in almost perfect agreement and show that (a) *the spots of the outer row are 71% more highly loaded than are those in the middle row,* (b) *the spots of the outer row are loaded 16% more highly than they would be if the load was uniformly distributed,* and (c) *the spots of the middle row are loaded 32% less highly than they would be if the load was uniformly distributed.*

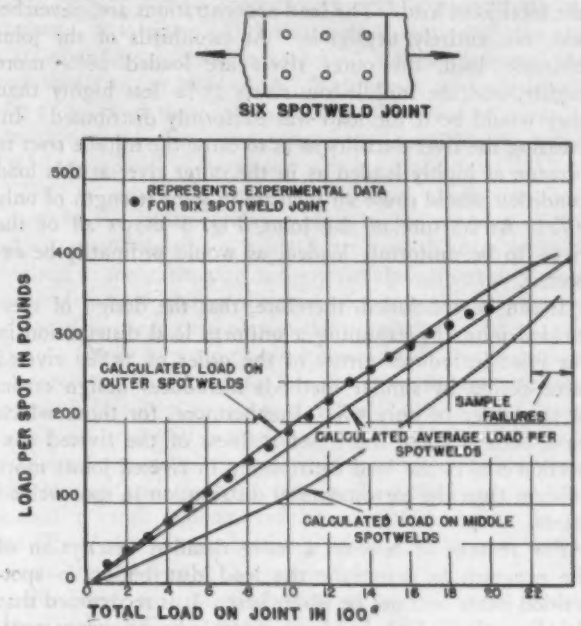
At a joint load equal to two-thirds of the ultimate, corresponding to the arbitrarily defined "yield" load in aircraft structural analysis, the inequality of the loads carried by the spot welds is equally apparent. The outer spots are 53% more highly loaded than are the middle spots; the spots of the outer row are loaded 13% more highly, and the middle spots 26% less highly than they would be if the load was uniformly distributed among the fasteners. *If the fastener ductilities were increased so as to cause the middle-row spot to be as highly loaded as is the outer spot at this load condition, the joint strength would be increased by 14%.* This last figure is of extreme importance, since it represents the optimum load advantage which can be gained by producing spot welds of greater ductility than those employed in this program.



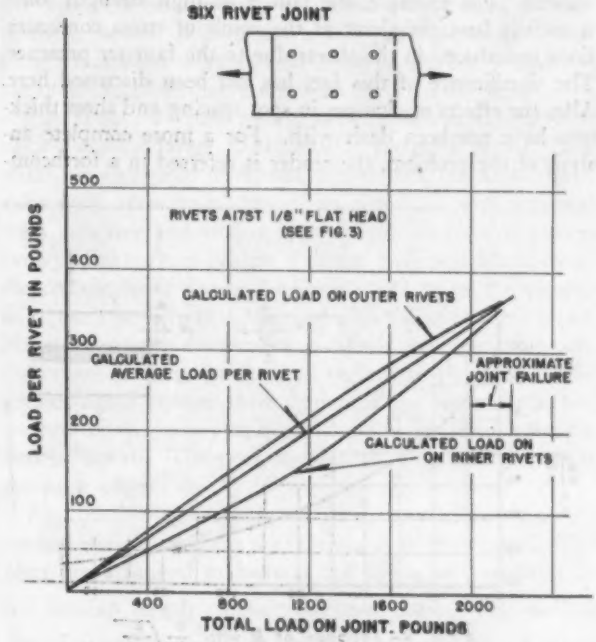
■ Fig. 4 – Load-relative displacement curves for four- and six-spot-weld joints

At the fracture load, both the experimental and measured values indicate that the load distribution among the spots is practically uniform. It is to be noted, however, that this is a relatively unimportant conclusion; the joint design is based on a joint load which is a fraction of the ultimate load. At these lower loads, the effects of the load concentration in the outer spots cannot be ignored.

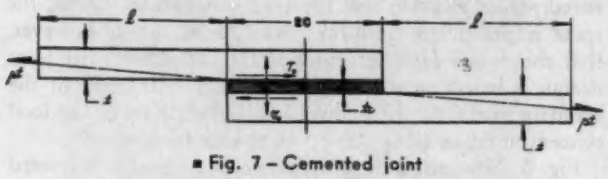
Fig. 6 shows the load distribution in a three-row riveted joint, based on the load-deflection curve for a rivet as shown on Fig. 4. It is clear that the decreased stiffness of the rivet, compared with that of the spot weld, causes the



■ Fig. 5 – Calculated and experimental load distribution in six-spot-weld joint



■ Fig. 6 – Calculated load distribution in six-rivet joint

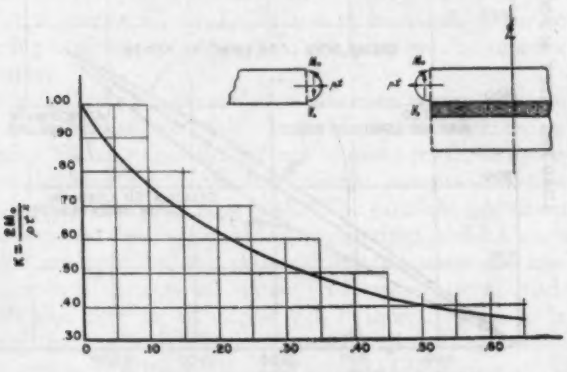


■ Fig. 7 – Cemented joint

load in the joint to be distributed more nearly uniformly than in comparable spot-welded joints at all values of the total joint load. The load concentrations are, nevertheless, not entirely negligible. At two-thirds of the joint ultimate load, the outer rivets are loaded 28% more highly, and the middle-row rivets 15% less highly than they would be if the load was uniformly distributed. Increasing the rivet ductility so as to cause the middle rivet to become as highly loaded as in the outer rivet at this load condition would cause an increase in joint strength of only 7%. At fracture of the joint, Fig. 6 shows all of the rivets to be uniformly loaded, as would ordinarily be expected.

It can be concluded, therefore, that the design of spot-welded joints by assuming a uniform load distribution in the joint introduces errors of the order of 15%; riveted joint design by similar methods introduces design errors of the order of only 5%. Furthermore, for those which have been studied, the lesser stiffness of the riveted connection makes the load distribution in riveted joints more efficient than the corresponding distribution in spot-welded joints.

For reasons of brevity, a more detailed description of the program to determine the load distribution in spot-welded joints will not be given here. It is recognized that the discussion which has been given is by no means complete. Greater attention should be given to an analysis of the accuracy of the gages and the data. Also, the theoretical aspects of the problem should be more carefully reviewed. For example, the failure of high strength joints is usually brought about as the result of stress concentrations introduced in the sheets due to the fastener presence. The significance of this fact has not been discussed here. Also, the effects of changes in spot spacing and sheet thickness have not been dealt with. For a more complete analysis of the problem, the reader is referred to a forthcoming paper which will describe the investigation in greater detail.



■ Fig. 8 – Variation of k with $\frac{c}{t}\sqrt{\frac{p}{E}}$

■ Analysis of Cemented Joints

For the continuous, cemented joint, we have found it possible to determine analytically the elastic stress distribution in the cement layer with great precision. The mathematical methods utilized in the solution, in keeping with the difficulty of the problem, are extremely tedious and complex. Nevertheless, a number of interesting conclusions are drawn from the results and are significant with regard to both continuous and discontinuous joints.

The problem is formulated as shown in Fig. 7. Two sheets of thickness t are cemented together with an overlap length $2c$. The joint is considered to be wide in the direction perpendicular to the plan view of Fig. 7 and the sheets are assumed to extend a distance l to either side of the lapped region, the magnitude of l being at least several times the lap length $2c$. The physical properties of the sheet material are notated E, G, and ν, representing Young's modulus, shear modulus, and Poisson's ratio, respectively. Both the sheet and the cement are assumed to be elastic. The cement layer thickness is called t_c and the physical properties of the cement are E_c and G_c, representing the Young's and shear moduli, respectively.

The stresses in the cement which are found to be of importance are indicated schematically on Fig. 7. They are the shearing stress τ_0 in the plane of the layer, and the normal stress σ_0 in a direction normal to this plane. The normal stress in the cement, parallel to the cement plane, is found to be of relatively unimportant magnitude. When writing the values of τ_0 and σ_0, it is convenient to compare their magnitudes with that of the mean normal stress p which is applied as a tensile load in the two sheets.

The problem has been divided into two parts: (1) the case where the cement layer is so thin that its effect on the flexibility of the joint may be neglected, and (2) the case in which the joint flexibility is mainly due to that of the cement layer. The second case is significant for the analysis of cemented metal sheets and is valid whenever

$$\frac{t}{E} \lesssim \frac{l}{10}\,\frac{t_c}{E_c} \quad \text{and} \quad \frac{t}{G} \lesssim \frac{l}{10}\,\frac{t_c}{G_c}$$

The first solution is applicable to cases of cemented wooden and plastic sheets and is valid whenever

$$t_c/E_c \lesssim \frac{l}{10}\,t/E \quad \text{and} \quad t_c/G_c \lesssim \frac{l}{10}\,t/G.$$

The reasoning behind the establishment of these order of magnitude relations, as well as the analytic details of the entire study, will be found in a paper by M. Goland and E. Reissner.[1] The results given here are reproduced from this paper.

Before evaluating the stresses in the cement, it is necessary to calculate a "joint load factor," designated k. The value of k is a measure of the moment load transmitted by the two uncemented sheet portions to the lapped portion (referred to as the "joint"). If the factor k is multiplied by one-half the product of the applied tensile stress p and the square of the thickness of the sheets, the edge moments on the joint ends per unit of joint width are obtained. In

[1] See Transactions of the ASME – Journal of Applied Mechanics, Vol. 11, March, 1944, pp. A-17 - A-27: "The Stresses in Cemented Joints," by M. Goland and E. Reissner.

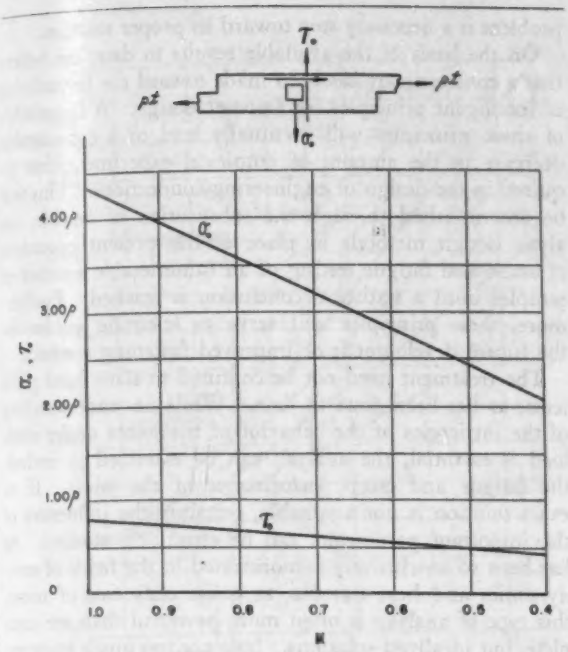


Fig. 9 – Maximum stress concentrations in an inflexible cement layer, plotted as a function of k

addition to these moments, it is readily seen that the applied tensile load will be carried through the sheets to the joint edges without change of magnitude. The value of k is found to be dependent on the parameter $\frac{c}{t}\sqrt{\frac{p}{E}}$, as is shown in Fig. 8. Here $\frac{c}{t}$ represents the ratio of one-half the joint overlap length to the sheet thickness and E is the Young's modulus of the sheet material. As the applied joint load p increases, the magnitude of k is seen to decrease. The result of this is to cause the joint edge moment to increase with load, but at a reduced rate.

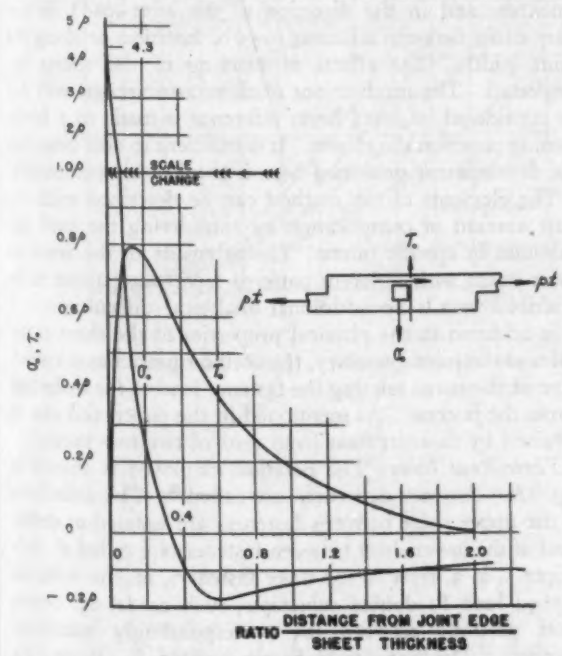


Fig. 10 – Stress distribution along shear plane of inflexible layer joint in vicinity of edges, calculated for $k = 1.0$

Fig. 9 shows the maximum stress concentrations in an inflexible cement layer, plotted as a function of the factor k. The maximum cement stresses are surprisingly high, the σ_0 stress being from two to four times as large as p and the shear stress τ_0 varying from $0.45p$ to $0.79p$. The concentrations of stress are reduced as the joint load increases, that is, as k decreases, but the reduction of k is offset by the increase of p and is not sufficient to prevent the steady rise of the numerical magnitudes of the maximum stresses.

It is of extreme interest to note the existence of such large cement stresses, particularly with regard to the σ_0 stresses. It is observed, during actual joint tests, that the start of failure is characterized by a splitting apart of the two sheets at the joint edges. This action is explained by the presence of the high σ_0 stresses. For easy reference, a descriptive title for the σ_0 stresses is desirable; the name "tearing stresses" is suggested as being appropriate.

Fig. 10 shows the stress distribution along the shear plane of the inflexible layer joint in the vicinity of the edges, calculated for a value $k = 1.0$. The σ_0 stress is highest at the joint edge and rapidly damps out to a small value ($0.1p$) within a distance along the shear plane of about two sheet thicknesses; the reversal of the σ_0 stress from a tensile to a compressive stress within this distance is of interest. The τ_0 stress has the value zero at the joint edges; it rapidly rises to a peak value and once more decays to almost zero within a distance of two sheet thicknesses. In joints with relatively inflexible cement layers, therefore, the tearing and shear stresses in the cement are concentrated in small regions near the ends of the joint. In the middle of the joint, a condition of practically pure tension exists, the stress in each sheet being $0.5p$.

For joints with relatively flexible cement layers, results of entirely different nature are obtained. To begin with, whereas the stress distribution in the cement for the inflexible layer case is independent of both the joint dimensions and the physical properties of the cement material, the cement stresses in flexible layers are dependent on both of these factors. Furthermore, the cement stresses are no longer restricted to small end zones in flexible layers, but are of appreciable magnitude over all the shear plane.

Fig. 11 shows typical σ_0 and τ_0 distributions in joint with flexible cement layers, calculated for several k values. The joints are composed of representative materials and dimensions (say, aluminum-alloy sheets cemented with a typical resin adhesive and with a joint overlap to joint thickness ratio of five). It is evident that the increased flexibility of the cement layer causes the shear stress τ_0 in the cement to be more uniformly distributed over the joint shear plane. Nevertheless, the distribution still indicates a sizeable concentration of shear in the end regions of the joint. The stresses again reverse their sign near the joint edges, but are not damped away as quickly as they were in inflexible cement layers. The peak σ_0 and τ_0 values still persist at the joint edges.

Figs. 12 and 13 are plots of the maximum values of the tearing stress σ_0 and the shear stress τ_0 in the cement. The plots are arranged to include the effects of variations in the semilap length to sheet thickness ratio $\frac{c}{t}$, as well as the effects of variations in the sheet and cement materials. The definitions of β and γ, which appear on the figures are

$$\beta = \sqrt{8\,\frac{E_a}{E}\,\frac{t}{t_c}} \qquad \gamma = \sqrt[4]{6\,\frac{G_c}{E}\,\frac{t}{t_c}}$$

It is at once clear that the magnitudes of the peak cement stresses are very much lower in joints with flexible cement layers than in joints with relatively inflexible layers. Their orders of magnitudes for representative joints are indicated on Fig. 11. For $\frac{c}{l} = 5$, the maximum σ_0 value is about $0.12p$ and that of τ_0 is approximately the same magnitude. It is to be noted that even more extreme reductions in the σ_0 values than in the τ_0 values are brought about by increased cement layer flexibilities.

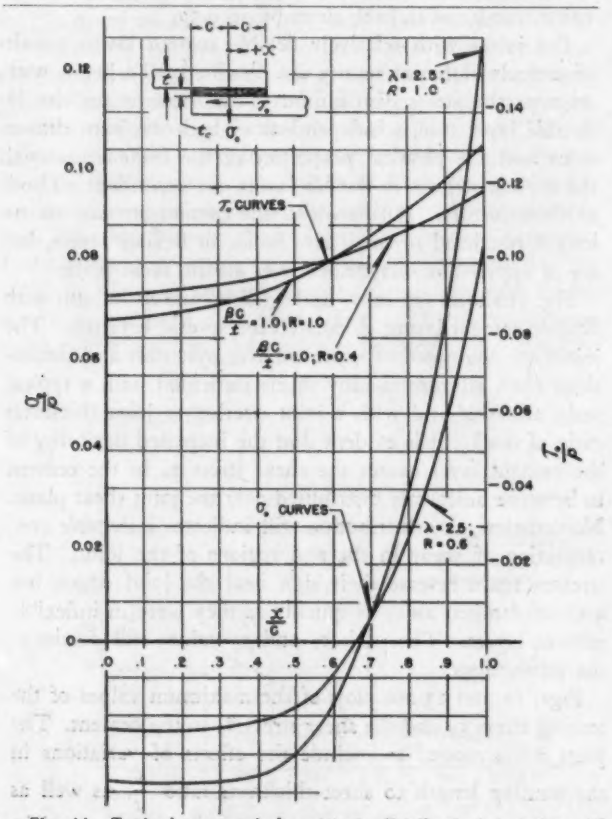
The flexible cement layer analysis is applicable to joints composed of thin metal sheets cemented together. The adhesive materials used in such joints are particularly weak in tension. Hence, the σ_0 stresses, although smaller than those encountered in inflexible layer joints are still of sufficient magnitude to cause splitting failure at joint edges.

■ Conclusion

The continuous joint results have been given in detail not only because of their interest with regard to cemented joints, but also because they provide a qualitative insight to the behavior of discontinuous joints. The significance of the presence of the large tearing stresses cannot be underestimated. They are undoubtedly at least partially responsible for the failure of the edge fasteners in spot-welded and other similar connections. They are also of tremendous importance in determining the fatigue characteristics of the connections. It is clear from the results that a careful consideration of the tearing stresses in every problem is a necessary step toward its proper solution.

On the basis of the available results to date, we believe that a concrete start has been made toward the formulation of intelligent principles for fastener design. A knowledge of these principles will eventually lead to a considerable decrease in the amount of empirical experimentation required in the design of engineering connections. This will be accomplished through the substitution of flexible, analytic design methods in place of the present procedure of static and fatigue testing of an innumerable number of samples until a statistical conclusion is reached. Furthermore, these principles will serve as scientific guides for the future development of improved fastening methods.

The treatment need not be confined to static load problems, as has been done to date. While an understanding of the intricacies of the behavior of the joints under static load is essential, the analysis can be extended to include the fatigue and creep performance of the joints. If an exact solution is not attainable, certainly the influences of the important parameters can be effectively studied. As has been so conclusively demonstrated in the fields of aerodynamics and heat transfer, to quote only two of many, this type of analysis is often more powerful than are complete, but idealized solutions. It is not too much to expect that the future will bring tremendous advances in the art of fastening together structural parts, accompanied by a keener interest on the part of all engineers to understand and appreciate the intricacies of fastening design.



■ Fig. 11 – Typical tear and shear stress distributions in joints with flexible cement layers

Appendix I

Calculation of Load Distribution in Multifastener Joints

The method is based on simplified concepts of joint action and is essentially geometrical in character. Equalities are written between the relative deflections of the midplanes of the two sheets and the deflections across the fasteners. It is assumed during the course of the development, that the longitudinal sheet strains (averaged across the sheet thickness and in the direction of the joint load) do not vary either between adjacent rows of fasteners or along the joint width. The effects of bending of the sheets are neglected. The implications of these assumptions will not be considered in detail here; reference is made to a forthcoming paper on the subject. It is sufficient to note now that the development presented here is checked experimentally.

The elements of the method can be described with the least amount of complication by considering the load distribution in specific joints. The extension of the work to cover joints with different patterns from those about to be described entails no additional analytical difficulties.

In addition to the physical properties of the sheet material and the joint geometry, the method presumes a knowledge of the curve relating the fastener load to the deflection across the fastener. As mentioned in the paper, this can be obtained by measurement from tests of two-row joints.

Three-Row Joint – The notation employed is shown in Fig. 1A. Fastener deflections are called δ. The deflections of the upper sheet between fasteners are notated a; deflections of the lower sheet between fasteners are called a'. Subscripts 1, 2, 3, refer to the three fasteners, in line with the applied load P; double subscripts, such as 12, 23, denote sheet portions between the correspondingly numbered fasteners. The fastener loads are notated P_1, P_2 and P_3. The uniform fastener spacing is m.

Then, from conditions of symmetry

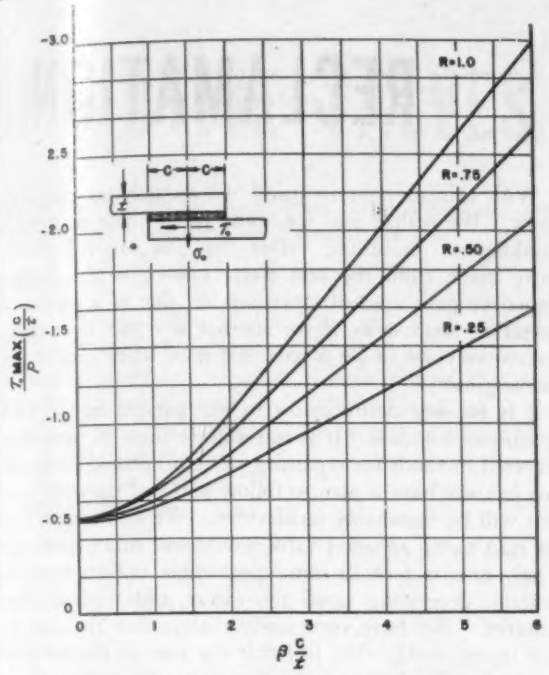


Fig. 12 – Maximum values of shear stress in the cement

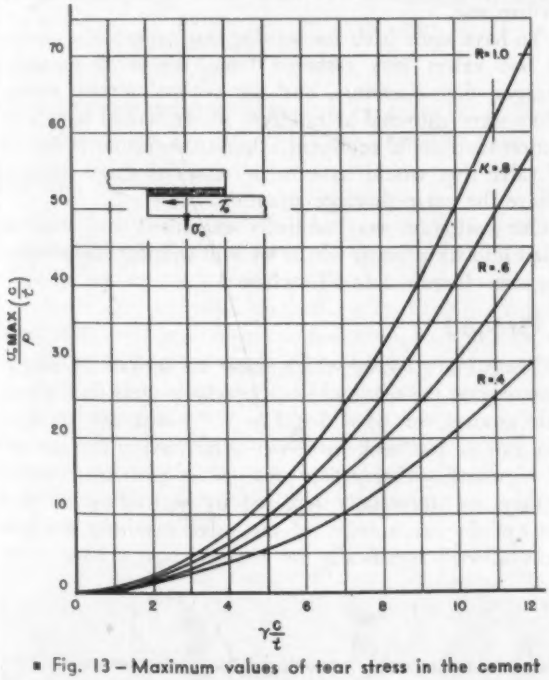


Fig. 13 – Maximum values of tear stress in the cement

$$\delta_1 = \delta_3$$

$$a_{12} = a'_{23} \qquad a_{23} = a'_{12} \tag{1}$$

Taking into account the simplifying assumptions already mentioned, then

$$a_{12} = \frac{(P - P_1)m}{wtE} \tag{2}$$

Where P is the applied load associated with each line of fasteners, t is the sheet thickness, w is the fastener spacing in the direction of the joint width, and E is the Young's modulus of the sheet material. Also

$$a_{23} = \frac{P_1 m}{wtE} \tag{3}$$

From the geometry of Fig. 1A, it is evident that

$$\delta_1 + a'_{12} = \delta_2 + a_{12} \tag{4}$$

or

$$\delta_1 - \delta_2 = \frac{(P - 2P_1)m}{wtE} \tag{5}$$

The conditions for static equilibrium require that

$$P = 2P_1 + P_2 \tag{6}$$

and so equation (5) can be rewritten

$$\delta_1 - \delta_2 = \frac{P_2 m}{wtE} \tag{7}$$

Since the relationship between the fastener load and the fastener deflection is assumed to be known, equations (6) and (7) are two simultaneous equations involving the two unknowns P_1 and P_2.

A direct method for calculating the load distribution in the joint for all values of the total load P is as follows: Choose a value for P_2 and find the corresponding value of δ_2 from the load-relative deflection curve for the fastener. From equation (7) calculate δ_1 and then find P_1. From equation (6) compute the total load P. Repetition of this procedure for several values of P_2 will allow the construction of curves relating the magnitudes of the fastener loads P_1 and P_2 to the total joint load P.

The experimental check of the method requires that P_1 be calculated directly from the data measured on three-row joints. This is done by means of the relation

$$P_1 = \frac{a_{23}}{a_{12} + a_{23}} P \tag{8}$$

Four-Row Joint – The notation used is similar to that defined for the three-row joint and is shown schematically in Fig. 1B. The analysis of the four-row joint is similar in outline to the previous solution.

By symmetry

$$\delta_1 = \delta_4 \qquad \delta_2 = \delta_3$$

$$a_{12} = a'_{34} \qquad a_{34} = a'_{12} \tag{9}$$

$$a_{23} = a'_{23}$$

and approximately

$$a_{12} = \frac{(P - P_1)m}{wtE} \tag{10}$$

$$a_{23} = \frac{(P - P_1 - P_2)m}{wtE} \tag{11}$$

$$a_{34} = \frac{P_1 m}{wtE} \tag{12}$$

From Fig. 1B

$$\delta_1 + a'_{12} = \delta_2 + a_{12} \tag{13}$$

or

$$\delta_1 - \delta_2 = \frac{(P - 2P_1)m}{wtE} \tag{14}$$

From conditions for static equilibrium,

$$P = 2(P_1 + P_2) \tag{15}$$

so that equation (14) becomes

$$a_1 - a_2 = \frac{2P_2 m}{wtE} \tag{16}$$

Since the load-relative deflection curve for an individual fastener is presumed to be known, equations (15) and (16) form two simultaneous equations in the two unknowns P_1 and P_2. A calculation procedure similar to that for the three-row joint can be used to facilitate their solution.

RECLAMATION O UT

The information presented in this paper was gathered through the combined efforts of the Engine Valve Group of the Reclamation of Parts Subcommittee of the SAE-Ordnance Vehicle Maintenance Committee. Members of the Engine Valve Group are:

Norman Hoertz, chairman, Thompson Products, Inc.; A. J. Brezina, Thompson Products, Inc.; R. V. Cox, Aluminum Industries, Inc.; E. P. Gohn, Atlantic Refining Co.; J. C. Hale, Cleveland, Columbus & Cincinnati Highway, Inc.; F. J. Kelly, Wilcox-Rich Division, Eaton Mfg. Co.; L. V. LaRou, Wall-Colmonoy Corp.; A. W. Lindblad, Metallizing Co. of America; C. J. Long, Wall-Colmonoy Corp.; H. Osborn, Red Star Transit Co.; B. Park, Central Greyhound Lines, Inc.; W. G. Piwonka, Cleveland Transit System; H. Potter, Sealed Power Corp.; J. R. Spence, Stoody Co.; A. Townhill, Toledo Steel Products Co.; W. A. Wissler, Union Carbide & Carbon Research Laboratories, Inc.*

IT is our purpose in presenting this paper to give a brief report on the work done thus far by the SAE-Ordnance Vehicle Maintenance Committee working on the reclamation of automotive valves.

We will discuss briefly all ways and means of repairing valves. Much of our test work is still in process so we cannot give definite conclusions in all cases. However, we feel that our work will provide a few new ideas that may aid many fleet operators who are in distress due to lack of valves.

Upon starting our committee work, we discussed the idea of repairing valves with many garage men, fleet operators, and valve manufacturers. The number of answers were as numerous as the number of men contacted. The consensus of opinion of both the garage man and the fleet operator is that they prefer to limit their valve reclaiming to regrinding the seat and stem tip. Pre-war practice had been to repair only the seat and stem tip. Valves found with cracked heads, deep pits, burned areas, or seats ground to a knife edge on the circumference were always thrown away with no thought of reclaiming them. This was the logical thing to do since in normal times a new replacement valve was readily obtainable at a cost much lower than any form of repaired part.

Today, however, the valve manufacturing industry, like many other industries which require high-alloy steels, has felt the pinch of material shortages and alloy use limitations.

In many cases the lack of replacement valves has forced fleet operators to use their own ingenuity to repair valves by hard-facing, welding, or other methods, in order to keep their equipment rolling. These emergency repairs have developed to a point where it was felt that the good points of all repair methods, if properly gathered, might serve as a valuable contribution to our transportation system.

[This paper was presented at the SAE War Engineering-Annual Meeting, Detroit, Mich., Jan. 11, 1944.]

* Acknowledgment is also made to the following for their contributions: L. G. Baines, Transit Supply Co.; W. C. Bendel, Municipal Railway Co.; C. S. Berry, Aero Mayflower Transit Co.; W. W. Churchhill, Washington Motor Coach System; O. E. Harder, Battelle Memorial Institute; T. C. Howe, North Coast Transportation Co.; S. L. Hoyt, Battelle Memorial Institute; W. R. Hubka, Seattle Transit System; C. Hudson, Tennessee Valley Authority; W. W. Kunz, International Railway Co.; F. Patras, Southwestern Greyhound Lines, Inc.; E. Pekarek, Thompson Products, Inc.; Capt. G. C. Richardson, Office of the Chief of Ordnance – Detroit; C. K. Wilson, Metallizing Engineering Co., Inc.; D. K. Wilson, New York Power & Light Corp.; R. W. Ziifle, New Orleans Public Service, Inc.

With this thought in mind we started our committee work. We realize and we want you to bear in mind in thinking of reclaiming valves that the original pre-war valve made from the best steels known to the industry sometime gave unsatisfactory service, due to a number of operating factors; so, if we attempt a repair by welding, we are very apt to get a part that is of lesser quality than the original.

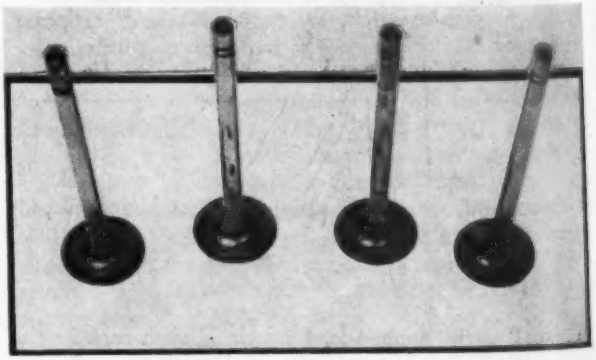
It is not our desire to try to put garages in the valve-reclaiming business. It is our sincere hope to present engineered methods for repairing valves so people attempting this job will have a plan to follow and end up with a part that will be reasonably satisfactory. We knew right from the start that a repaired valve would cost much more than a new one, in fact, in some cases three to four times the amount, depending upon the extent and type of repair required. We have completely disregarded the cost feature in our work. We feel that the user of the valve can best make the decision on valve repair by comparing the urgency with the cost. This applies to both military and civilian use.

To have some basis for starting our program, a quantity of 600 valves was gathered from the scrap barrels of garages, fleet operators, and car dealers' service stations. They were gathered at random, so we would have a collection that would represent a fair cross-section of the type of valve that would have to be repaired if we intend to relieve the valve shortage situation.

Our collection was carefully examined and was classified into six groups, which we will call, for identification purposes, Groups I to VI inclusive.

■ Group I

Group I are valves which show no serious burning or guttering on the seat and no appreciable wear on the stem, lock grooves, or tip. See Fig. 1. Generally speaking, they can be put back in service after having the seat and tip reground. This group presents no particular problem as they are universally repaired by regrinding the seats and tips on any number of the valve repairing machines manufactured specifically for that purpose. Most of the



■ Fig. 1 – Valves of Group I show no serious burning or guttering on the seat and no appreciable wear on the stem, lock grooves, or tip

UTOMOTIVE VALVES

by NORMAN HOERTZ
Service Engineer, Thompson Products, Inc.

machines have been on the market competitively for several years, and all of them do a satisfactory job.

Group II

Group II are valves which have the seats guttered, pounded, or burned to a point where refacing will bring the outer diameter to a knife edge or reduce the head diameter to a point where it will be too small to fit the seat. See Fig. 2. To repair them, metal of some kind must be added to the seat.

Group III

Group III are valves which have deeply burned areas, gutters, or cracks. See Fig. 3. To repair them, the cracks and burned areas must be welded.

Group IV

Group IV represents valves with stems worn. See Fig. 4. Here again is a case where metal must be added to make them usable.

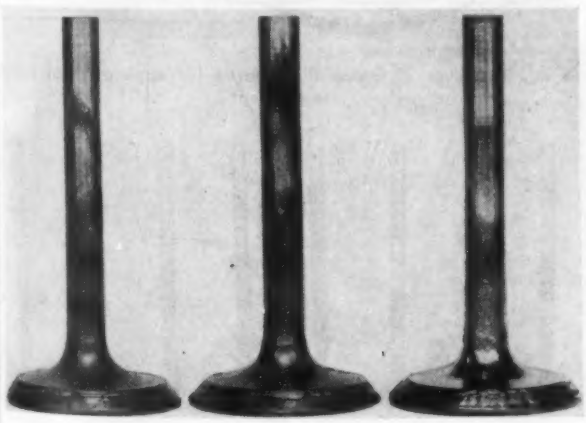
Group V

Group V represents valves needing repair to the retainer groove or tip. See Fig. 5. This is also a welder's job.

Group VI

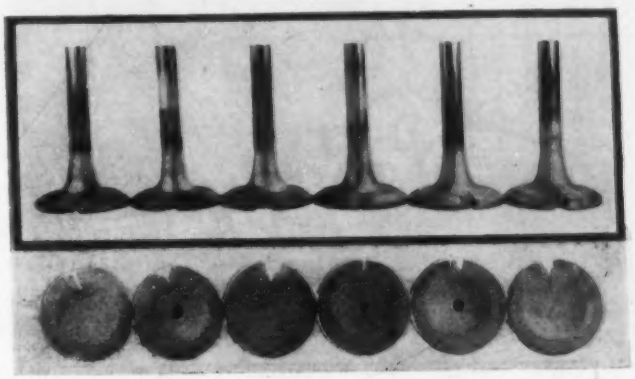
Group VI represents valves which are obviously damaged beyond repair. See Fig. 6. In actual practice valves will fall in one or more of the groups, but for convenience in our discussion, we will keep them isolated and determine the operations necessary to put each of them back in service.

Two operations common to all of the repairable groups are cleaning and inspection. Cleaning can best be accomplished by wire-brushing, grit-blasting or electrolytic cleaning. The electrolytic method of cleaning is very efficient and is probably the best method to use, particularly for operators who plan to do valve reclaiming on a large scale. See Fig. 7.

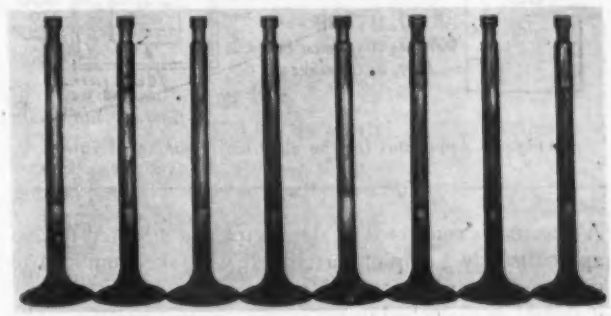


■ Fig. 2 – Valves of Group II have the seats badly guttered, pounded, or burned

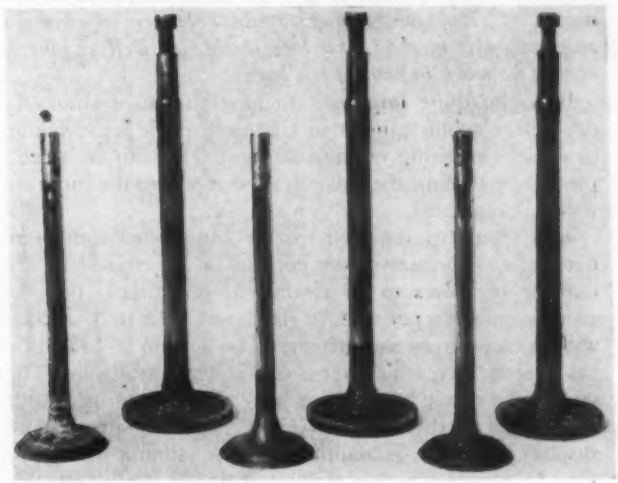
The electrolytic method consists of a small pot containing a solution composed of 40% $NA_2 CO_3$, technical anhydrous, and 60% NaOH flakes heated to a temperature of 800 to 900 F. The valve to be cleaned is immersed in the solution and connected to a 6- or 8-v source of d-c electric current. The valve is connected to the negative side of the current source and the pot to the positive side.



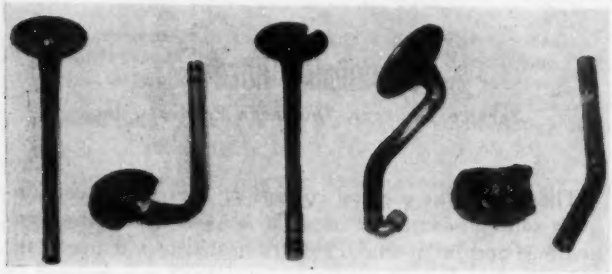
■ Fig. 3 – Valves of Group III have deeply burned areas, gutters, or cracks



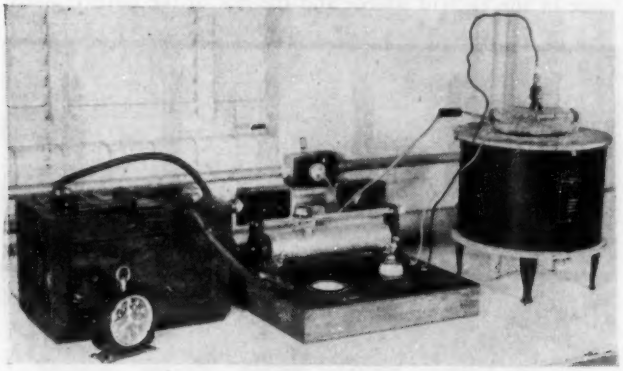
■ Fig. 4 – Valves of Group IV have worn stems

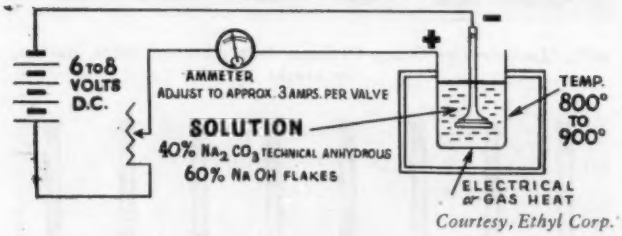


■ Fig. 5 – Valves of Group V need repair to the retainer groove or tip



■ Fig. 6 – Valves of Group VI are obviously damaged beyond repair





Courtesy, Ethyl Corp.

■ Fig. 7 – Apparatus for the electrical cleaning of valves

A rheostat is connected in the circuit and adjusted to pass approximately 3 amp of current. It will take from 5 to 10 min to clean an average valve. After the valve is cleaned, it should be quenched in water at room temperature. The quench is used to remove the coating of molten salt which would be very difficult to get rid of if the valve cooled normally. *Extreme caution should be taken in cleaning valves by this method, and face shields, as well as gloves, should be worn to avoid flesh burns.*

It is also quite important to have the valve absolutely dry before submerging it in the molten salt for cleaning, to avoid splattering of the solution. This can be accomplished by heating the valve to a point where the moisture will be evaporated.

The apparatus required can be constructed quite economically. The size of the pot can be determined by the number of valves to be cleaned at one time. For the average garage a pot 6 in. in diameter and 6 to 8 in. deep will be found to be satisfactory. The pot can be heated by gas or electricity, whichever is conveniently available. The source of d-c may be a storage battery or a motor-generator set. A pot of this size may clean 20 to 30 valves before a solution change is required. These figures will vary according to the size and amount of deposit on the valve.

After cleaning, the valves should be inspected. It is very important to inspect the neck of the valve for defects or fatigue cracks. Cracks are exposed best on the electrolytic cleaned or wire-brushed valve. Grit-blasting, although doing a good cleaning job, tends to cover the small cracks and, therefore, is not as suitable as the other methods.

There are several methods for the determination of cracks, these are: visual inspection by the naked eye, the use of magnifying binoculars, or equipment such as zyglo or magnaflux.

If naked eye inspection is used, it is best to submerge the valves in gasoline and then dust with chalk. The most dangerous cracks will become visible, although minute cracks may remain undetected.

The use of binoculars ranging from 7 to 14 power is fair for this work, but again minute cracks can be easily overlooked.

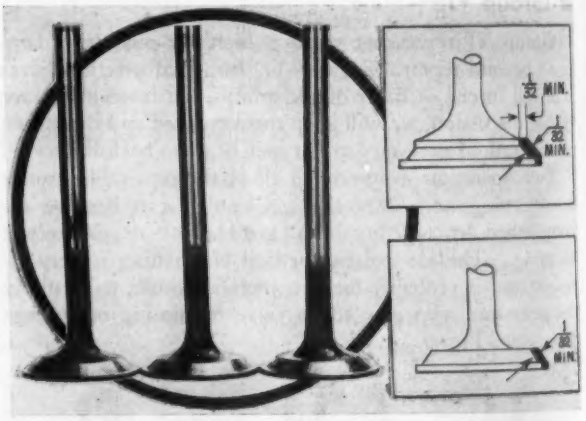
Zyglo or magnaflux are good methods but are not generally available to the average garage or fleet. Magnaflux is very good for ferritic valves, but cannot be used for austenitic valves.

We recommend that all valves be subjected to one of the suggested methods before any repair work is started. If any signs of cracks are found in the valve neck, it should be scrapped.

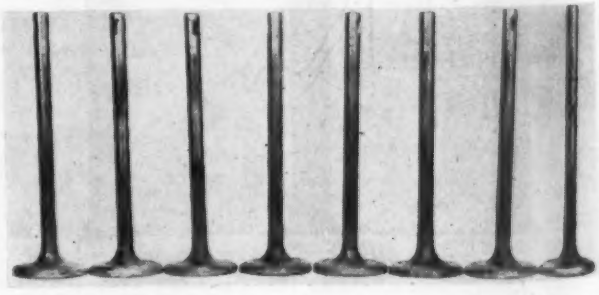
After inspection, shot-blasting will serve as an aid to cover and close any minute cracks that may have been present but undetected. This also work hardens and increases tensile strength. Shot-blasting is highly recommended, if available.

■ Repair – Group I

After inspection, the valves are ready for repair operations. In the case of Group I, the seat and tip need only



■ Fig. 8 – Valves of Group II prepared for application of metal to the seat



■ Fig. 9 – Valves of Group II after metal has been welded to the seat

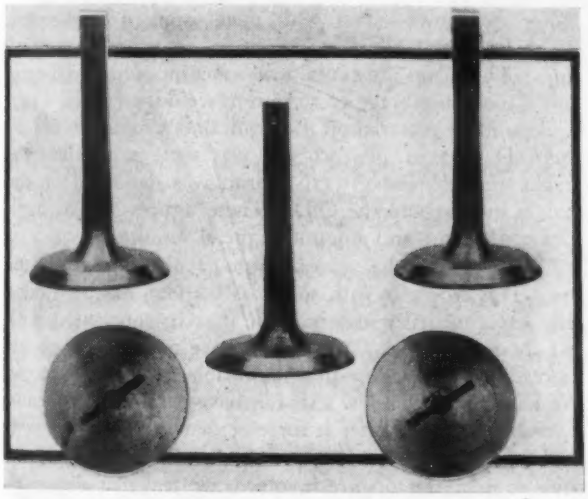
regrinding and the valve is ready for use. This type of repair is common to everyone so no further discussion is required.

■ Repair – Group II

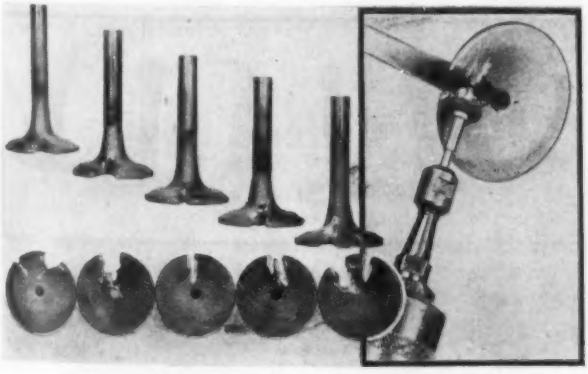
Our Group II valves which require metal added to the seat present a greater problem to us.

First, the valve is prepared for the application of metal by grinding a groove or recess in the seat area. See Fig. 8. This recess may be formed as illustrated depending on the equipment available and the design of the valve. Generally speaking, we recommend a minimum thickness of metal of 1/32 in. and it may go up to ⅛ in. in thickness or more. As a rule, the heavier the coating applied, the more skill and care is required to make a sound weld. It is common practice in the valve industry to manufacture original-equipment valves with stellite ranging from 1/32 to 1/16 in. in thickness.

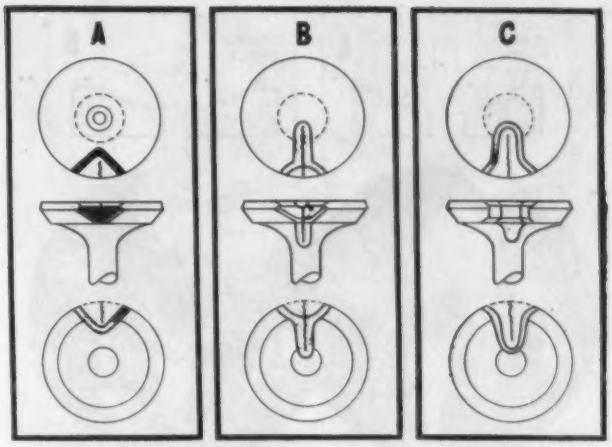
Fig. 9 shows Group II valves with hard-facing applied. For this group we recommend the use of any hard-facing material which is on the market. If hard-face material is not available, we recommend stainless weld rod 18-8, 19-9, or 25-20. We have not run sufficient tests to make recommendations as to the relative merit of these materials. Anyone interested should contact the various manufacturers of this type of material and get detailed instructions on its use.



■ Fig. 10 – Valves of Group II completely finished, ready for reuse



■ Fig. 11 – Valves of Group III – crack surfaces are ground to the base metal in preparation for welding the cracks



■ Fig. 12 – Valves of Group III – preparing cracks for welding

Fig. 10 shows the Group II valves completely finished, ready for reuse. This picture illustrates valves that have the under section of the head and part of the radii cleaned up by grinding. This grind operation is not absolutely necessary. For appearance sake, it is recommended; from an operation standpoint, it will operate just as well without it; so, this is left to the discretion of the operator. It will depend largely on the availability of equipment for the operation. This type of repair, although not too common, is used to good advantage by some fleet operators. It is definitely more expensive than using a new valve. We might add that the largest percentage of valves discarded fall into this group; so, this operation carried out on a large scale would give considerable relief to the demands for replacement valves.

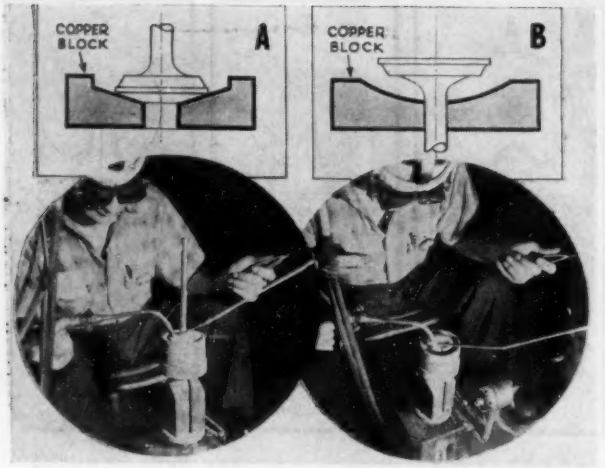
■ Repair – Group III

We will now consider Group III. Valves of this type would normally be scrapped and we would also highly recommend the same if there were any possibility of obtaining a new one. However, if there is no alternative, we can offer some relief. By carefully preparing the sample and by doing a good welding job, we can make them serve a few thousand extra miles. This is for a case of strict emergency and may, under these conditions, be found to be satisfactory and helpful.

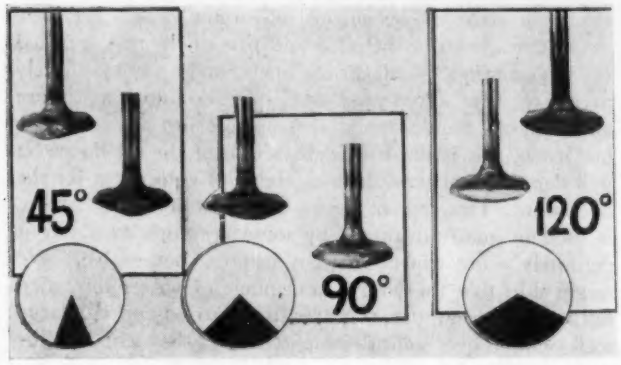
Valves with deep cracks or burned spots are prepared in a manner similar to the way a dentist prepares a tooth for filling. All burned stock is thoroughly ground away. The surface of the cracks must be ground to absolutely base metal. This grinding can best be accomplished with a small hand, pneumatic, or electrically driven grinder. See Fig. 11.

To illustrate better the valve preparation, we have Fig. 12, illustrations *A*, *B*, and *C*. Illustration *A* shows a small radial crack which is prepared by removing a small V-shaped section, grinding a small chamfer on both the top and under section.

Illustration *B* shows a valve with a deep radial crack. This can be prepared by removing a small V-shaped section on the outer edge, and merely grinding a chamfer on the edges of the crack. This valve will have a crack remaining in the heart of the weld and is not the best method of preparation. A better way to prepare the deep cracks is to remove the entire section as is illustrated in *C*, again chamfering the outer edges.



■ Fig. 13 – Valves of Group III – supports for valves during welding



■ Fig. 14 – Valves with cracks welded

Caution and care must be taken to avoid as much as possible all signs of cracks and areas not cleaned up to base metal. This point cannot be emphasized too highly, as success or failure of the job may depend entirely on this operation.

After this preparatory operation is performed, the actual welding job is begun. Since welding of this particular nature entails working with a relatively large mass of molten metal, it is essential that a skilled man be given the job. Thin sections are very easily overheated and melted away.

The use of a copper pad or block is recommended to help control solidification, prevent burning thin sections, and support the valve in the proper position. In sketch *A*, Fig. 13, the concave base gives support to the edges of the head which require cooling and does not remove heat too rapidly from the heavier section. A soft reducing oxygen-acetylene flame is required for welding most welding rods. The choice of the size of tip required should be at the discretion of the welder. Usually tip Nos. 2, 3, and 4 are adequate with the use of a W-15 Oxweld torch. The selection of the proper welding rod is mainly a matter of personal choice; nichrome (80 Ni, 20 Cr), No. 6 stellite, other hard-face materials, and stainless rods have all been used. The size of rod for easier application is ⅛ in. in diameter, although 3/16 in. is quite commonly used.

A preliminary preheating and slight melting of the valve material before application of the rod may, in some cases, remove the remains of a shallow crack at the bottom of the V-grind. The metal should be applied in small drops, without any sudden removal of the torch from the molten area. Keeping the welding rod close to the flame between applications aids materially at this time. While filling in the gap, the metal should be applied in an arc along the edges, depending on the surface tension of the molten metal to prevent its dropping through. Near the edge of the head a smaller tip should be used.

After the repair has been made on the under side of the head, it should be turned over, as in *B*, and the same general procedure repeated from the top surface of the head. Care must be taken to prevent excessive melting, since in this case no copper is in contact with the head.

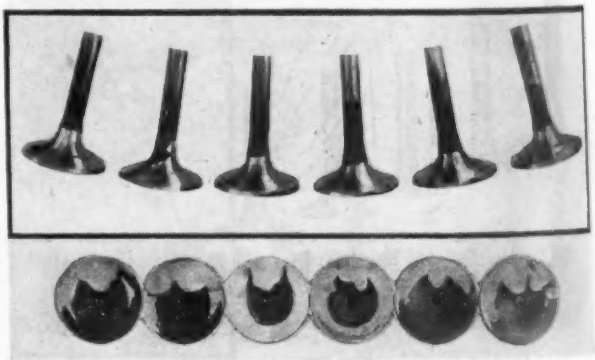
Fig. 14 shows valves after the cracks have been welded. This picture is not part of the Group III series but illustrates to what extent welding can be carried out. At the extreme left are valves with a 45-deg section replaced by welding. In the center is a 90-deg section, and on the right are two valves with 120 deg replaced. This operation requires considerable skill but can be done successfully.

After the cracks and burned areas are filled with weld metal, the radius and O.D. of the head should be rough ground, the top of the head cleaned, and the seat grooved to allow for application of a hard-surface metal. See Fig. 15. It is necessary to add the metal to the seat since the crack welding operation may create sufficient distortion so that it would be impossible to clean up the seat without it. This groove should be made similar to the ones illustrated for Group II.

Fig. 16 shows valves with metal applied to the seat. At this point it may be interesting to consider welding time. The valves illustrated here are 2 in. in head diameter and it took an average of 16 min to weld the cracks, 6 min to deposit the material on the head, and 5 min to build up the O.D.; 6½ in. of 3/16 rod was used in welding the cracks and 3½ in. of 3/16-in. rod was deposited on the seat in building up the O.D. These figures give a rough idea of the time and amount of metal required.

Fig. 17 shows the valves finish ground and ready for use. This type of repair, so far as I know, has been done only experimentally and most of it is in connection with our committee work. Some fleets have made welds of this kind for emergencies with a reasonable amount of success. We have run only a few tests but have obtained sufficiently favorable results to offer it for emergency repairs.

Since our work on this project is still incomplete and since so many factors are involved, we feel that on valves which have one or more deep cracks, it is practically impossible to make any predictions as to the results one may



■ Fig. 15 – Valves of Group III with cracks welded and prepared for welding seat

expect. We must bear in mind that valves coming to us in this condition took a terrific beating in their original run, so unless we are completely surprised, don't expect too much from them.

■ Repair – Group IV

This concludes repairs on valve heads, and we now come to Group IV – worn valve stems. Valves with worn stems can be repaired by metallizing – either of two methods of preparation may be used.

The stem may be cut 0.020 in. undersize on the radius, which is 0.040 in. on the diameter, and fuse bonded, or it may be cut the same amount undersize with a square-nosed tool, approximately 0.050 in. wide with 16 threads per inch and steel grit-blasted to form a key for the sprayed metal. The prepared stem is then built up to 0.010 to 0.015 over the required size with Tufton or any stainless steel. The stems are then ground and the valves are ready for use. In doing this operation an operator can make the stems oversize to fit an oversize guide in the event new guides are unobtainable. In grinding metallized material, it is recommended that a 3746 K 5 Crystolon (Norton Co.) or a 36 NE (Carborundum Co.) wheel be used. Ordinary wheels load and glaze too rapidly. This process is used quite commonly and can be safely recommended. See Fig. 18.

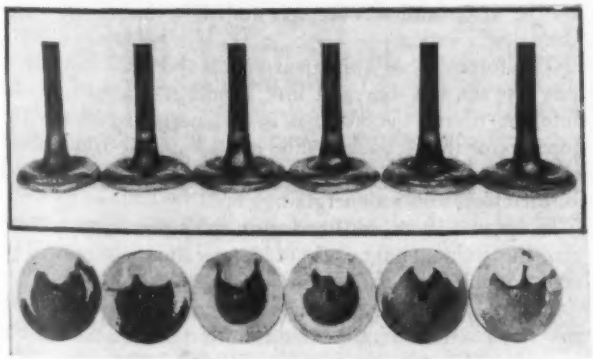
Another method for reclaiming stems is chrome plating. Worn stems should be ground to remove all oxide and then plated 0.001 to 0.002 large and ground back to the desired size. This process may not be available as a general rule, but is satisfactory from an operation standpoint.

■ Repair – Group V

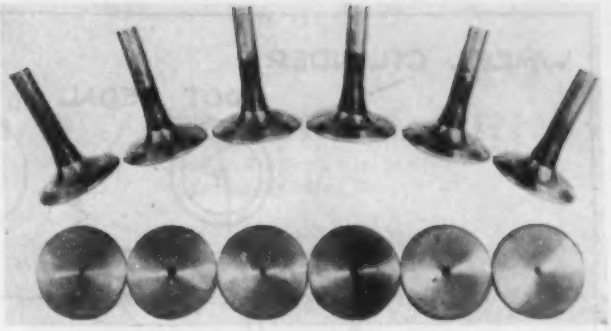
Valves needing repair on the stem tips or retainer groove must be repaired by welding. Generally speaking, the most common groove wear comes with the old horseshoe lock or on installations where the valve keys and spring caps were loose or improperly fitted. Most wear conditions on both the groove and tip can be repaired by welding with stainless or any other hard-face material. With reasonable care the groove can be machined or ground to the desired size.

■ Repair – Group VI

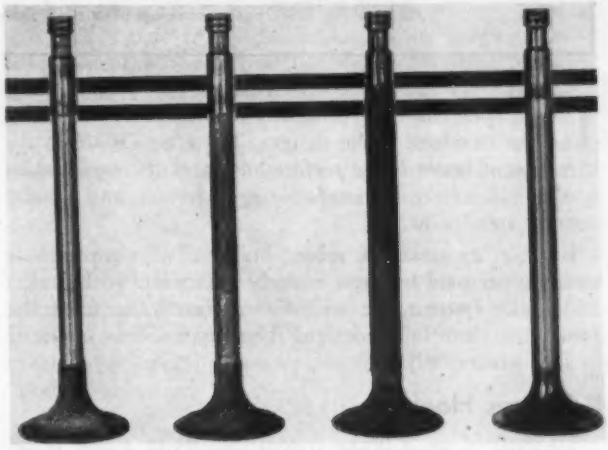
Group VI listed as any valve that is considered irreparable. It is impossible for us to describe a valve of this kind in detail, since this will vary from one shop to the next, depending on the equipment available and need for this type of work. Generally speaking, valves with heads broken off, tip ends of stems broken, or badly bent stems, can be considered in this classification.



■ Fig. 16 – Valves of Group III with seats welded ready for finish grind



■ Fig. 17 – Valves of Group III finish ground and ready for use



■ Fig. 18 – Valves of Group IV – procedure for metallizing the stems

■ Conclusions

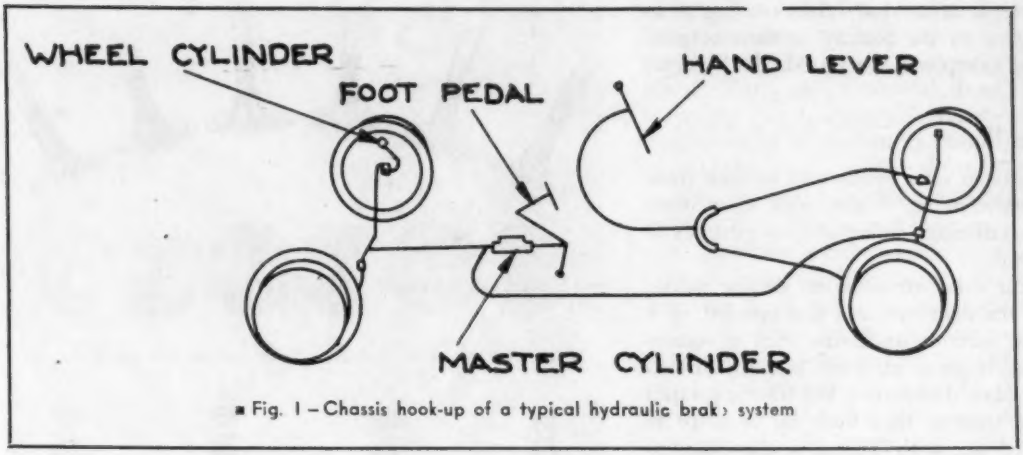
In closing, it must be considered that the repair of automotive valves is quite complicated so it can be carried out best in a small department, set up especially for this purpose, with suitable equipment to carry on the required operations.

In our opinion the prime requisites for satisfactory repair and renewal of valve faces and stems are:

1. Good preparation.
2. Welding skill.
3. Good, smooth flowing rod.
4. Skilled grinding and machining to produce the finished product.

In writing this paper, we did not intend to write an instruction sheet for these operations. We merely want to point out, in a general way, the work and operations that are required.

Our committee work has not progressed to a point where we can make definite commitments. We are still working on this problem and hope to write, at some later date, a complete report on this subject. At this early date, it was our intention to explain the work we have done thus far, and, if any of this discussion presents ideas that can be worked out by our garages and fleet operators to help keep equipment rolling, we will feel that our purpose is well fulfilled.



■ Fig. 1 – Chassis hook-up of a typical hydraulic brake system

BRAKE

THIS paper has been compiled to present our thoughts on the trends of brake design. These trends are in the direction of better *brake performance* and of *simplification* of the brakes from a *manufacturing*, a *service*, and a *maintenance* standpoint.

In order to make the subject-matter of this paper more easily understood by those entirely unfamiliar with brakes and brake systems, we will discuss, briefly, some of the fundamentals of brake design. The paper will be confined to shoe brakes only.

■ Chassis Hook-Up

To most of us, a brake system consists of a foot pedal and a hand lever; so, to start with, we will show in Fig. 1 what a typical hydraulic brake system looks like under the floor boards.

When the foot pedal is depressed, brake fluid is pushed out of the master cylinder through tubes and into brake wheel cylinders which spread the shoes out into the drum to give braking action. Usually, the brakes in the rear wheels are also connected to a hand lever by a mechanical hook-up for parking purposes.

■ Types of Brakes

We will consider the three most common types of internal shoe brakes, namely, the nonservo, the servo, and the balanced brake.

Very often the terms "servo" and "self-energizing" are considered as being the same. However, they are not – and right here would be a good place to explain how they differ. This can be done most clearly with diagrams.

Nonservo Type – Fig. 2 shows a two-shoe, nonservo brake. These shoes are anchored at the bottom ends and spread apart at the top ends by means of a mechanical cam or a hydraulic wheel cylinder to press the shoes against the drum. When this is done, while the drum is rotating counterclockwise, as shown in the diagram, there is a frictional force produced at the drum, represented by F_1. You will note that this force has a moment arm a about the anchor, which tends to rotate the left-hand shoe counterclockwise about its anchor, and thus press it harder into the drum. This is self-energization, and results in an increase in the frictional force F_1, the same as if the applying force had been increased.

[This paper was presented at a meeting of the Chicago Section of the SAE, Chicago, Ill., April 11, 1944.]

Applying the right-hand shoe produces a frictional force F_2 at the drum in the direction shown by the arrow. This force has a moment arm b about the anchor pin and tends to rotate the shoe counterclockwise and *away* from the drum. This shoe is termed nonenergizing, and results in a decrease in the frictional force F_2, the same as if the applying force had been decreased.

The frictional force F_1 on the forward shoe may be several times greater than the frictional force F_2 on the reverse shoe, depending on the coefficient of friction of the linings of the two shoes.

Thus a brake of this type contains a self-energizing shoe and a nonenergizing shoe – but no servo action.

Servo Type – The diagram in Fig. 3 shows a servo brake. In this type, the shoes have a common anchor at their top ends. They are applied at a point near the anchor, and are connected at the bottom by a floating link, which is usually made adjustable to compensate for lining wear. When the shoes are spread apart by the cam or hydraulic cylinder, and the drum is rotating in the direction shown by the arrow, a frictional force F_1 is produced at the drum. Since this shoe, which is known as the primary shoe, is not anchored to the backing plate at its lower end, it starts to rotate with the drum. As may be seen, this movement and force is now transmitted to the other shoe, which is known as the secondary shoe, through the connecting link to become the applying force of the secondary shoe. This is servo action.

It must be remembered that, although the primary shoe is not anchored directly to the backing plate, the frictional force F_1 produced by this shoe still acts as a retarding force on the drum and this force is added to that of the secondary shoe.

The force F_1 has a moment arm a about the connecting pin between the shoe and link, tending to press it harder into the drum; therefore, it is an energizing shoe. The force F_2 of the secondary shoe has a moment arm b about the anchor, tending to press it harder into the drum and is, therefore, also an energizing shoe.

If the two shoes are lined with the same friction material, the frictional force F_2 of the secondary shoe will be about double that of the primary shoe F_1.

Thus a servo brake contains two energizing shoes and also servo action. In the opposite direction of braking, the secondary shoe becomes a primary shoe.

Balanced Type – Fig. 4 shows a balanced brake, with the two shoes individually applied and individually an-

KE DESIGN TRENDS

by B. E. HOUSE and R. A. GOEPFRICH
Bendix Products Division,
Bendix Aviation Corp.

THE trend of brake design is in the direction of better brake performance and simplification of the brake from the manufacturing, service, and maintenance points of view.

In the six types of brakes reviewed by the authors, performance was improved by center-mounting them with respect to both the applying and the anchoring forces.

Simplicity in manufacture and service was achieved by reducing the number of pieces in the brake to a minimum by eliminating many conventional parts.

Simplicity in maintenance is accomplished by eliminating major or anchor-pin adjustment, by simplifying the adjustment for lining wear, and by making it easy to disassemble and reassemble the shoes when they are relined.

THE AUTHORS: B. E. HOUSE (M'17), chief engineer of the military brake department of Bendix Products Division, Bendix Aviation Corp., has been associated with the automotive industry continuously since 1911. Before joining the Bendix company in 1928, he was employed by Timken-Detroit Axle Co., Remy Electric Co., Pan Motor Co., and Buick Motor Co. He served in the Signal Corps in France during World War I. R. A. GOEPFRICH (M'44) is assistant chief engineer of the military brake department of Bendix Products Division. A graduate of the University of Notre Dame, Mr. Goepfrich became connected with Bendix as brake development engineer in 1927. He has patented over 60 inventions, mostly concerned with improvements on brakes and brake design.

chored at diametrically opposite points, as shown in the diagram. When the drum is rotating in the direction shown by the arrow, and the shoes are individually actuated to press them against the drum, the frictional force F_1 of the top shoe has a moment arm a about its anchor tending to press it harder into the drum; and the frictional force F_2 of the bottom shoe has a moment arm b about its anchor tending to press it harder into the drum; thus both shoes are self-energizing. However, this brake contains no servo action.

If the applying forces and the types of linings on the two shoes are alike, the frictional forces F_1 and F_2 will be the same, and both shoes will do the same amount of work. Also, if the brake is constructed with geometrically similar applying forces and anchoring points for the opposite direction of drum rotation, then the brake will have the same effectiveness in both directions.

As a comparison of the torque capacity of the various types of brakes which we have discussed in this paper, we have plotted a curve, as shown in Fig. 5.

You will note from this curve that the balanced brake is somewhat more effective than the nonservo, and that the servo is more effective than the balanced brake.

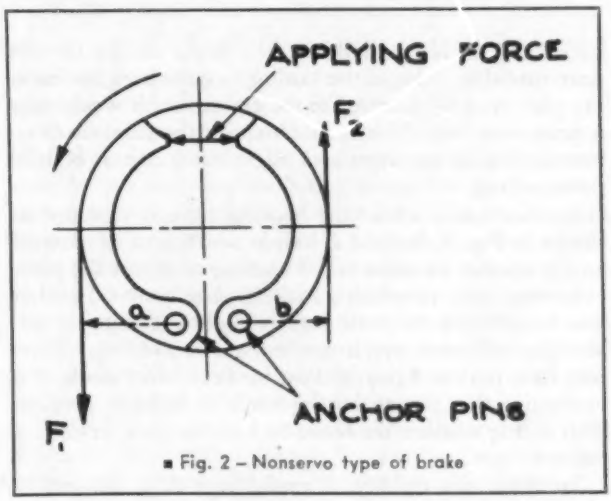
This can be attributed to the fact that the balanced brake has two self-energizing shoes as against one self-energizing and one nonenergizing shoe of the nonservo brake, while the servo brake has two self-energizing shoes plus servo action.

The truck, bus, and automobile fields, due to high-speed operation, require brakes having good controllability, and balanced brakes are desirable for these types of vehicles. The industrial field, which is usually low-speed operation, requires an effective brake, for which the servo type is most suitable.

Anchor Loads

In most brakes of today, whether they be of the nonservo, servo, or balanced type, the braking load L is transmitted to the anchor, as indicated diagrammatically in Fig. 6, and thence through the backing plate or the mounting plate to either the axle housing on the rear brakes, or to the steering knuckle on the front brakes. Also, in most of the conventional brakes, the anchor is mounted to a comparatively thin backing plate, as shown diagrammatically in Fig. 7, and the braking load is transmitted to the anchor pin at a distance from the backing plate, as indicated by the arrow L.

This cantilever loading, with its moment arm a, will cause the backing plate to distort. When the distortion is in such a direction that it causes the shoe to wedge be-



■ Fig. 2 – Nonservo type of brake

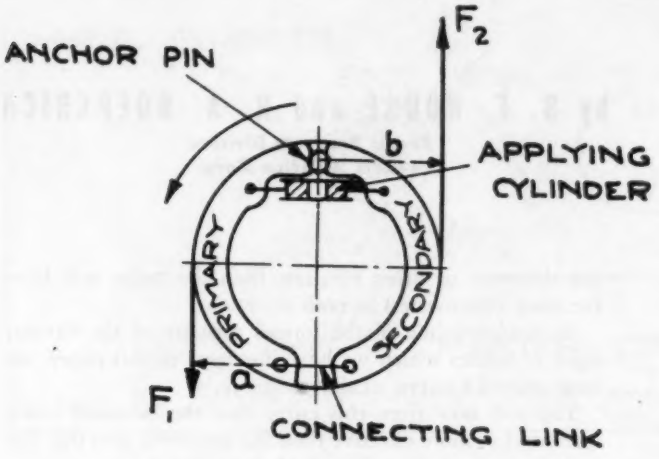
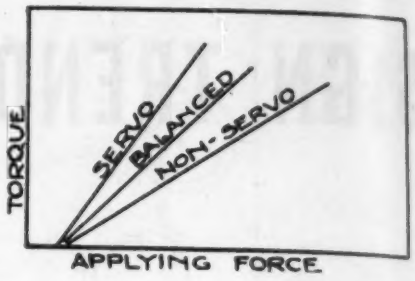
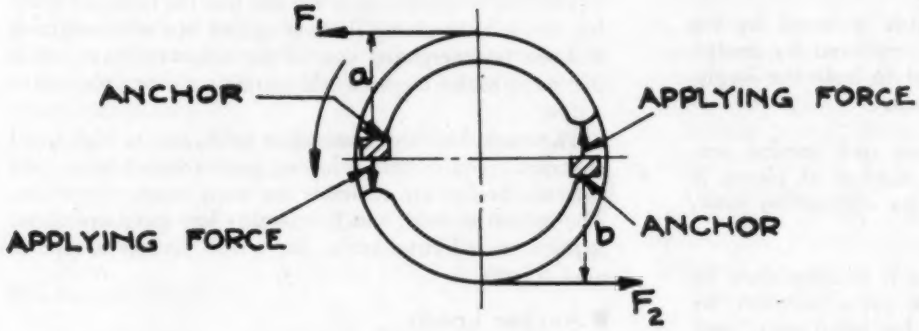


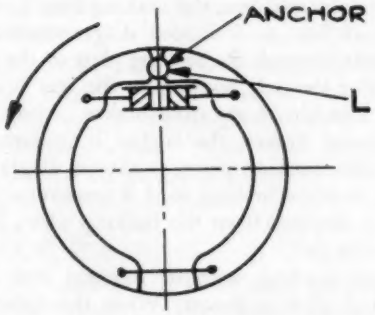
Fig. 3 (left) – Servo type of brake



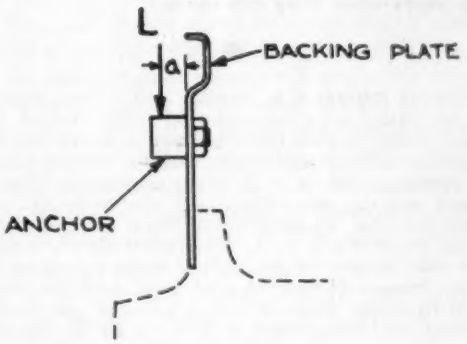
■ Fig. 5 – Comparison of torque capacity of various types of brakes



■ Fig. 4 (left) – Balanced type of brake



■ Fig. 6 (left) – Anchor loads



■ Fig. 7 (right) – Mounting of anchor

tween the anchor and drum, the brake might become uncontrollable. Also, if the loading is quite high the backing plate may be distorted to the extent that it would take a permanent set. This would destroy the original clearance setting of the shoes and might cause one or both of them to drag.

In some cases, where the backing plate is mounted as shown in Fig. 8, the load L has, in addition to its moment arm a, another moment arm b tending to distort the plate.

In some tests, in which a backing plate mounted anchor was loaded with the same load it might encounter in service, the deflection which resulted was astounding. These tests have indicated that, at least for heavy-duty work, it is imperative that the anchor be made as rigid as possible. This is true whether the brake be a nonservo, a servo, or a balanced type.

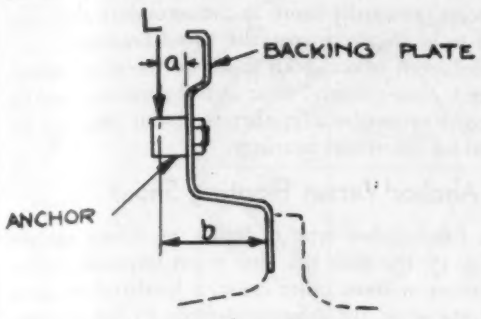
To attain this rigidity, it would be well if the anchor load were symmetrically located with respect to both the backing plate and the axle mounting, as shown in Fig. 9.

■ Drum Distortion and Lining Wear

Since neither drums, shoes, anchors, nor backing plates can be made perfectly rigid, the forces imposed upon these various elements during a brake application cause deflections or distortions.

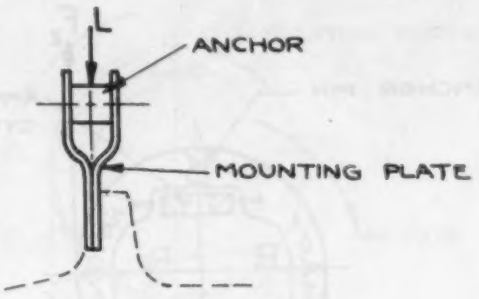
Nonservo Type – Let us assume that in this brake the two shoes are lined with materials having the same coefficient of friction and a straight-bore wheel cylinder, that is, a wheel cylinder having the same diameter bore at each end, is used.

For a given line pressure, this straight-bore cylinder produces equal applying forces A and B on the two shoes. See Fig. 10. Statically, that is, with the drum not rotating, these applying forces produce equal radial forces, P_1 and



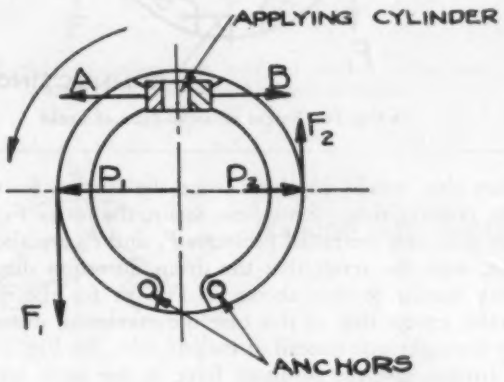
■ Fig. 8 (left) – Mounting of anchor with additional moment arm tending to distort the plate

■ Fig. 9 (right) – Anchor load symmetrically located with respect to the backing plate and the axle mounting

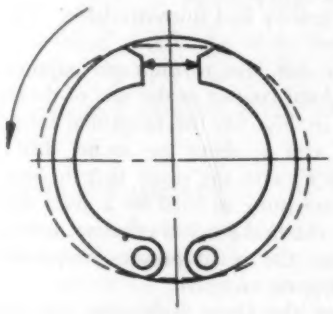


P_2, on the two shoes. However, we found earlier that when a drum is rotating in the direction shown by the arrow, the frictional force F_1 of the forward shoe is increased due to its energizing action. Since this has the same effect as increasing the applying force A, it also has the same effect as increasing the radial pressure P_1. Conversely, the frictional force F_2 of the reverse shoe is decreased due to the nonenergization of this shoe, with a proportional decrease in the effective radial pressure P_2.

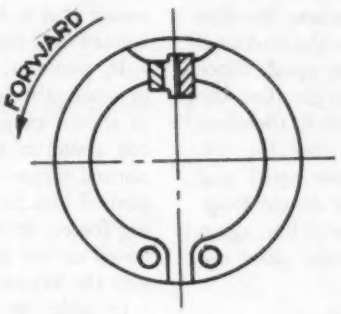
The forces P_1 and P_2 cause the drum to distort outwardly at the points where these forces are greatest. Since force P_1 is greater than force P_2, as was shown above, the left side of the drum will be distorted outwardly farther than the right side. The distortion diagram is shown exaggerated by the dotted lines in Fig. 11, the solid line being the undistorted drum.



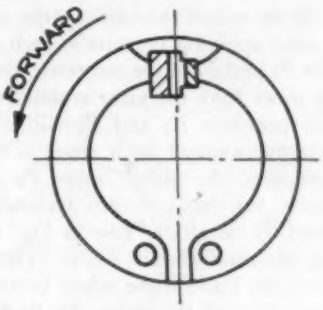
■ Fig. 10 – Forces acting on nonservo type of brake



■ Fig. 11 – Distortion diagram for nonservo type of brake



■ Fig. 12 – Double-diameter wheel cylinder with larger diameter acting on the rear shoe



■ Fig. 13 – Double-diameter wheel cylinder with larger diameter acting on the forward shoe

The difference between the radial forces P_1 and P_2 constitutes a resultant force acting in the direction of the greater, or to the left in this case. This force is imposed upon the wheel bearings.

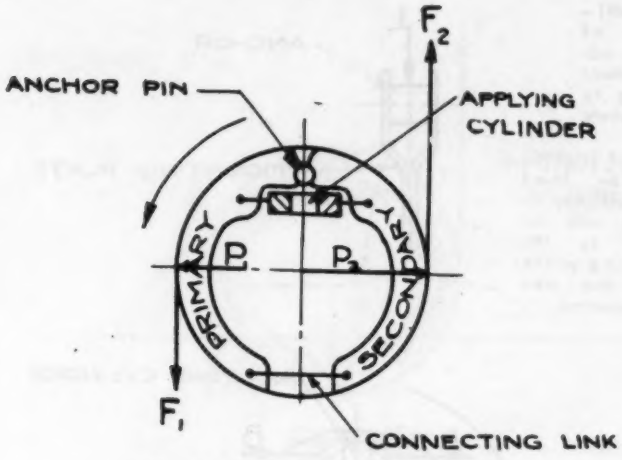
Since lining wear is a function of the radial pressure, the forward shoe, having the greater radial pressure, will wear faster than the reverse shoe. This may be equalized somewhat by putting a slower wearing lining on the forward shoe, or a faster wearing lining on the reverse shoe. However, it is generally, although not universally true that a slow-wearing lining has a lower coefficient of friction than a fast-wearing lining. Some manufacturers have tried to equalize the wear of the two shoes by using a double-diameter wheel cylinder, in which the larger diameter acted on the rear shoe, as shown in Fig. 12.

Another object was to make the brake more effective by making the rear shoe do more work than it would with a conventional straight-bore cylinder.

This arrangement has the disadvantage that if the forward cylinder diameter is made smaller than a normal straight-bore cylinder, the brake would become less effective; conversely, if the rear cylinder is made larger, it would require more pedal travel for the same shoe movement and consequently necessitate more frequent adjustments.

Another arrangement which has been used is shown in Fig. 13. In this case, the larger diameter cylinder was used on the forward shoe to increase the effectiveness of the brake. This, of course, upset the equal wear considerably but was accepted by the manufacturer.

Servo Type – Let us assume that both shoes of this servo brake are lined with materials having the same coefficient of friction. In this case the friction force F_2 of the secon-



■ Fig. 14 – Forces on servo type of brake

dary shoe would be about twice the friction force F_1 of the primary shoe. Since here, again, the forces F_1 and F_2 are different, the radial pressures P_1 and P_2 are also different, with the result that the drum distortion diagram is very similar to that shown in Fig. 11 for the nonservo brake, except that in this case the maximum distortion is on the right side instead of the left side. See Fig. 14.

In this case the resultant force to the right imposes a load on the wheel bearing and also causes the secondary shoe to wear faster than the primary shoe.

Balanced Type – In the balanced brake, if both shoes are lined with materials having the same coefficient of friction, and if the wheel cylinders or the applying means produce the same applying pressure at each shoe, then the frictional forces F_1 and F_2 of the respective shoes will be equal. Since both shoes have the same amount of self-energization, the radial pressures P_1 and P_2 will also be equal; therefore the lining wear of both shoes is the same. See Fig. 15. Since, now, the radial forces P_1 and P_2 are equal and opposite, the drum distorts symmetrically, as shown exaggerated by the dotted line in Fig. 16, the solid line again being the undistorted drum. Thus this brake does not impose any load on the wheel bearings.

Summarizing the above, we find that:

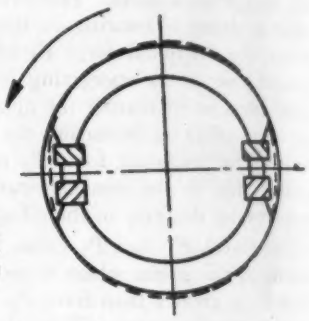
In the nonservo brake, the forward shoe usually does more work and wears out faster than the reverse shoe, and that the drum distorts outwardly more at the forward shoe, causing a load to be imposed upon the wheel bearings.

In the servo brake, the secondary shoe usually does more work and wears out faster than the primary shoe, and the drum distorts outwardly more at the secondary shoe causing a load to be imposed upon the wheel bearings.

In the balanced brake, both shoes do the same amount of work and, consequently, wear at the same rate, and the drum distorts symmetrically; therefore, this does not impose a load on the wheel bearings.

■ Fixed Anchor Versus Floating Shoes

If, in a fixed-anchor type of brake, as shown exaggerated in Fig. 17, the shoe has, due to an improper anchor-pin adjustment or some other cause, a localized contact at the toe of the shoe, the tangential force F_1 has a greater moment arm a_1 about the anchor than the normal tangential force F, with the result that the shoe is pushed harder into the drum for a given applying force. In this case, the brake is more effective, and may be so to the extent that it becomes grabby and uncontrollable. This is termed "toe contact."

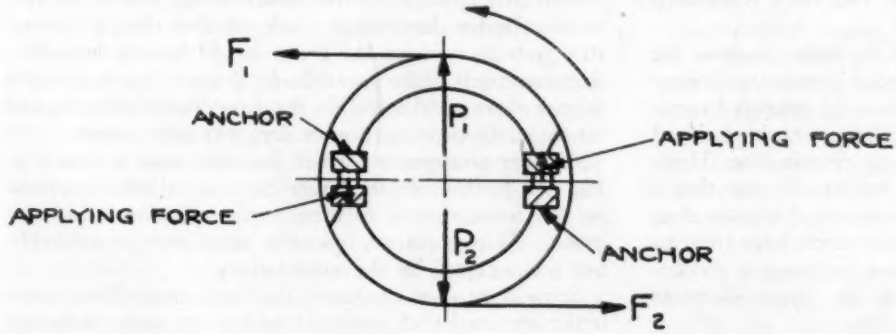


■ Fig. 16 – Distortion diagram for balanced type of brake

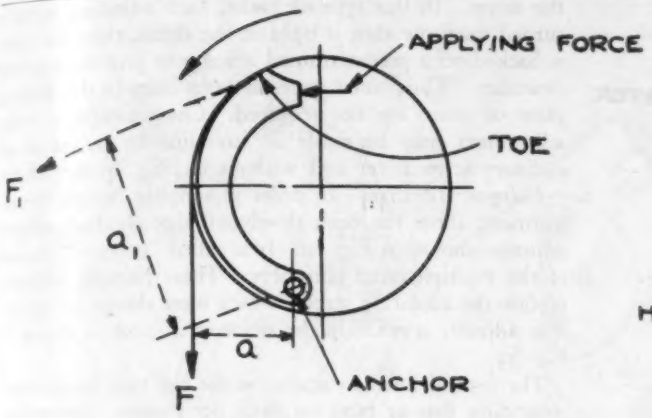
If, however, the shoe has, due to improper adjustment or some other cause, a local contact at the heel of the shoe, as shown exaggerated in Fig. 18, the tangential force F_1 has a shorter moment arm a_1 about the anchor than the normal tangential force F, with the result that the shoe is pushed into the drum not quite so hard for a given applying force. In this case the brake is less effective, and may be so to the extent that the pedal pressure required to stop the vehicle may become excessive.

In order to overcome the above difficulties, the fixed round anchors may be replaced by a flat anchor, as shown in Fig. 19. With the flat anchor, the maladjustment termed "toe or heel contact," which is possible with the fixed anchor, cannot exist because the shoe can slide on the anchor and thereby center itself in the drum.

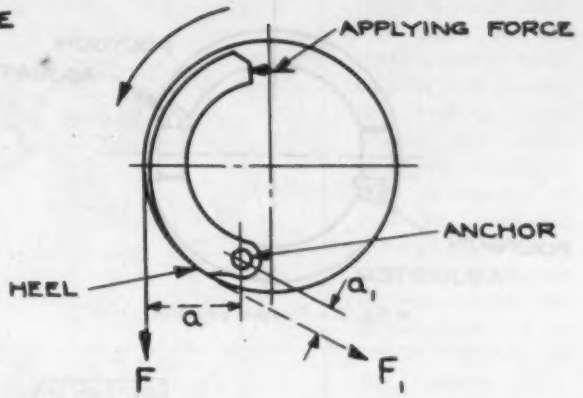
This floating-shoe principle may also be incorporated in



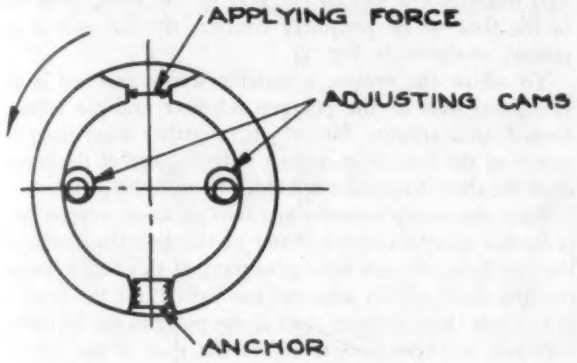
■ Fig. 15 – Forces on balanced type of brake



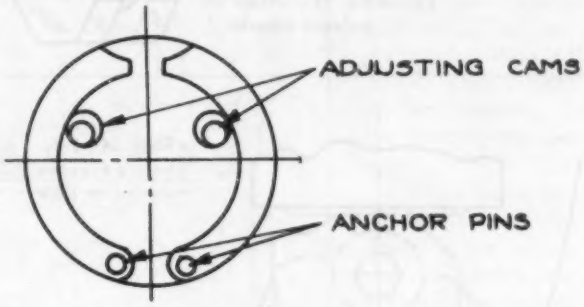
■ Fig. 17 – Forces resulting from localized contact at the toe of the shoe of a fixed-anchor type of brake



■ Fig. 18 – Forces resulting from localized contact at the heel of the shoe of a fixed-anchor type of brake



■ Fig. 19 – Application of the floating-shoe principle



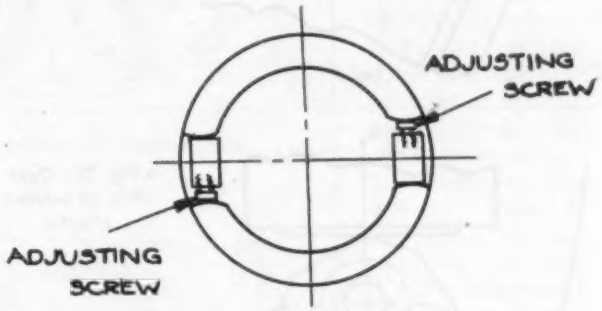
■ Fig. 20 – Adjusting cams used on conventional anchored-shoe brake

the servo and balanced brakes to eliminate the anchor adjustment.

■ Brake Adjustment

In the conventional anchored-shoe brake, it is usual to provide two adjustments; one to adjust the shoes initially to compensate for manufacturing tolerances; the other to adjust the shoes to compensate for lining wear.

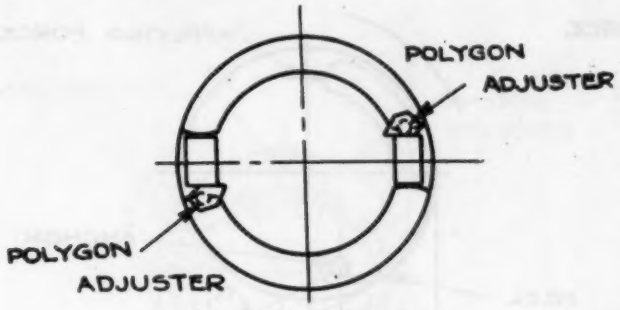
From Fig. 20 it will be seen that when the shoe is adjusted outwardly by means of the adjusting cam, it will travel in a predetermined path about the anchor pin. If, now, due to manufacturing tolerances, the dimensions of the shoes, the location of the anchor pin, and so on, are such that when the shoes are adjusted toward the drum, the clearance at the toe of the shoe is much greater than at the heel of the shoe, or vice versa, something must be done about it. Consequently, the anchor pins are usually made eccentric so that the position of the shoe may be changed to give the proper clearances at each end of the shoe. To accomplish this, either the drum or backing plate is provided with openings through which a feeler gage may be inserted between the drum and shoe lining, and the eccentric anchor pin is turned until this feeler gage is snug. Each end of each shoe is adjusted to the proper clearance in this manner. This anchor-pin adjustment is usually only necessary when the brakes are first installed on the vehicle, and when the shoes are relined. It, together with the cam adjustment, which is also necessary when adjusting the anchors, is known as a major adjustment.



■ Fig. 21 – Adjusting screws used on balanced brake

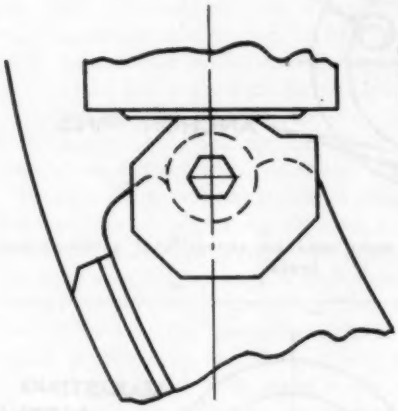
Subsequent adjustments to compensate for lining wear are made by the cam adjustments alone, and are called a service or minor adjustment.

In the balanced brake, shown in Fig. 21, the rounded ends of the shoes anchor on the flat ends of the cylinder pistons in one direction of drum rotation, and on the flat heads of the adjusting screws in the other direction of drum rotation. Thus, since the point of rotation of the shoe about its anchor is not fixed, it is called a "floating" shoe. This type of brake does not require a separate or major adjustment to compensate for manufacturing tolerances owing to the fact that when the adjusting screw is turned to move the shoe toward the drum, if either end of the shoe contacts the drum first, the shoe can slide on the flat anchor and flat adjusting screw to find its center in

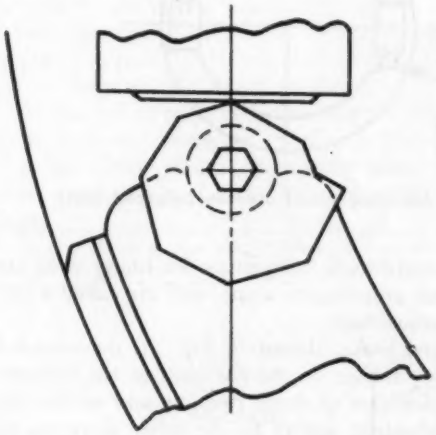


■ Fig. 22 – Polygon adjusters

■ Fig. 23 – Details of polygon adjuster



■ Fig. 24 – Polygon adjuster mounted on shoe



■ Fig. 25 – Operation of polygon adjuster

the drum. In this type of brake, each adjusting screw is turned until the shoe is tight in the drum, then the screw is backed off a predetermined amount to give the required clearance. Thus feeler gages or peep holes in the backing plate or drum are not required. Consequently, a brake adjustment may be made at any time by means of an ordinary screwdriver and without jacking up the vehicle.

Polygon Adjusters – In order to simplify further the adjustment, there has been developed the so-called polygon adjuster, shown in Fig. 22. It is called "polygon" because of the multiple-faced periphery. These polygon adjusters replace the adjusting screws which were shown in Fig. 21. The adjuster is made in the shape of a spool, as shown in Fig. 23.

The central hub is circular, while the two flanges have coinciding flats or faces on their peripheries. Succeeding faces are located at greater distances from the center of the hub.

In practice, the polygon adjuster is assembled with its hub fitted into a slot in the end of the shoe, while one of the flats on its periphery contacts the flat end of the pistons, as shown in Fig. 24.

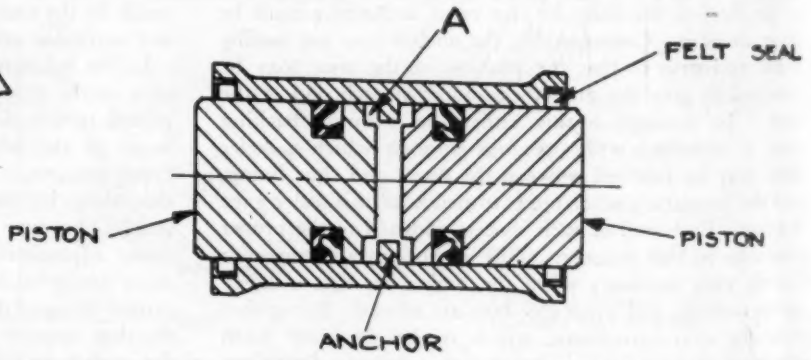
To adjust the brakes, a suitable tool is inserted in the hexagonal hole of the polygon adjuster and the adjuster turned until another flat, which is farther away from the center of the hub, is in contact with the end of the piston; thus the shoe is moved outwardly toward the drum.

Since the corner between any two adjacent polygon faces is farther away from the center of the hub than either of the two faces, we can take advantage of this fact to ensure that the shoes are not adjusted too tightly into the drum so as to cause them to drag; thus if the polygon can be turned over one of these corners before the shoe is too tight in the drum, as shown in Fig. 25, the shoe will drop back slightly after the polygon passes over this corner and clearance between the shoe and drum is always assured. If the polygon cannot be turned over one of these corners after the shoe is tight in the drum, it will drop back to the preceding flat, giving the same clearance as we had and indicating that the brake is not ready for adjustment.

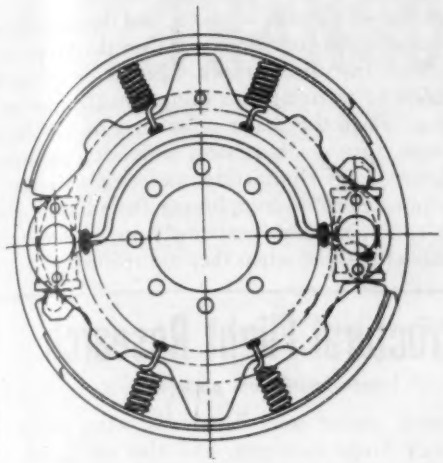
■ Wheel Cylinder

You will note, from Fig. 26, that in this design the anchor is incorporated in the wheel cylinder body. It consists of an annular ring in the wheel cylinder bore. During braking, the braking load of the shoe, which contacts the outer end of the piston is transferred through this piston to this annular ring.

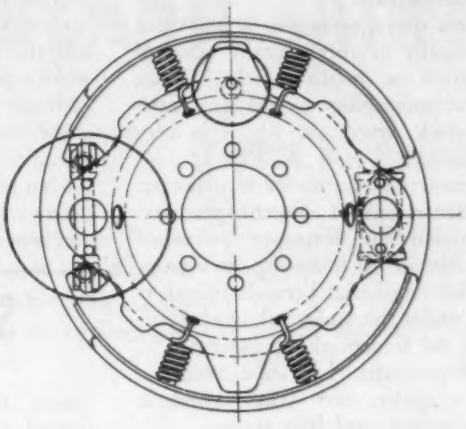
When the brakes are applied, both pistons may leave their seats on the annular ring. Then, when the shoes



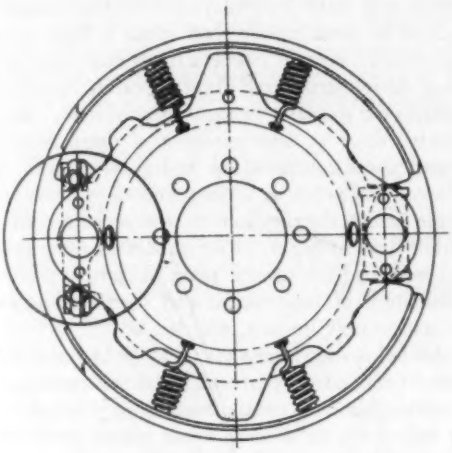
■ Fig. 26 – Wheel cylinder



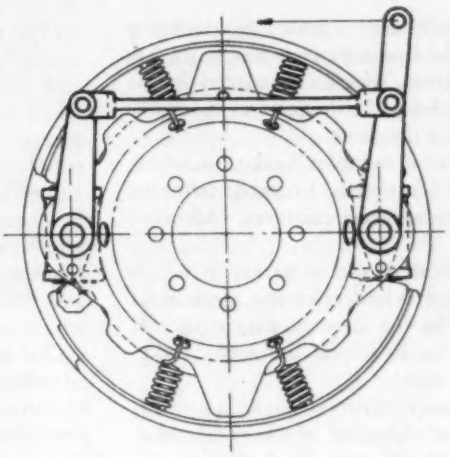
■ Fig. 27 – Hydraulic twinplex brake



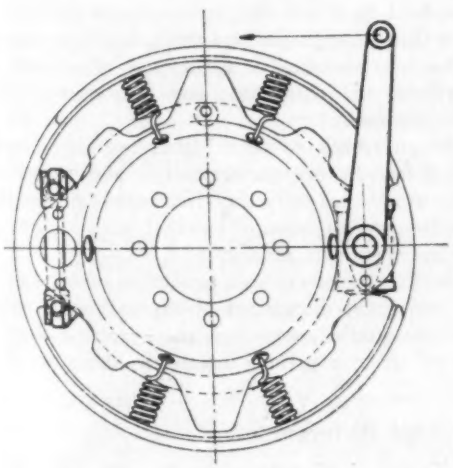
■ Fig. 28 – Hydraulic duo-servo brake



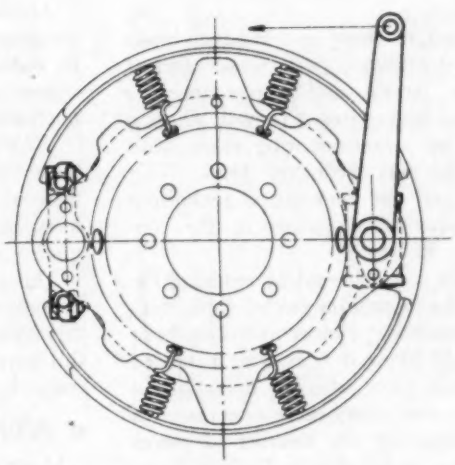
■ Fig. 29 – Hydraulic nonservo brake



■ Fig. 30 – Mechanical twinplex brake



■ Fig. 31 – Mechanical duo-servo brake



■ Fig. 32 – Mechanical nonservo brake

contact the drum, they will rotate therewith until the anchoring pistons move to their seats to take the braking load. During this movement, the volume of fluid in chamber *A* must be forced out through the restricted clearance between the reduced diameter of the piston and the inside diameter of the annular ring. This produces a dashpot or cushioning action to prevent the piston from returning too rapidly to its seat. This eliminates the so-called "click" which is more or less inherent in the floating-shoe design.

For heavy-duty service, it would be desirable to eliminate the rubber boot which serves to keep foreign matter from entering the wheel cylinder.

In most brakes, the wheel cylinder is very close to the drum and the heat produced by braking has a deteriorating effect on the rubber. To overcome this difficulty, there is provided a felt seal as shown. This type of seal is being tested and, to date, has proved satisfactory.

■ Design Trend

In addition to considering brake designs from a manufacturing, a service, and a maintenance standpoint, as mentioned in the beginning of the paper, we believe that interchangeability is also important. For instance, if some of the

various brake components can be made interchangeable between the various brake types, a marked reduction in tool inventory can be accomplished.

Also, some manufacturers today, especially in the truck field, offer either hydraulically or mechanically actuated brakes as optional equipment on certain vehicles. Thus, the above feature of interchangeability would also materially reduce the dealer-stock inventory, which is very desirable.

Keeping in mind the features of simplicity in manufacture, service, and maintenance, and of interchangeability, the Bendix Products Division has developed a line of brakes which we will describe in the following six figures. The balanced type of brake, which has been described, is being produced by Bendix under the trademark "twinplex" and the servo type under the trademark "duo-servo."

Fig. 27 shows a centermounted, hydraulic twinplex brake, which consists of a spider, two wheel cylinders, two shoes, two polygon adjusters, and four springs.

Fig. 28 shows a hydraulic duo-servo brake, in which one of the wheel cylinders of the hydraulic twinplex brake has been replaced by a floating strut and an eccentric added to locate the shoes in the released position. All other brake parts remain the same.

Fig. 29 shows a hydraulic nonservo brake, in which one of the wheel cylinders of the hydraulic twinplex brake has been replaced by a fixed strut, which is mounted in the same manner as the original wheel cylinder was mounted. All other brake parts remain the same.

Fig. 30 shows a mechanical twinplex brake, in which the two hydraulic wheel cylinders of the hydraulic twinplex are replaced by two mechanical wheel cylinders. All other parts remain the same.

Fig. 31 shows a mechanical duo-servo brake, in which one of the mechanical wheel cylinders of the mechanical twinplex brake is replaced by the same floating strut and eccentric as was used in the hydraulic duo-servo brake. All other parts remain the same.

Fig. 32 shows the mechanical nonservo brake, in which one of the mechanical wheel cylinders of the mechanical twinplex brake is replaced by the same fixed strut as was used in the hydraulic nonservo brake. All other parts remain the same.

■ Conclusion

This paper was presented to show the trend of brake design from the standpoint of better performance and of simplicity in manufacture, service, and maintenance.

Thus, in the six types of brakes just reviewed, the performance was improved by centermounting them with respect to both the applying and anchoring forces. This eliminated offset loading and the consequent anchor-pin and backing-plate deflections – vital factors in the controllability of the brake.

Simplicity in manufacture was achieved by reducing the number of pieces in the brake to a minimum by eliminating such conventional parts as anchors, cylinder mounting bolts and lockwashers, shoe hold-down devices, and threaded adjusting parts. Since each piece adds an operation in assembling it to the brake, and many more operations in producing it, materially lessening the number of pieces represents a considerable step in the direction of simplicity in manufacture. Also, by reducing the number of *different* parts, as is accomplished by making all spider plates, cylinder assemblies, shoes, adjusters, and shoe return springs alike, an added advantage is obtained by cutting down the tool inventory.

This reduction of number of pieces and the interchangeability not only of the components within the brake itself, but also between the various types of brakes, simplifies the service problem by reducing, to a minimum, the number of different parts which the dealer must carry on his shelves.

As we have just demonstrated, simplicity in maintenance is achieved by eliminating the so-called major or anchor-pin adjustment, by simplifying the adjustment for lining wear, and by making it extremely easy to disassemble and reassemble the shoes when they are relined.

Structural Flight Research

continued from page 400

mount stresses, engine cowl flap loads, wing flap loads, control surface hinge moments, and also wing, tail, and fuselage loads during maneuvers. With the technique and equipment which is now used, it is believed that many of the measurements are within $\pm 5\%$ and some are within $\pm 2\frac{1}{2}\%$. The general policy at Lockheed has been to make comparatively few basic measurements and to concentrate on the quality of these rather than make a large number of doubtful measurements. Some aircraft companies which have started doing structural flight research have undertaken or attempted to make vast numbers of stress measurements. Such things as determination of detail stress distribution and concentrations are being attempted. The author believes this policy is wrong since it is much easier to determine stress distribution in the structure on the ground rather than in flight. The important thing to concentrate on now and for several years to come is the study and determination of basic static and maneuvering loads which are inaccurately known, not details.

One of the fields in which structural flight research has an important function to perform is in the investigation of forces on airplanes at really high speeds where compressibility effects are so serious, wind tunnel results need flight-test substantiation because of unknown factors such as wind tunnel wall corrections, turbulence, and scale effects. These factors must be studied in flight so that high-speed tunnel results can be proved or corrected.

Another field in which our knowledge is seriously inadequate is that of fatigue stresses on the airplane structure. In order to ascertain the life expectancy of a newly designed airplane, the magnitude and frequency of stress fluctuations are needed.

With design trends toward higher working stresses, lower design load factors, greater speeds, and booster controls which greatly magnify the effectiveness of the pilot, more accurate determination of external loads and the frequency of application is needed.

This paper has purposely been general in nature and has not gone into detail discussions of the various structural flight problems studied, since it is the expressed object of this paper to create a greater interest in structural flight research.

■ Additional Reference

NACA Technical Memorandum No. 678, July, 1932, "Increase in the Maximum Lift of an Airplane Wing Due to a Sudden Increase in Its Effective Angle of Attack Resulting from a Gust," by Max Kramer.

TRANSOCEANIC AIR CARGO

by

LT.-COMDR. NEIL MacCOULL, U.S.N.R. (Ret.)

The opinions and assertions expressed in this paper are the private ones of the author and are not to be construed as official or reflecting the views of the Navy Department or the Naval Service at large.

IN order to make an examination of the fundamentals of air cargo transport, this very complicated subject has been simplified by assuming values for some of the less important variables that are consistent with practice and that may be obtained for specific conditions of flight, if not for a wide range of conditions in any one plane.

The first step in this study of the fundamentals of air cargo transport is a discussion of the division of useful load into fuel and cargo. To compare planes with each other, a term called Ultimate Range has been introduced, which is the range possible in still air, if all the useful load were given up to fuel.

The next step is an examination of the effect of speed on the quantity of cargo which may be delivered in a given time. A term called the Delivery Factor has been introduced as a means of comparing various aircraft on this basis.

This paper examines both the airplane and the airship as cargo carriers. One of the points brought out in regard to the airship is that it develops its highest delivery factor when the fuel used is one-third of the useful load, or one-half of the cargo load, regardless of the length of voyage or size of the ship. This means that the ship's speed for maximum delivery factor would be adjusted so as to consume one-third of its useful load as fuel, regardless of the length of the voyage.

Another point of comparison is the fuel used per ton-mile, which may be a matter of considerable importance where there is a fuel shortage. This is a point where the airship has a pronounced advantage over the plane, especially on long voyages.

In conclusion, it is shown that the fact that an airship can fly at moderate speeds well below those of airplanes, makes possible higher delivery factors for very long voyages than for planes, these airship speeds nevertheless being well in excess of surface speeds at sea.

PART I—AIRPLANES

THE American public has become so air-minded during the last few years that many characteristics of planes are off-hand knowledge, such as the normal speed of various classes of planes, their size, and engine power. Information on the relationship of speed, cruising range, and payload of planes, however, is not as well understood by the layman. In fact, he has but little understanding of any limitation to aircraft performance, apparently believing that some magic of science will always supply whatever he needs in due time. Nevertheless, there are limitations to what science can do.

Limitation to the flying range of aircraft and the relation between range and cargo capacity are admittedly of crucial importance in peacetime. But this subject has become of far greater importance in this war also than is generally realized. With the increased demands for transporting military supplies by surface vessels, there has been an increasing clamor to use planes for this purpose.

Because of the difficulty of comparing various planes when rated for range with entirely different divisions of the useful load into cargo and fuel load, an arbitrary common denominator called the Ultimate Range has been adopted. It is a range in still air, which could be covered by a plane if all the useful load that can be carried is taken up with fuel and oil, thereby leaving no other useful load

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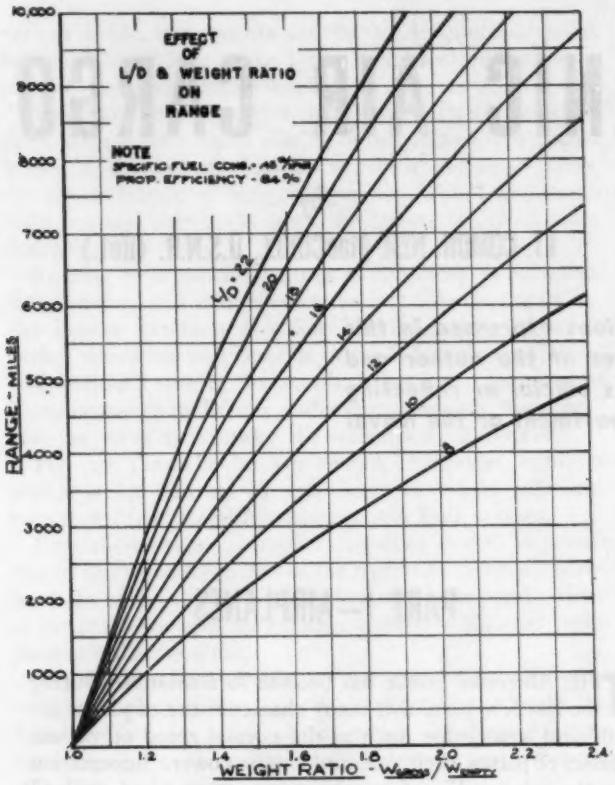


■ Fig. 1

of any sort, nor even a crew or armament, unless they are included in the empty weight of the plane.

Data on a large number of planes indicate an increase of ultimate range with size, but in general there is only a moderate gain with planes over 40,000 lb gross weight. The unwary might conclude that a curve drawn through such data could be extrapolated for future planes of far greater size, and that the larger the plane, the longer its range, as size is increased indefinitely. There may be a natural tendency for such thinking from an unconscious association with the advantages in building naval vessels in continually increasing size; but, the physical conditions are utterly different, and no such parallel should be assumed.

Fortunately, it is not difficult to lay down a few fundamental conditions on which the flying range of a plane depends, so that they may be examined individually to see to what extent size or speed may be expected to affect it.

Starting with the simple relation between power, force, and speed, and using terminology customary in airplane practice, we have:

$$\text{Thrust Horsepower} = \frac{(\text{Speed in mph}) \times (\text{Drag in lb})}{375} \qquad (1)$$

Substituting the ratio, lift/drag, which will be indicated hereafter as L/D:

$$\text{Thrust Horsepower} = \frac{(\text{Speed}) \times (\text{Lift})}{375\ L/D} \qquad (2)$$

Obviously, the term lift represents the weight of the plane complete with fuel and cargo, and the power indicated is the power required to keep the plane in flight, or the power developed by the propeller thrust. The power supplied to the propellers, that is, the engine power, will be greater by a factor representing the propeller efficiency. This is, in general, not far from 84%, assuming that it will be possible to provide a propeller of suitable pitch for any flying condition encountered. Equation (2) may, therefore, be rewritten:

$$\text{Weight} = \frac{315 \times L/D \times \text{Hp}}{\text{Speed}} \qquad (3)$$

or

$$\frac{\text{Weight}}{\text{Hp}} = \frac{315\ L/D}{\text{Speed}} = \text{Power Loading} \qquad (4)$$

Next, assuming the size of engine and propellers adjusted to the power required at each operating condition under consideration, the fuel (and oil) consumption of the engines will be roughly 0.45 lb per hp per hr, and may be expressed:

$$\frac{\text{Lb Fuel}}{\text{Mile}} = \frac{\text{Weight}}{700\ L/D} \qquad (5)$$

These simple general equations are based on fundamentals, and are applicable to any plane at any altitude. The approximations made for convenience, may be altered where more exact results are desired. Thus, the constant in equation (5), is:

$$700 = \frac{375 \times (\text{Propeller Efficiency})}{\text{Specific Fuel Consumption in lb per hp per hr}}$$

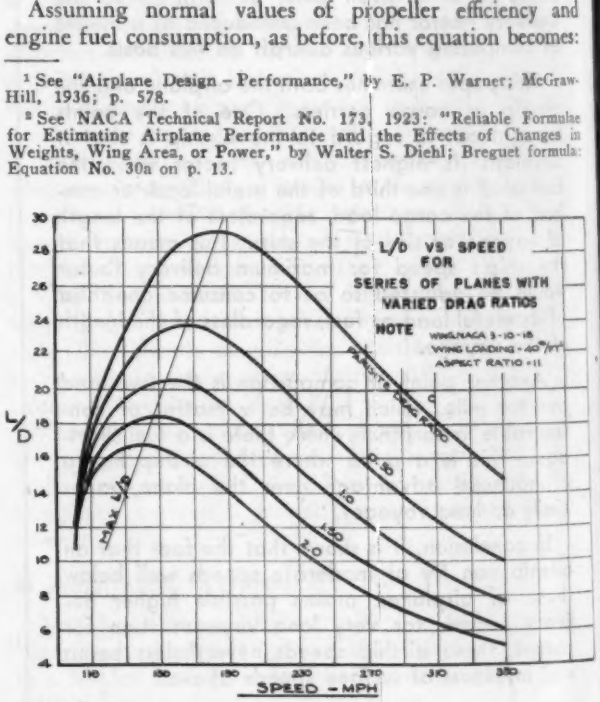
These equations have many applications in the prediction of airplane performance. For instance, the actual range of a plane may be determined by integrating the area under a curve plotted between the miles per pound of fuel, and the gross plane weight as it changes during flight.[1] It may also be calculated directly from the equation:[2]

$$\text{Range in miles} = \frac{863.5 \times \left(\frac{\text{Propeller}}{\text{Efficiency}}\right) \times L/D \times \text{Log}\left(\frac{\text{Gross Weight}}{\text{Empty Weight}}\right)}{\text{Fuel Consumption in lb per hp per hr}} \qquad (6)$$

Assuming normal values of propeller efficiency and engine fuel consumption, as before, this equation becomes:

[1] See "Airplane Design – Performance," by E. P. Warner; McGraw-Hill, 1936; p. 578.

[2] See NACA Technical Report No. 173, 1923: "Reliable Formulae for Estimating Airplane Performance and the Effects of Changes in Weights, Wing Area, or Power," by Walter S. Diehl; Breguet formula: Equation No. 30a on p. 13.



■ Fig. 2

$$\text{Ultimate Range} = 1610\, L/D \times \text{Log}\left(\frac{\text{Gross Weight}}{\text{Empty Weight}}\right) \quad (7)$$

or

$$L/D = \frac{\text{Range}}{1610 \times \text{Log}\left(\frac{\text{Gross Weight}}{\text{Empty Weight}}\right)} \quad (8)$$

For convenience in using this equation, Fig. 1 has been prepared. Either this figure or equation (8) provides a simple means of determining the effective L/D at which a flight is made by a plane, from the total fuel required for a given voyage. The weight ratio would be the ratio of the plane weight before and after the voyage in question. The ultimate range is then determined by the same equation or figure, by use of the L/D just obtained, and the ratio of gross to empty weights of the plane.

Since this equation incorporates all factors which control the ultimate range of a plane, a separate study is considered justified of the two principal factors: (a) weight ratio, and (b) aerodynamic ratio, L/D.

$$\text{WEIGHT RATIO} = \left(\frac{\text{Gross Weight}}{\text{Empty Weight}}\right)$$

Plotting the ratio of gross to net weight for a large number of planes, against their gross weight, reveals a gradual increase in this ratio from about 1.4 for planes under 10,000 lb to about 1.8 for planes over 50,000 lb gross. The curves appear to flatten out for larger planes, with no clear indication of any further rise as the size is increased. The points are surprisingly consistent, considering all the variables involved. For instance, most of the planes in the group over 50,000 lb gross have values lying between 1.65 and 2.0, a variation of only 10% from the mean. Only a few are claimed to have the higher value, and it seems to be about the practical limit in present-day design.

The flatness of this curve for planes over 50,000 lb gross is of sufficient interest to warrant comment. It is a fact that the wing loading, that is, the weight of plane structures per square foot of wing area, has been increasing with gross weight. This has been in spite of changes in design by which loads are better distributed in the largest planes. The four-engine plane has been a factor in this direction, and the six-engine plane will carry it further. Disposition of fuel tanks in the wings, and stowage of cargo there also, are further examples of this expedient. Another factor which has helped has been the smaller factors of safety required by large planes for maneuvering stresses. They are not required to pull out of dives and make changes in direction as rapidly as the smaller and more maneuverable fighters.

The fact that the ratio of gross to empty weight has been held substantially constant for the larger planes is due to increased wing loading, by which the gross weight carried per square foot of wing area has been increased at about the same rate as the unit weight of the plane structure. It must not be overlooked that in doing so, the take-off speed of the planes has been increased. This seems to be acceptable for military service, but it is still to be determined if such increased speeds as are required for the large planes associated with long-range military operations, will not unduly increase the hazards of commercial operations.

From this examination, the following may be concluded in regard to the ratio of gross to empty weight, which is one of the two factors in determining the ultimate range of a plane:

1. At the present state of the art the best value obtained is about 2.

2. There is no apparent trend toward a higher value from increasing the size of planes on similar lines.

3. Increasing the size of a plane may be expected to be accompanied by increasing take-off speeds.

Since the present limit of the weight ratio appears to be about 2, this value will be used throughout this paper, and formula (7) then becomes:

$$\text{Ultimate Range} = 485\, L/D \quad (9)$$

It should not be overlooked that but few planes have yet been built with this high ratio, and it is, therefore, still somewhat in the future for airplanes in general.

AERODYNAMIC RATIO = L/D

The second principal factor in the range formula (7) is the aerodynamic ratio L/D, in which the drag is made up of both the profile drag[3] of the wing and the parasitic drag of the fuselage, tail surfaces, nacelles, and so on, and the induced drag of the wing, resulting from its angle of incidence. Plots of these L/D ratios for a large group of planes, indicate a small increase with size, which varies approximately as the 0.14 power of the gross weight. Thus, doubling the gross weight would increase the L/D about 10%. This is almost the same increase which would be expected from the corresponding Reynolds numbers.[4, 5] In practice, it is difficult to secure this advantage because small irregularities or waviness in the outer surface can upset the transition from streamline to turbulent flow. (Tests on a typical plane indicate that the parasitic drag may be increased as much as 50% by constructional irregularities.[6]) Another element is introduced when the fuselage and nacelles are not increased in proportion to the size of the wings, and their parasitic drag is relatively smaller. This is particularly true of bombers and planes designed for loads of high density. Planes designed primarily as cargo carriers, require much more space for their loads and thus their fuselages may be expected to be increased in size to such an extent as to preclude a reduction in their share of the total drag. Illustrations of the Curtis Commando plane (Army C-46, Navy R5C-1) emphasize the large fuselages typical of cargo planes.

An examination of the effect of parasitic drag on airplane performance is indicated in Fig. 2. Here are shown the variations of L/D with sea-level speed for a series of airplanes having several ratios of parasitic to profile drag, a wing loading of 40 lb per sq ft, and an aspect ratio of 11. They are all derived from test data on the NACA 3-10-18 tapered wing.[7] This family of curves may represent the characteristics of almost any modern plane. While it is plotted primarily for an average wing loading of 40 lb per sq ft at sea level, other wing loadings or altitudes are equivalent to multiplying the speed scale by a constant:

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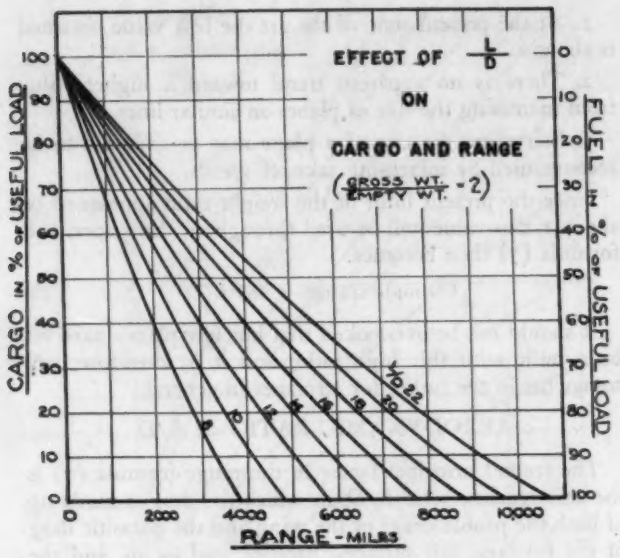
[3] Approximately the minimum drag of the wing at the angle of incidence for zero lift.

[4] See NACA Technical Report No. 627, 1938: "The Experimental and Calculated Characteristics of 22 Tapered Wings," by Raymond F. Anderson; p. 12.

[5] See NACA Technical Report No. 586, 1937: "Airfoil Section Characteristics as Affected by Variations of the Reynolds Number," by Eastman N. Jacobs and Albert Sherman; p. 30.

[6] See NACA Technical Report No. 667, 1939: "Determination of the Profile Drag of an Airplane Wing in Flight at High Reynolds Numbers," by Joseph Bicknell; p. 5.

[7] See NACA Technical Report No. 627, 1938: "The Experimental and Calculated Characteristics of 22 Tapered Wings," by Raymond F. Anderson; p. 7.



■ Fig. 3

$$K = \sqrt{\frac{\text{Wing Loading}}{40} \times \frac{\text{Air Density at Sea Level}}{\text{Air Density at Altitude}}}$$

Notice the straight-line increase of speed for maximum L/D as the drag approaches the bare wing. Since all the planes covered by these curves have the same landing speed, about 100 mph without flaps and approximately 85 mph with flaps, this increase of speed at maximum L/D represents an increase in speed range.. While high speed, of course, entails a corresponding increase in power, the fuel consumed per mile will not be changed, assuming no change in L/D or powerplant efficiency.

In the foregoing, the ranges referred to have been maxima *with no payload,* and are, therefore, only of academic interest. Any cargo carried must necessarily reduce the fuel load, and thus the range. An examination of this effect is indicated in Fig. 3, which has been prepared[8] from Fig. 1, for planes having a weight ratio of 2. For simplicity, the cargo load may be assumed as gasoline. Starting with a useful load of 100% of gasoline, the per cent of this load remaining as cargo, at the end of any length of voyage, may be read directly. For instance, the plane with a drag ratio of 1.5 in Fig. 2, having a maximum L/D of 18.3, would be able to deliver 47% of its useful load as cargo at its destination after covering 4000 miles. If it were necessary to carry fuel for a round trip, as it might be in military operations, this range would be equivalent to a cruising radius of only 2000 miles.

The foregoing study has been concentrated on range, and payload *per voyage.* This is, of course, vital when a desired amount of cargo must be delivered by only one trip of a plane. But when repeated trips are required, average flying speed affects the number of trips which can be flown in a given time, and it will be found that there is some critical speed for each length of voyage at which the greatest total delivery of cargo can be made in a month or any other time period. Too high a speed will require too much of the useful load to be given up to fuel for the trip, even though more trips can be flown. Too low a speed will reduce the number of trips flown more than can be gained by the increased cargo carried per trip.

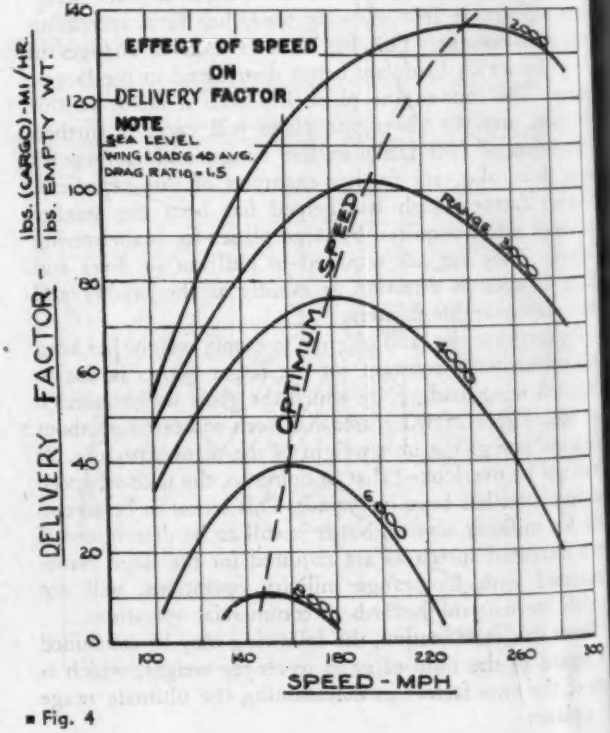
A factor indicating this effect of speed on the quantity of cargo delivered in a given time has been called a Delivery Factor in this paper. This factor is of interest for military cargo, as a measure of the amount of cargo which can be delivered in a given time by a given quantity of critical material used in the plane's structure. In times of peace it would be of interest as an indicator of the operating conditions which will deliver the maximum quantity of cargo, over which fixed charges may be distributed for overhead expenses, amortization, interest, depreciation, and so on. As used here this factor represents the number of pounds of cargo delivered per hour, over an extended period, per pound of plane's empty weight.[9] It is the product of three elements: (1) the ratio of useful load to empty weight of the plane; (2) the per cent of useful load carried as cargo; and (3) the plane's speed. Such delivery factors are plotted in Fig. 4, where the speeds used were taken from Fig. 2 for the plane with a drag ratio of 1.5, at various L/D values. These corresponding L/D values were in turn converted into per cent of cargo carried, from Fig. 3.

It is apparent that there is an optimum speed which results in a maximum total cargo-handling ability for each length of voyage. Notice how rapidly the delivery factor falls off for voyages on either side of the optimum. This is an important fact not often realized.

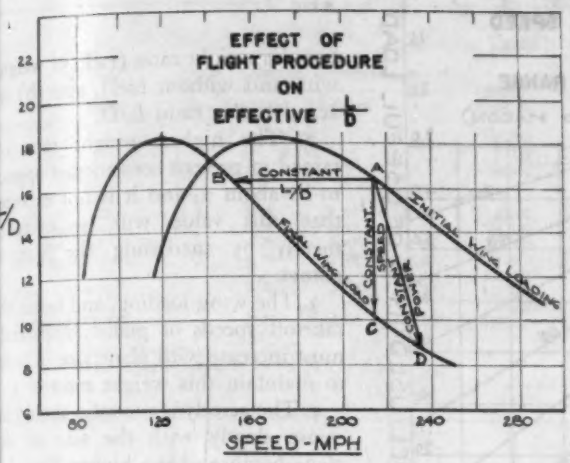
The dotted line connecting the peak values of these curves indicates the increase of the optimum speed with decrease in length of voyage. It starts at the ultimate range where the delivery factor is zero, and the speed corresponds to the maximum L/D for the plane under consideration (Fig. 2). As the voyage length is decreased, the optimum speed increases until it becomes equal to the maximum speed of the plane with the power installed. The maximum delivery factor occurs at zero range where the cargo is

[8] See Appendix I for the method of construction.

[9] A corresponding term used in other studies: ton-miles of cargo per month per pound of empty weight, based on 2000 hr. of flying per year, is approximately one-twelfth as large.



■ Fig. 4



■ Fig. 5

100% of the useful load, and the speed is the maximum for the plane.

An example of the use of the delivery factor is shown for the delivery by a plane carrying a useful load of cargo and fuel of 28 tons, over a 3000-mile voyage, and flying one-half the time for a month, or an average of 12 hr per day. The values of the aerodynamic ratio L/D for this plane are assumed to be equivalent to those of the plane of Fig. 2 having a 1.5 drag ratio. The computations are given in Table 1.

Table 1 – Cargo Delivery as Affected by Speed
3000-Mile Voyage

Average Flying Speed, mph	Cargo Delivered per Trip, tons	Number of Trips per Month	Cargo Delivered per Month, tons	Delivery Factor
120	15.5	14.4	223	65
160	16.2	19.2	311	93
200	14.3	24.0	343	102
240	10.9	28.8	305	93
300	5.05	36.0	182	55

Where the largest pay cargo per trip would be carried at 160 mph, the maximum delivery of cargo per month would be carried at 200 mph. Note that the maximum delivery factor indicates the speed for maximum cargo delivery per month.

At this point it is necessary to digress a little to examine various changes in flight procedure which may be used to compensate for the decreasing weight of a plane as its fuel load is consumed during a voyage. If flight is continued at the initial speed and angle of incidence for level flight, the plane would climb at an increasing rate as the fuel weight diminishes. There are several procedures open to a pilot:

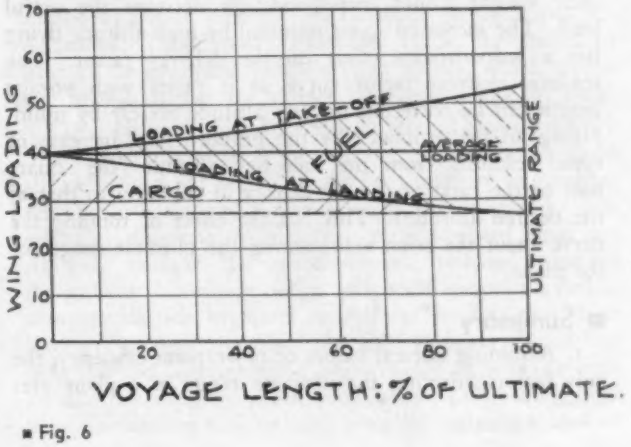
1. He may fly at *constant L/D*, maintaining a constant angle of incidence, and decreasing his speed and power to hold a fixed altitude. The speed at the end of the flight would be about 71% of the initial, resulting in an average speed of about 84% of the initial – if the initial fuel load equalled the net weight of the plane (2:1 gross to empty weight ratio). The final power, consequently, must be reduced to about 36% of the initial. The process is indicated in Fig. 5, where L/D curves are shown for the plane of Fig. 2 for a wing loading with the initial fuel load, and a loading half as great corresponding to the end of its voyage with no fuel. If the initial flight had started at point A, the flight conditions would have followed along the horizontal line from A to B.

2. The pilot may fly at *constant speed*, but decrease his angle of incidence and power accordingly. This would cause a steadily decreasing L/D value, as indicated by the vertical line from A to C, which would result in a shorter range, but a flying speed about 18% higher. The final power in this example would be 80% of the initial, but the range would be only about 81% of that of procedure No. 1.

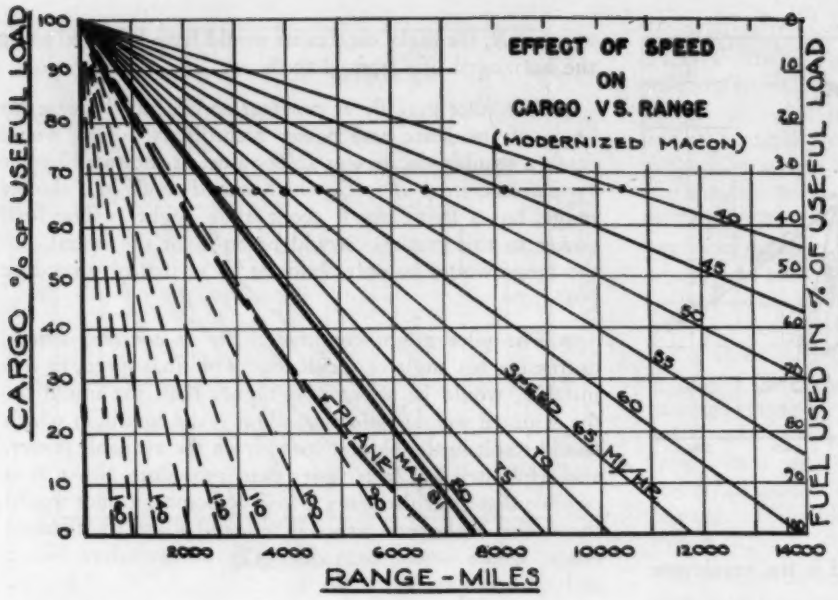
3. The pilot might continue to fly at *constant power*, decreasing his angle of incidence. The final speed, in this instance, would be about 8% higher than the initial for the assumed weight ratio of 2. This is the condition which would result in the highest speed from the available power, and while it is but little faster than procedure No. 2, it is possible that after dropping a load of bombs, a pilot would choose this high-speed return if he could afford the reduced range, which would be about 95% of procedure No. 2 and 77% of procedure No. 1.

4. The pilot might maintain *maximum L/D*, and hold the *initial speed*, angle of *incidence*, and *power* throughout a voyage, and allow the plane to climb to high altitude as the fuel is consumed. A plane just able to maintain sea-level flight with full load, would maintain identical speed and L/D value if the altitude varied so that the air density will fall off at the same rate as the plane weight. Thus, at the end of a maximum-range flight, the plane would be at about 22,000 ft, provided the engines were so supercharged as to develop the same power as at sea level. Where high mountains must be crossed near the end of a voyage, this is a promising procedure.

In this paper it has been assumed, for convenience, that flight would be carried out under procedure No. 1. Under this condition the average speed for the whole voyage will be about the average of the speeds corresponding to the wing loadings at the beginning and end of the voyage. This would be about 85% of the initial speed, where the weight ratio is 2. The plane loadings would be as indicated in Fig. 6, for voyages of different lengths, but all with an average wing loading of 40 lb per sq ft. For the ultimate range, where the initial load is twice the final load, as has been assumed previously in this paper, flight would start at a wing loading of about 53 lb per sq ft and



■ Fig. 6



■ Fig. 7

end at 28 lb per sq ft, with no cargo. For progressively shorter voyages with progressively larger cargo loadings the initial loading would be less and the final loading would be greater until they both become 40 lb per sq ft at zero voyage.

It will be recognized that the take-off and landing speeds will follow a relationship similar to the wing loadings: The two would average about 85 mph (with flaps) for voyages of any length, but the take-off speed will increase and the landing speed decrease on voyages of increasingly greater length until the ultimate range would have a take-off speed of about 101 mph (119 without flaps) and a landing speed at the end of the voyage of about 72 mph.

■ Effect of Altitude

It was noted previously that equation (7), which defines the range of a plane, does not contain a factor for altitude. If a plane flies at high altitude, the speed must be increased as the air density is decreased, but for a given L/D the drag will remain constant if the weight is not changed. The power will, of course, increase at the same rate as the speed, and, with equal powerplant efficiency, the miles per pound of fuel will not be changed. Thus, the altitude at which a plane flies will have no inherent effect on range, unless the desired altitude is so high that larger engines must be installed than would be required for take-off, and their weight would correspondingly decrease the useful load. The increased speed required by high-altitude flying has a proportionate effect on the delivery factor. The sea-level delivery factor curve as it varies with voyage length can be corrected for any altitude merely by multiplying all the ordinates by the proportionate increase of speed resulting from the increased altitude (the square root of the ratio of the air density at sea level to that at the desired altitude). This has the effect of rotating the curve about the point representing the ultimate range for the plane.[10]

■ Summary

1. Assuming normal values of powerplant efficiency, the only factors affecting the ultimate range of a plane are: (a) the weight ratio (ratio of weights with and without fuel), and (b) the aerodynamic ratio L/D.

2. The highest weight ratio obtained in present construction appears to be about 2, and it is not expected that this value will be exceeded merely by increasing the size of planes.

3. The wing loading, and hence the take-off speeds of planes, apparently must increase with plane size in order to maintain this weight ratio of 2.

4. The aerodynamic ratio should increase slowly with the size of the plane because of the higher Reynolds numbers.

5. The delivery factor, which indicates the quantity of cargo over which the overhead costs of operation must be distributed, includes the effect of speed. The highest delivery factor for a plane occurs at a speed depending on the length of voyage. The longer the voyage, the lower this speed.

6. Altitude, while having no effect on ultimate range, does affect the delivery factor; the shorter the range, the greater the effect. Conversely, the longer the range, the less the effect, diminishing to zero effect at the ultimate range.

PART II—AIRSHIPS

The static lift of an airship is a function of the volume of air displaced by the lifting gas. For helium gas, at sea level and 65 F, it may be expressed as:

$$\text{Lift (lb)} = 62{,}000 \left(\frac{\text{Vol}_G}{10^6}\right) \qquad (9)$$

where $\left(\frac{\text{Vol}_G}{10^6}\right)$ is the gas volume in millions of cubic feet. (The constant, 62,000, varies with temperature, being 69,400 at 0 F; 65,000 at 30 F; 56,000 at 100 F. When the gas has "superheat," that is, it is warmer than the surrounding air, it has additional buoyancy.) In addition to the static lift, an airship may have dynamic lift similar to that of an airplane, when flying with its axis inclined to the line of flight. While the aspect ratio is extremely low, the lift at small angles of incidence is secured with a very small increase in drag, and the resulting L/D ratio may be of considerable value, especially for loads which may be brought up to the ship in flight by hook-on airplanes. It seems practical to fly an airship with a dynamic lift of at least 6% of its static lift.

The drag of an airship is normally expressed in terms of the volume of its envelope, which is related to the gas volume by a factor r, which averages about 0.88 for a number of rigid airships as may be seen in Table 2. This is of course only normal design practice and may be varied at will by the designer.

The equation for drag at sea level is:[13]

$$\text{Drag (lb)} = 25.6\, C_D \left(\frac{\text{Vol}_E}{10^6}\right)^{\frac{2}{3}} V^2 \qquad (10)$$

[10] See Fig. 12.

Table 2 – Airship Data

Airship	Volume, millions of cu ft: Envelope	Helium[a]	r	Useful Lead, lb	Dimensions, ft (approximate): L	D	Empty Weight, lb	Useful Load: Empty Weight	Gross Lift 62 lb per M	Normal Power Available	Hp per Ton of Useful Load
Non-Rigid											
L	0.123			2,200	147.5	40	5,420	0.406	7,620	290	264
G	0.197			3,320	190	45	8,980	0.370	12,200	652	393
MC-2 (Metal Clad)	0.200			3,900	150	53	8,700	0.448	12,400	400	205
Rigid											
Bodensee LZ-120	0.80	0.67	0.84	12,800	426	62	28,700	0.446	41,500	1040	163
Shenandoah ZR-1	2.38	2.01	0.87	44,100	680	79	80,400	0.548	124,500	1610	73
Los Angeles, LZ-126, ZR-3	2.76	2.47	0.89	60,000	680	90	93,000	0.645	153,000	2200	73
Graf Zeppelin LZ-127	4.10	3.70	0.90	71,000 100,000[b]	776	100	158,000	0.450 0.633[b]	229,000	2750	78
Macon ZRS-5	7.40	6.50	0.878	160,600 190,200[b]	785	133	242,400 212,800[b]	0.662 0.890	403,000	4480	58 47
Hindenburg LZ-129	7.65	6.80	0.89	186,300	804	135	235,700	0.790	422,000	3200	34

[a] 95% maximum for rigids.
[b] Corrected for modern engines.
[c] Hydrogen.

where C_D is the drag coefficient, $\left(\frac{Vo'_E}{10^6}\right)$ is the envelope volume in millions of cubic feet, and V is the speed in miles per hour. The drag coefficient used here is different from the coefficient normally used in airplane design, because it is more convenient to deal directly with volumes in airship computations. The term $C_D\left(\text{Vol}\right)^{\frac{2}{3}}$ is sometimes referred to as "drag area" in airship work.

Since Thrust Hp $= \frac{\text{(Drag) (Speed)}}{375}$, from equation (1),

$$\text{Engine Hp} = \frac{C_D\left(\frac{\text{Vol}_E}{10^6}\right)^{\frac{2}{3}} V^3}{14.5\eta} \tag{11}$$

where η is propeller efficiency.

Fuel consumption in lb per mile, w, is accordingly:

$$w = \frac{C_D\left(\frac{\text{Vol}_E}{10^6}\right)^{\frac{2}{3}} V^2 c}{14.52\eta} \tag{12}$$

where c is the specific fuel and oil consumption, in lb per hp per hr. Assuming normal values as before for specific fuel and oil consumption of 0.45 lb per hp per hr, a propeller efficiency of only 73% because of the lower ratio of ship's speed to propeller tip speed (V/ND) in airships than in planes; and a drag coefficient[11, 12, 13, 14] of 0.0236, equation (12) becomes:

$$w = \frac{\left(\frac{\text{Vol}_E}{10^6}\right)^{\frac{2}{3}} V^2}{1010} = \frac{V^2}{K} \tag{13}$$

where K is a constant for any particular ship.

With these general equations as guides, the performance will now be worked up of the airship ZRS-5, the U.S.S. Macon. This ship had an envelope volume of 7,400,000 cu ft, which, from equation (13), had a fuel consumption which may be expressed as:

$$w = \frac{V^2}{266} \tag{14}$$

Similarly, formula (11) for this ship becomes:

$$\text{Engine Hp} = \frac{V^3}{118} \tag{15}$$

The gross lift was 403,000 lb, and the empty weight was 242,400 lb, which included powerplants of 4480 hp which weighed 11.1 lb per hp, exclusive of the water recovery system. These engines were obviously far more conservatively rated than is the present practice in long-range airplane design, where in large installations they weigh not more than 4.5 lb per hp, at their cruising output (one-half take-off power), including propellers, cowling, oil supply, and so on. It seems justifiable to assume that the same engines could drive an airship as well as a long-range plane with comparable performance and reliability. Therefore, for an airship built today, and similar in size to the Macon, the empty weight could be reduced about 29,600 lb by this reduction in weight from the use of modern powerplants. The empty weight would then be 212,800 lb, and thus the useful load would be 190,200 lb.

If all this load were given up to fuel, as was done in the simplified analysis of airplane performance in Part I, the ultimate range of the airship would thus be:

$$R_a = \frac{\text{Total Fuel Load}}{\text{Fuel Consumed per Mile}} = \frac{190{,}100 \times 266}{V^2} = \frac{50{,}700{,}000}{V^2} \tag{16}$$

Values for ultimate range obtained from this formula are shown for various speeds in Table 3, together with the engine power required, as computed from formula (15). For speeds above the maximum of 81 mph with the 4480 hp installed, reductions in the useful load are made for the additional power required, on the basis of 4.5 lb per hp. Additional deductions should be made for increased strength of structure for these high speeds, which would still further reduce the ranges for speeds above 81 mph.

These ultimate ranges have been plotted against the per cent of fuel aboard after various voyages, in Fig. 7. Since all useful load has been assumed for simplicity as gasoline, this figure is somewhat similar to Fig. 3 of Part I. The principal difference is that the constant-speed lines are straight, because the air resistance remains constant throughout the voyage when weight of consumed fuel is compensated for by water recovery. The fact that these lines are straight simplifies plotting. A heavy dotted line indicates the corresponding performance of a plane with an ultimate range of about 6800 miles. This requirement might be met by a plane with a weight ratio of 2, and an

[11] See NACA Technical Report No. 291, 1928: "Drag of C-Class Airship Hulls of Various Fineness Ratios," by A. F. Zahm, R. H. Smith, and F. A. Louden.

[12] See NACA Technical Report No. 318, 1929: "Speed and Deceleration Trials of U. S. S. Los Angeles, September, 1927," by S. S. De France and C. P. Burgess.

[13] See NACA Technical Report No. 394, 1931: "Airship Model Tests in the Variable-Density Wind Tunnel," by Ira H. Abbott.

[14] See NACA Technical Report No. 397, 1931: "The Drag Characteristics of Several Airships Determined by Deceleration Tests," by F. L. Thompson and H. W. Kirschbaum.

Table 3 – Power and Ultimate Range – Modernized Macon at Various Speeds

Speed, mph	Hp	Powerplant Weight at 4.5 lb per Hp	Useful Load, lb	Hp per Ton Useful Load	Ultimate Range, statute miles
140a	23,250	104,600	105,800	439	1,420
130a	19,620	88,300	122,100	321	1,910
120a	14,650	65,900	144,500	203	2,640
110a	11,300	50,800	159,600	142	3,490
100a	8,475	38,200	172,200	99	4,550
90a	6,180	27,800	182,600	68	5,950
81	4,480	20,200	190,200	47.2	7,600
80	4,330	20,200	190,200	45.6	7,800
70	2,900	20,200	190,200	30.5	10,200
60	1,830	20,200	190,200	19.3	13,900
50	1,060	20,200	190,200	11.2	20,000
40	542	20,200	190,200	5.7	31,200
30	229	20,200	190,200	2.4	55,600

a No allowance made for addition structural weight required by speeds higher than 81 mph.

L/D of 14, such as shown in Fig. 3, or an L/D of 18.3 with a weight ratio of 1.83, and so on.

This figure illustrates several important facts about airships:

1. The per cent of the useful load which can be delivered at any range increases rapidly as the speed is reduced.

2. The superior load-carrying characteristics of the airship over long ranges are due to the lower power and thus lower fuel consumption required per unit of cargo, because of its lower speed. The power required per ton of cargo in airplanes may be more than five times as great as required by the airship.

3. While speeds even higher than most economical for a plane can be secured by an airship, they are obviously impractical. Compare the 150-mph line for the airship with the airplane indicated, which according to Fig. 2 would fly at about this speed if at maximum L/D. These curves indicate clearly that the fields of utility for the airship and airplane are distinctly separate. High speed is the province of the plane, just as long-range and high percentage cargo loads at medium speeds are the province of the airship. Any attempt of airships to compete with a plane on the basis of speed would sacrifice their inherent superiority when operating at their optimum speeds, or some intermediate speed such as typified by the present-day practice of limiting engine power to that which will provide a top speed not in excess of about 100 mph.

4. A higher percentage of the useful load can be delivered by the airship than by a plane, even on ranges within the practical limits of the plane. On a 3000-mile voyage, the plane under consideration would consume over 50% of its total useful load as fuel for its engines, while the airship would consume only a little over 20% at a cruising speed of 60 mph. Thus for each ton of fuel, the plane could deliver 1 ton of cargo, while the airship could deliver about 4 tons. As the distance is increased, the difference becomes more noticeable. For a 5000-mile voyage, which is only a 2500-mile round trip, the airship could deliver more than seven times as much cargo per ton of fuel as the airplane indicated.

It should not be overlooked that these data are taken from the actual performance of an experimental airship built ten years ago.

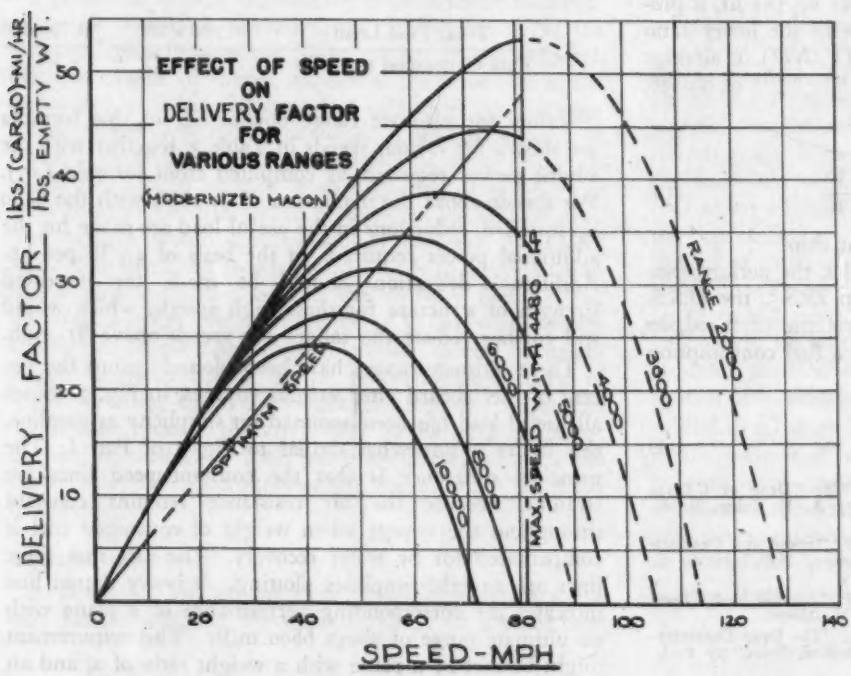
■ Effect of Speed

When time for the delivery of a given quantity of cargo is of vital importance, speed enters the picture as it did in the case of the airplane. The higher the speed, the more trips that can be made in a given time.

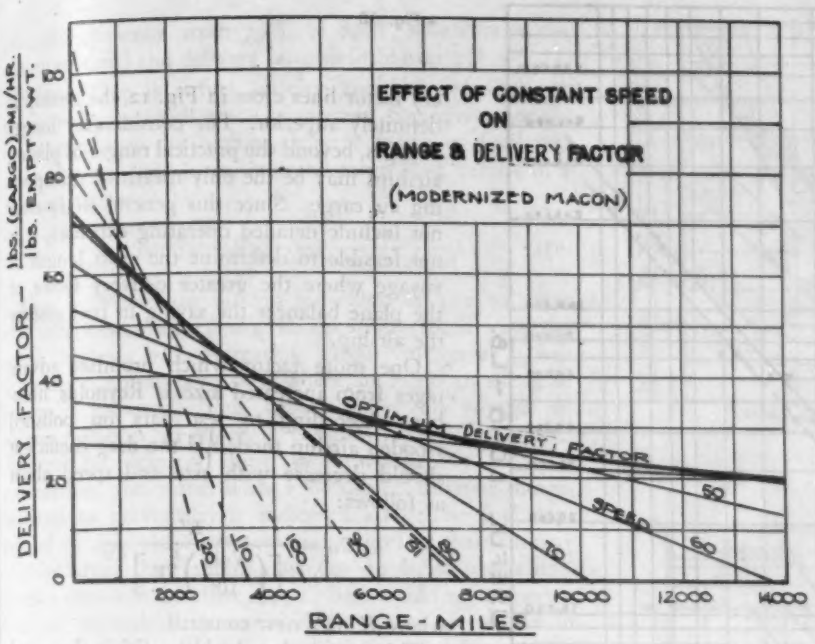
Curves showing the effects of speed and range on the term delivery factor as used in Part I are given in Fig. 8, which is similar to Fig. 4 of Part I. The delivery factors for these two types of aircraft are, however, not necessarily directly comparable, as will be discussed further on. The ratio of useful load to empty weight in this instance was $\frac{190{,}200}{212{,}800} = 0.895$. The percentages of fuel aboard, item (2) above, were taken from Fig. 7. Notice the clean-cut optimum speed for each length voyage.

Possibly a more useful plot of the same data is given in Fig. 9, which shows the effect of departure from the optimum speed, where conditions may warrant so doing. This family of straight lines was plotted between points representing the ultimate range from Table 3, laid off on the horizontal scale, and points laid out on the vertical scale representing the product of speed and ratio of useful load to empty weight.

The envelope of all these straight lines is a curve indicating the maximum delivery factor for any range, and the points of tangency indicate the optimum speed at each range. These curves may be obtained by formula as well as by graphical means, as indicated in the Appendix. By substituting data for the modernized Macon, in equations (K), (F), and (H) in the Appen-



■ Fig. 8



■ Fig. 9

dix, the following are obtainable:

$$\text{Optimum Delivery Factor} = \frac{2450}{\sqrt{\text{Voyage}}} \qquad (17)$$

$$\text{Optimum Speed} = \frac{4100}{\sqrt{\text{Voyage}}} \qquad (18)$$

where speed and voyage are expressed in miles per hour and statute miles, respectively.

An interesting fact is revealed when the voyages and speeds are indicated on Fig. 7 at which the delivery factors are maximum. These points are shown by small circles. Notice that they occur in all cases where one-third of the useful load has been consumed for fuel. Hence, at optimum speed, the cargo weight will be twice the weight of fuel required for a voyage, regardless of its length. This is proved mathematically in the Appendix [equation (G)], and holds for any airship, regardless of size.

■ Effect of Size

So far, the study of cargo carrying performance of airships has been confined to a ship the size of the Macon. An examination will next be made of the effect of size on performance.

An important point of difference from airplanes, which depend entirely on dynamic force for lift, is the buoyancy of lighter-than-air ships due to the displacement of air. As a result, the effect of increasing size is very different. If the empty weight increases with the cube of linear dimensions, it will increase at the same rate as its gross lift. This means that the ratio of gross to empty weight would remain constant, and that the useful load would increase as the cube of the ship's linear dimensions. At the same time, the power required to drive it at a given speed would increase only with the square of its linear dimensions. Thus, if all the ship's dimensions were doubled, the useful load would be increased eight times, while the power and fuel consumption rate at the same speed would be increased only four times. The range and cargo carried per pound of fuel would thus increase in direct proportion to the linear dimensions, or the cube root of the volume of the ship.

Experience has shown that the conditions are really better than this. As shown in Fig. 10, the useful load increases about as the 1.12 power of the volume quite consistently from small blimps through the largest rigid airships. This means that the useful load per pound of ship's structure actually increases with size, as shown in Fig. 11. It would appear that there is no limit to the increasing efficiency of airships with size, as size may be increased indefinitely, which is apparently not the case with planes.

Since at optimimum speed the fuel required will always be one-third of the useful load, and the pay load may therefore be two-thirds of the useful load, one of the economic advantages of increasing the size of airships is reflected in the increased optimum speed for a given voyage.

The delivery factor is plotted in Fig. 12 for a series of airships differing only in size. Dotted lines indicate the relative performance of an airplane, which may be represented by the plane discussed in Part I, having an ultimate range of 6800 miles, at sea level and at 20,000 ft. In order to make this comparison, the delivery factors for the plane have been corrected as follows:

Delivery factors indicate the quantity of cargo delivered in a given time, by a given size of aircraft, and they are therefore a measure of the cargo quantity over which fixed charges may be distributed. Apparently fixed charges are roughly the same for airplanes and airships. This is indicated by the relative costs available on various aircraft per ton of useful load.

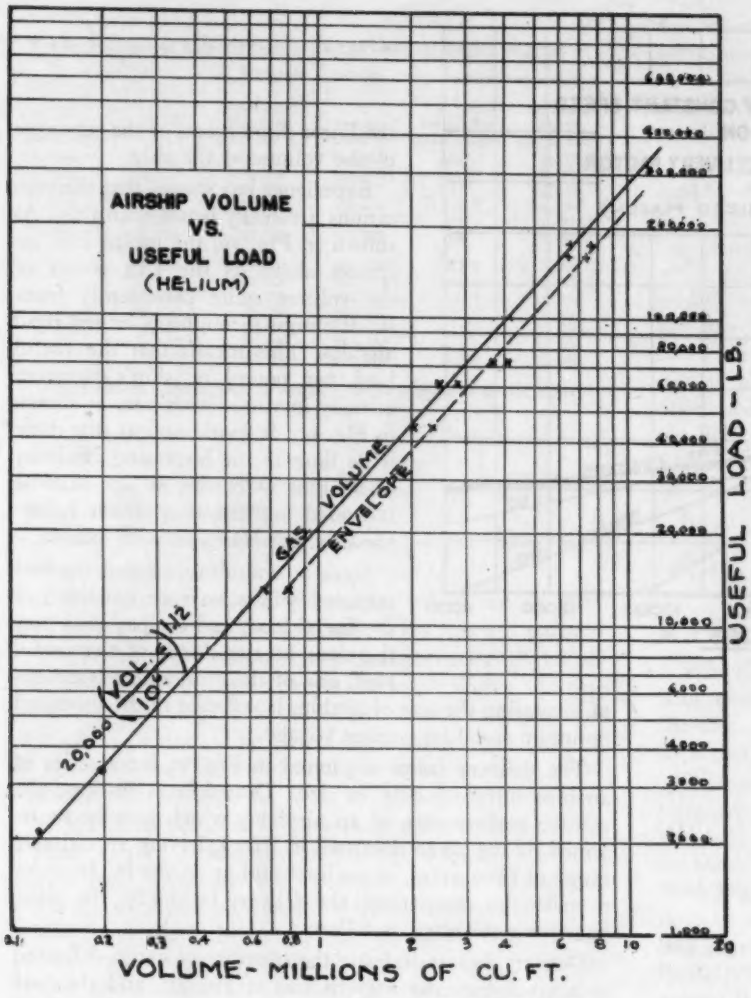
For the Macon at $2,600,000, it was $27,400 per ton.

For the China Clipper at $450,000, it was $26,000 per ton.

For the Douglas DC-4 at $350,000, it was $30,000 per ton.

Hence, the only correction made to the plane data compared to the airship curves in Fig. 12 was for the different effects of layover time at the end of each voyage, which, of course, effects the true average speed for the time interval over which fixed charges are distributed.

It was assumed that the layover time for cargo transfer, crew liberty, engine check, and so on, will be the same for either the airship or the airplane; also, that the plane will make a voyage in one 24-hr day, with a 24-hr total layover, so that its average or effective flying time will be one-half its normal cruising speed. If the airship will make one-half the flying speed of the plane, it will require 48 hr for the same voyage, and with the same 24-hr layover, its average speed will be two-thirds of its cruising speed, or of the plane's effective speed. In this example, the comparative delivery factors of the plane were, therefore, multiplied by two-thirds before comparing with the airship. Every specific comparison should be analyzed in a similar manner,



■ Fig. 10

giving consideration to the actual speeds flown, the time of flight, and the layover time, but the example given appears typical and is, therefore, used in superposing the curve for the plane on the airship curves of Fig. 12.

Fig. 13 shows the corresponding optimum speeds for the airships in Fig. 12. These curves are limited for top speeds, which may be secured by powerplants which are arbitrarily assumed as weighing 10% of the ship's structure, and 4.5 lb per hp. The rapid increase of performance with size is evident, in regard to both the maximum delivery factor and optimum speed. For instance, on a 4000-mile voyage, the delivery factor of a 10,000,000-cu ft ship would be about three times as great as of a 1,000,000-cu ft ship, and the optimum speed would be almost 80% higher.

These curves of airplane and airship performance again show that they are distinctly complementary carriers of cargo, and not rivals. For voyages above which the delivery factor lines cross in Fig. 12, the airship is definitely superior. For considerably longer voyages, beyond the practical ranges of planes, airships may be the only means of transporting air cargo. Since this general study does not include detailed operating expenses, it is not feasible to determine the exact length of voyage where the greater delivery factor of the plane balances the saving in fuel cost by the airship.

One more factor which promises advantages from increased size is Reynolds number. According to test data on polished wooden airship models,[13] the drag coefficient should decrease with size and speed about as follows:

$$C_D = \frac{K'}{\left[\left(\frac{\text{Vol}_E}{10^6}\right)^{\frac{1}{3}} V\right]^{0.15}}$$

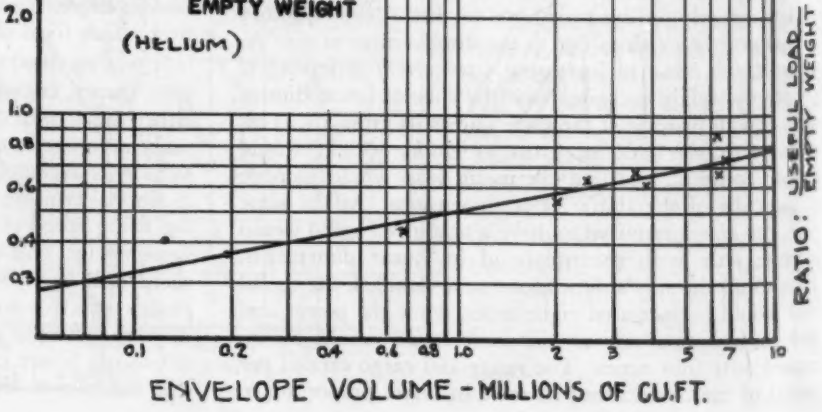
where K' is a new constant.

This means that doubling either the speed or the length of the ship should reduce the drag coefficient to 90% of its original value. Such a gain from Reynolds number is, of course, similar to the gain expected in the decreased parasitic drag of an airplane fuselage, but since small irregularities in surface smoothness may neutralize such a gain, it is discounted here for airships as it probably should be for planes, until experience indicates that it can be realized in practice.

Notice that the equations for both the optimum speed and delivery factor contain the expression:

$$\sqrt{\frac{\text{Propeller Efficiency}}{(\text{Drag Coefficient})\ (\text{Specific Fuel Consumption})}}$$

From this can be seen the effect of changes of any or all these factors. For example, a 15% reduction either in the fuel consumption, from 0.45 to 0.38 lb per hp-hr; or drag coefficient, from 0.0236 to 0.02; or a 15% increase of



■ Fig. 11

propeller efficiency, from 73% to 84%, would raise the optimum speed and delivery factor a little over 7%.

Where there is a fuel scarcity, the fuel required per ton-mile of cargo will prove of interest. Strangely enough, the dictum that the highest delivery factor is developed when the fuel load is one-half the cargo load, results in a relation such as:

$$\text{Lb Fuel per Ton-Mile} = \frac{1000}{\text{Voyage in miles}} \qquad (19)$$

This holds for any airship regardless of size or other characteristics, while flying at optimum speed. As the speed increases with decreasing length voyages as it will if optimum, a limit is reached where the full normal engine power is required. This cut-off where all voyages of shorter distances are flown at constant speed is, of course, different for different ships, as shown in Fig. 14. For comparison, the plane with a 6800-mile ultimate range, referred to previously, is indicated also. The fuel consumed by this plane per ton-mile is greater than for any airship larger than about 2,000,000 cu ft, at any range. Even a 9000-mile ultimate range plane would require more fuel per ton-mile than any airship the size of the Macon or larger. While the difference is not great for voyages less than 2500 miles, it becomes increasingly marked on longer voyages – the fuel required by the plane *increases* exponentially, and becomes infinite at its ultimate range, while the fuel required by the airship *decreases* steadily as the optimum speed diminishes. The curve illustrates clearly the great advantages possessed by the airship because of its ability to fly at lower, more economical speeds than the airplane.

■ Future

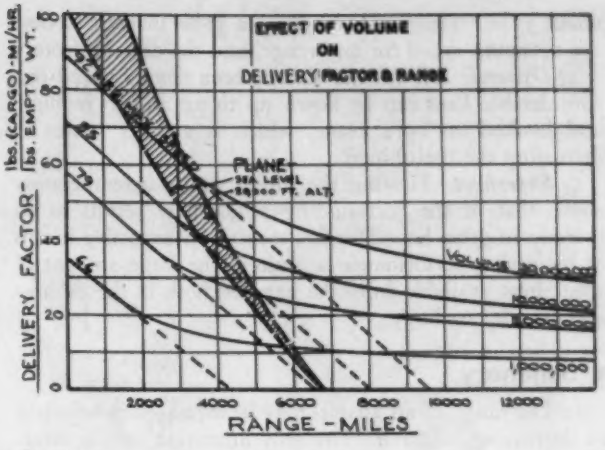
The study of airships which has just been presented has purposely been extremely conservative, as may have been implied from centering it around a ship built over 10 years ago, with no change except the installation of modern airplane engines. This should not give the impression that there are no possibilities for improving their performance. While the number of scientists who have been able to give their time to airship development has been far smaller than the number of those who have caused the prodigious development of airplanes in the interim, several avenues for considerable improvement are open. Among them are the following which are listed briefly:

1. *Stern Propulsion* – A suitably designed propeller at the stern can transform much of the energy in the ship's wake into useful work, as reflected by a considerable increase in propeller efficiency.

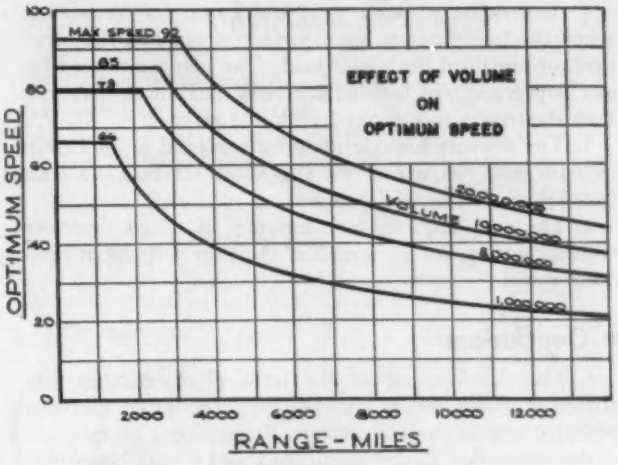
2. *Aerodynamic Drag* – Much can be done to clean up the air resistance, as has been done with airplanes. The elimination of protruding cars and engine nacelles will be in this category. Changes in the hull shape, corresponding to the remarkable developments in reducing the profile drag of airplane wings, may be expected.

These two developments may be sufficient to increase the performance of a conventional 10,000,000-cu ft ship, so as to equal the otherwise equivalent 20,000,000-cu ft ship shown in Figs. 12 and 13.

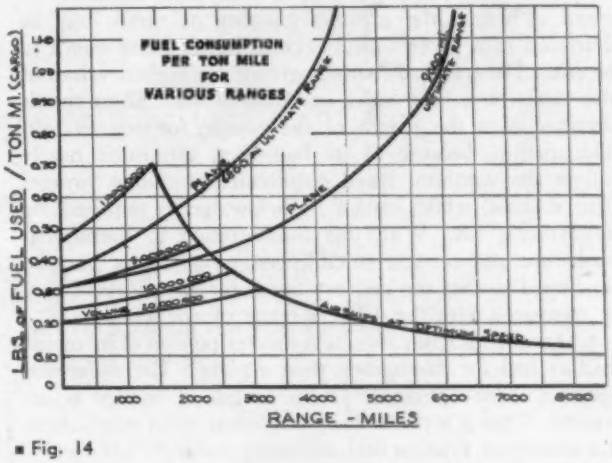
3. *Structural Weight* – There is probably no development which would make such a real improvement in performance as a noticeable reduction in the structural weight of the ship. This is because any reduction in structural weight increases the ratio of useful to empty weight, which directly affects both the optimum speed and earning factor, by simultaneously increasing the numerator and decreasing the denominator. With the new high-strength alloys now available, more accurate methods of calculating stresses and strength, and possibly even entirely different construction, noticeable improvements are possible. For instance, a 15% reduction in empty weight of the Macon would increase the ratio of useful load to empty weight



■ Fig. 12



■ Fig. 13



■ Fig. 14

about 37%. This would result in a 37% increase of both the optimum speed for any range, and the earning factors.

4. *Dynamic Lift* – It has already been demonstrated that considerable load can be flown up to an airship in flight and hooked on by a plane, which is another means of increasing the useful load.

5. *Superheat* – Heating the lifting gas to a temperature above that of the surrounding atmosphere results in an increase in gross lift. This has seemed an attractive means of increasing performance because of the large amount of waste heat available from the engines, both in the exhaust and cylinder cooling.

Summary

1. The range of an airship may be increased indefinitely by decreasing its speed. The only limitation in this direction is the velocity of adverse winds which may be encountered.

2. The delivery factor is a maximum for an airship when the fuel required for a voyage is one-half the cargo load, or one-third the useful load. The optimum speed for any voyage, at which this factor is a maximum, decreases with the square root of the length of voyage.

3. The delivery factor and optimum speed of an airship increase with the size of the ship, both increasing a little faster than the linear dimensions.

4. The fuel required per ton-mile of cargo decreases with increasing voyage length if the ship is flying at optimum speed.

Conclusions

1. The development of the term ultimate range has proved a useful means of evaluating the cargo carrying performances of various aircraft. It facilitates an estimate of the proportion of the useful load which may be carried on any voyage shorter than the ultimate.

2. The term delivery factor, as used here, indicates the speed at which the greatest quantity of cargo may be delivered in a given time interval. It has been shown to be critical to speed, the speeds giving the highest values of this factor have been called optimum speeds. These speeds decrease with the length of the voyage for any aircraft. The airship, because of its buoyancy, can more nearly follow the optimum speed requirement for long voyages than a plane, which cannot fly below speeds required for its dynamic lift. With due consideration to operational procedure and costs, a suitable conversion factor may be developed so that the delivery factor for any airplane may be compared with the delivery factor of any airship.

3. An airship will consume a smaller portion of its useful load as fuel for its engines, than a plane. The difference becomes more noticeable as the length of voyage is increased. This is a particularly important point when there is a scarcity of aviation fuel, as during a war.

4. There is little conflict between the airship and the airplane in their fields of best performance. The airplane has superior delivery factor in the province of high-speed transportation and for short-range hauls where optimum speeds are high. The airship has superior delivery factor on the long voyages where large proportions of the useful load are required for cargo, where the optimum speed is lower than practical for a plane, and where fuel scarcity requires the least fuel to be used for a given cargo load.

5. The question arises if and where long voyages without refueling are necessary. Such instances occur frequently when round-trip flying must be carried on without refueling. Naval scouting operations would be typical of this category. So would the transport of military cargo during war, when it would be obviously a hardship for military commands to give up fuel for the return trip from stocks at remote points. The probabilities are that such stocks would have been accumulated with considerable difficulty. Particular note should be given to Major Richardson's statement[15] that "one-half the supply tonnage required to maintain an overseas Army consists of petroleum products." Considering the time required to deliver gasoline to strategic outposts by slow-moving tankers, it is not difficult to imagine that unexpected events might sometimes occur, which would justify delivery of emergency supplies by air, but where fuel for return voyages could not be provided.

6. There are other instances where long voyages without refueling may be desirable, if not mandatory. This is particularly true in the Pacific area. Air transportation from principal points across this ocean is usually made in hops of 2000 to 3000 miles. Where one such stepping stone may be missing, as at Wake or Guam, whether by enemy occupation or weather that interferes with navigation or landing, the airplane is hard pressed to make the remaining distances, if it can at all. Such trips provide an ideal field for airship operation. Since the airship is not well adapted to cargo carrying at high altitudes, as may be required in crossing mountainous terrain, it is obvious that marine operations, especially on the Pacific Ocean, represent an ideal field for airship cargo transportation.

Acknowledgment

Appreciation is expressed for the aid and kindly suggestions given by C. P. Burgess, O. K. Bingham, Ensign C. DeLauer and others.

Appendix I

Fig. 3 is a development of Fig. 1 where the weight ratio is 2. Points for each percentage of useful load at a given L/D were converted to an equivalent ratio of plane weight before and after consuming the fuel for a given voyage, and this ratio used in Fig. 1 or formula (7). For instance, consider the determination of the range corresponding to an L/D of 16, and a fuel consumption 20% of the useful load (which is conversely equivalent to a cargo 80% of the useful load). Since the ratio of gross weight to empty weight has been assumed as 2, the useful load is one-half the gross weight, and the ratio of weights of the plane before and after consuming this amount of fuel is:

$$\frac{\text{(Weight Empty)}+\text{(Useful Load)}}{\text{(Weight Empty)}+\text{(Cargo)}} = \frac{1+1}{1+0.80} = \frac{2}{1.8} = 1.11$$

which from Fig. 1 corresponds to 1000 miles.

Appendix II

Notation:

P = Payload, lb

F = Initial fuel load, lb

[15] See "Military Ground Force Fuels and Lubricants," by Major James A. Richardson, III. Paper presented at the SAE National Fuels and Lubricants Meeting, Tulsa, Okla., Nov. 5, 1943.

w = Fuel consumption, lb per mile
L = Length of voyage, statute miles
V = Ship's speed, statute miles per hr
U = Useful load, lb

Then:

$$P + F = U$$

$$w = \frac{V^2}{k}, \text{ where } k \text{ is a constant}$$

Derivations:

$$F = Lw = \frac{LV^2}{k} \tag{A}$$

$$P = U - F = U - \frac{LV^2}{k} \tag{B}$$

$$PV = UV - \frac{LV^3}{k} \tag{C}$$

$$\frac{d(PV)}{dV} = U - \frac{3LV^2}{k} \tag{D}$$

$$= 0 \text{ when } V_o^2 = \frac{UK}{3L} \tag{E}$$

where V_o = optimum speed.

The maximum ton-miles of payload is obtained when PV is a maximum; and this occurs when:

$$V_o^2 = \frac{UK}{3L} \tag{F}$$

Substituting this value of V in (A) gives:

$$F = \frac{LV_o^2}{k} = \frac{U}{3} \tag{G}$$

$$\text{Delivery Factor} = \frac{2}{3}\frac{UV_o}{\text{(Empty Weight)}} \tag{H}$$

Appendix III

The values of k and V_o may be obtained as follows:

Notation:

c = Specific fuel consumption, lb per hp per hr
η = Propeller efficiency
C_D = Drag coefficient of airship
$\frac{\text{Vol}_E}{10^6}$ = Envelope volume, millions of cu ft
$\frac{\text{Vol}_G}{10^6}$ = Total volume of helium carried, millions of cu ft
Hp = Engine power
V = Miles per hour
r = Ratio of $\left(\frac{\text{Vol}_G}{\text{Vol}_E}\right)$

$$\text{Drag (lb)} = C_D \frac{\rho}{2}\left(\frac{\text{Vol}_E}{10^6}\right)^{\frac{2}{3}} V^2 \times 10{,}000 \times 2.15$$

$$= 10{,}750\, C_{D\rho}\left(\frac{\text{Vol}_E}{10^6}\right)^{\frac{2}{3}} V^2 \text{ [Reference 13]}$$

$$\text{Hp} = \frac{\text{Drag} \times V}{375} = \frac{28.67\, C_{D\rho}\left(\frac{\text{Vol}_E}{10^6}\right)^{\frac{2}{3}} V^3}{375}$$

$$^*\text{Hp} = \frac{\left(\frac{\text{Vol}_E}{10^6}\right)^{\frac{2}{3}} V^3}{454} \tag{I}$$

*Where: $c = 0.45$; $C_D = 0.0236$; $\eta = 0.73$; $\rho = 0.002378$ (s.l.)

$$w = \frac{\text{Hp}\, c}{V} = 28.67\left(\frac{\text{Vol}_E}{10^6}\right)^{\frac{2}{3}} V^2 \frac{c\rho}{\eta} C_D$$

$$^*w = \frac{\left(\frac{\text{Vol}_E}{10^6}\right)^{\frac{2}{3}} V^2}{1008} \tag{J}$$

$$k = \frac{V^2}{w} = \frac{\eta}{28.67\left(\frac{\text{Vol}_E}{10^6}\right)^{\frac{2}{3}} cC_{D\rho}}$$

$$= \frac{14.65}{\left(\frac{\text{Vol}_E}{10^6}\right)^{\frac{2}{3}}} \times \frac{\eta}{cC_D}$$

$$^*k = \frac{1008}{\left(\frac{\text{Vol}_E}{10^6}\right)^{\frac{2}{3}}} \tag{K}$$

Appendix IV

Empiric approximations:

From Fig. 10:

$$U = 20{,}000\left(\frac{\text{Vol}_G}{10^6}\right)^{1.12} = 20{,}000\left(\frac{\text{Vol}_E}{10^6}\right)^{1.12} \cdot r^{1.12}$$

When $r = 0.88$

$$U = 17{,}350\left(\frac{\text{Vol}_E}{10^6}\right)^{1.12}$$

Optimum Speed:

From equations (F) and (K):

$$V_o = \sqrt{\frac{UK}{3L}} =$$

$$\sqrt{\frac{20{,}000}{3L}\left(\frac{\text{Vol}_E}{10^6}\right)^{1.12} \cdot r^{1.12} \cdot \frac{14.65}{\left(\frac{\text{Vol}_E}{10^6}\right)^{\frac{2}{3}}} \cdot \frac{\eta}{cC_d}}$$

$$= \sqrt{\frac{97{,}600}{L}\left(\frac{\text{Vol}_E}{10^6}\right)^{0.453} r^{1.12}}\ \sqrt{\frac{\eta}{cC_d}}$$

$$= \frac{312}{\sqrt{L}}\left(\frac{\text{Vol}_E}{10^6}\right)^{0.22} \cdot r^{0.56}\sqrt{\frac{\eta}{cC_d}}$$

$$= \frac{291}{\sqrt{L}}\left(\frac{\text{Vol}_E}{10^6}\right)^{0.22}\sqrt{\frac{\eta}{cD_d}} \tag{L}$$

$$^*V_o = \frac{2420}{\sqrt{L}}\left(\frac{\text{Vol}_E}{10^6}\right)^{0.22} \tag{M}$$

Delivery Factor at Optimum Speed:

From equation (H),

$$D_f = \frac{2}{3}\frac{UV_o}{\text{(Empty Weight)}}$$

From Fig. 11:

$$\frac{U}{\text{(Empty Weight)}} = 0.5\left(\frac{\text{Vol}_E}{10^6}\right)^{0.2}$$

Therefore:

$$D_f = \frac{2}{3} \cdot \frac{312}{\sqrt{L}}\left(\frac{\text{Vol}_E}{10^6}\right)^{0.22}\sqrt{\frac{\eta}{cC_d}} \cdot r^{0.56} \cdot 0.5\left(\frac{\text{Vol}_E}{10^6}\right)^{0.2}$$

$$= \frac{104}{\sqrt{L}}\left(\frac{\text{Vol}_E}{10^6}\right)^{0.42} r^{0.56}\sqrt{\frac{\eta}{cC_d}} \tag{N}$$

$$*D_f = \frac{928}{\sqrt{L}} \left(\frac{\text{Vol}_E}{10^6}\right)^{0.42} \qquad (O)$$

Maximum Speed:

Assuming engines weigh 4.5 lb per hp and 5% of gross lift:

From equation (9):

$$\text{Hp} = 690 \left(\frac{\text{Vol}_E}{10^6}\right) \cdot r$$

From equation (11):

$$V = \sqrt[3]{\frac{14.5\,\eta\,\text{Hp}}{C_d \left(\frac{\text{Vol}_E}{10^6}\right)^{\frac{2}{3}}}} = \sqrt[3]{\frac{14.5\,\eta\,690 \left(\frac{\text{Vol}_E}{10^6}\right)}{C_d \left(\frac{\text{Vol}_E}{10^6}\right)^{\frac{2}{3}}}} \cdot r$$

$$= \sqrt[3]{\frac{10{,}000}{C_d} \left(\frac{\text{Vol}_E}{10^6}\right)^{\frac{1}{3}}} \cdot r\eta$$

$$= 21.5 \left(\frac{\text{Vol}_E}{10^6}\right)^{\frac{1}{9}} \sqrt[3]{\frac{r\eta}{C_d}} \qquad (Q)$$

$$*V = 61 \left(\frac{\text{Vol}_E}{10^6}\right)^{\frac{1}{9}} \qquad (R)$$

DISCUSSION

T. A. Knowles (Goodyear Aircraft Corp.): The yardstick which Comdr. MacCoull has presented is an interesting one for comparing different types of aircraft. If I may be allowed to redefine delivery factor, it represents a measure of the quantity of cargo which a vehicle of unit weight is able to deliver in a given time. In this respect and in a present wartime sense, it may also be viewed as cargo-delivering ability measured in terms of the outlay of critical aircraft material.

In so far as the weight of an aircraft is directly indicative of the operational cost of such an aircraft, the factor also is representative of the fundamental operating economy. This weight-operating cost assumption is not completely valid, although it is an interesting first approximation in cases where only a few large units are purchased and where the corresponding depreciation and maintenance charges form a large portion of the total operating cost. This has been the case in the past in long-range over-water operations where only a few very large units are used on relatively infrequent schedules.

It is appreciated that, as long as the war continues, there will be military transport routes created on which the most important thing is to secure the transfer of cargo, and the final cost of delivering the cargo is inconsequential. This is distinct from the commercial approach in which the cost per ton-mile is most important if fares are to be held to reasonable levels in order to attract traffic. It should be observed that, even in the military sense, operating costs per ton-mile should not be neglected inasmuch as any excessive expenditures for operating costs merely represent the utilization of funds, material, and manpower in an inefficient manner when they could be more profitably employed elsewhere for the furtherance of the war effort.

Our own approach at Goodyear Aircraft has been along this commercial line, and our thinking has been geared to costs per ton-mile. From figures published in the course of hearings on route certificates for American Export and Pan American Airways, it is apparent that nonstop transoceanic operations with modern flying boats of the 60,000-80,000-lb classes in the pre-war period could be set at just over $3.00 per mile. The Hindenburg demonstration series across the Atlantic in 1936 established complete all-inclusive costs for this type of operation at $6.66 per airship mile, or roughly twice the cost per mile for the pre-war clipper services. For twice the cost per mile, however, the Hindenburg's passenger complement was something like five times the number of passengers which these flying boats could carry on a nonstop basis, with the net effect that the operational cost per passenger-mile was markedly in the airship's favor. Reduced to a potential cargo-carrying basis, the Hindenburg's figure worked out at about 11¢ per ton-mile of cargo in trans-Atlantic service.

Another series of background figures which tend to support Comdr. MacCoull's contention that the long-range routes belong to the airship, have been derived from a recent study of the comparative operating costs of fleets of airships and airplanes. Taking a 2500-mile distance as the minimum distance over which airships might be employed in nonstop operation, and assuming that like total investments are made in airships and in airplanes, it develops that approximately an equal quantity of cargo can be delivered per week by the airships and airplanes. The material required for the fleet of airplanes is roughly twice that needed for the fleet of airships, the installed horsepower of engines in the airplanes total some 7½ times that installed in the airships, the total flight personnel of the airplane service is about 1 2/3 that required for the airship service, and the fuel consumed per week in delivering the same quantity of cargo is roughly three times as much for the fleet of airplanes as it is for the fleet of airships. The cost for carrying each ton of cargo 1 mile over such a route by airplane is roughly twice that involved in transportation by airship.

Projecting this study to the longer nonstop routes, the radical falling off in airplane payload as compared to the large payload which the airship can still carry at the longer ranges, results in a marked increase in the number of airplanes required to deliver a like cargo in a given period of time with a corresponding increase in the above ratios for critical materials, flight personnel, and fuel consumed. The same reduced ability to carry payload manifests itself in markedly increased operating costs for airplanes used over long nonstop distances, and the ratio of airplane ton-mile operating cost increases to three or more times that of the airship as the 4000-mile trans-Atlantic operating distance is approached.

H. E. Hoben (American Airlines, Inc.): Throughout the discussion, the effect of winds is entirely ignored. Since winds do exist and since their effect will be much greater upon the slower flying airship than on the faster airplanes, it is felt that their omission is open to some criticism.

Comdr. MacCoull: The effect of wind has been omitted purposely as have other variables which would have made the generalizations of this paper impracticably complicated. Also, there is no simple mathematical solution to such a question, since an airship does not buck a wind as an airplane might. The great endurance of an airship in flight and the ample facilities available for an exceptionally well-organized meteorological station aboard have made it possible for ships such as the Hindenburg to adjust her course to follow favoring winds and weather. Dr. Eckner has demonstrated that this is actually practical, and that flight of an airship against headwinds is rarely necessary for extended periods in commercial long-range operation.

Mr. Hoben: How about the hazards of ice?

Comdr. MacCoull: The answer to this is quite similar to the question about headwinds: Airships should avoid bad icing conditions. In organized commercial operations, alternate terminals will probably be located in areas relatively free from ice, and passengers will be flown directly to their destinations in more northerly locations by hook-on planes. In fact it is probable that many passengers will board and leave these ships by planes hooked on, or launched from the airships in flight. Also, de-icing is being studied, and there seems to be reasonable prospects for prevention of serious ice accumulation.

Mr. Knowles: With respect to the queries of Mr. Hoben on the subject of headwinds and ice formation, the following may be added:

Due to the fact that airships would operate over long nonstop distances and due to the fact that their large fuel reserves and other operating characteristics permit wide deviations from a great-circle course, it is possible to navigate rigid airships on a weather map flight basis, and this technique was highly developed by the German operating group. The flight plan is initially chosen and is adjusted enroute so as to secure the maximum assistance or encounter the least resistance from the prevailing atmospheric circulation. Over a period of several years operations, average time and speed from the German flight logs reveal the actual accomplishment of flights on which headwinds have been balanced out. Since headwinds were obviously encountered, their net effect has been offset by the choice of alternative courses. With respect to the relationship of high-velocity headwinds to the airship's flight speed, the actually recorded scheduled completions of flights by the German airships is the best indication I know that the headwinds actually encountered have not been a serious deterrent to schedule keeping when the air speed of these ships has been as low as about 60 knots. Future airships would be somewhat faster and should, therefore, be influenced to a lesser degree. Schedules offered by commercial airship performance have been based on detailed weather map analyses of trans-Atlantic and trans-Pacific routes, and therefore include statistical appraisal of the actual influence of recorded headwinds on airships operating on these routes at these air speeds.

The best solution to the ice problem is to avoid it by not deliberately flying into such conditions. However, it should be pointed out that ice formation on airships is not nearly as serious as it is on airplanes, for the simple reason that there is merely an increase in the weight of the airship and there is no interference with the form

of the lifting surface, which is such a critical factor in the case of the airplane wing. Furthermore, by the use of dynamic lift, that is, by flying the airship at a slight angle of pitch, very large amounts of added weights may be carried dynamically. By this expedient it would be comparatively easy to carry something like 30 tons of additional weight if the captain were called upon to do so.

Capt. M. R. Pierce, U.S.N. (Ret.): In sailing-ship days, vessels from New England engaged in what was known as the three-cornered trade. A sailing vessel would clear from a New England port loaded with knick-knacks and notions and, taking advantage of the westerly winds, would cross over to Africa on a northern route. Its cargo would then be exchanged for slaves and ivory. The ship would then sail to the West Indies, taking advantage of the trade winds, where the cargo would be exchanged for rum and molasses. The third leg of the run would then be made up the Atlantic Coast to New England. In this manner the vessels usually kept a fair breeze all the way.

Something like this might be done with an airship route in the Pacific. An airship leaving San Francisco would carry the trade winds at her back all the way to Auckland, New Zealand, and/or Sydney, Australia. The second leg of the trip would be from Sydney to a large China port, such as Hong Kong or Shanghai. The return trip to the states would be made by the great-circle route, bearing the great-circle route in accordance with the current weather map. In this way the ship would have fair winds a large percentage of the time. During such a trip the ship would pass over many important islands on the first and second legs. Deliveries could be made to and from these islands by airplane trapeze.

Capt. C. V. S. Knox, U.S.N.: In regard to the comments on the triangular sailing course in the Pacific, it should be noted that, in a year or more of operation by German zeppelins, they came up with a season's average of zero headwinds. In other words, the other way to beat headwinds is to avoid them.

Chairman William B. Stout: What will happen to an airship in a thunderstorm?

Capt. Knox: Experience by the Navy with the smaller nonrigid airships in Southern waters indicated remarkable success in passing through thunderstorm areas. The ships were flown low over the water, and if it proved impossible to seek out a light spot, it had been found possible to fly through the frontal condition.

In order to clear a point, the writer of the paper stated that all calculations were based on the use of the safe helium and did not take advantage of the greater lift of hydrogen with its obvious risks.

Since there is a definite penalty incurred by flying airships out of equilibrium, normal long-range operations might require airships to be flown heavy during the first part of their voyage and light toward the end of the voyage. This is avoided when using hydrogen by valving out the relatively cheap hydrogen as the voyage progresses so as to keep the ship in equilibrium, returning flying it light and down by the nose with a possible reduction in air-speed with a given power. Comdr. MacCoull stated that calculations for the airships, as given in the paper, included weight allowance for suitable water-recovery apparatus. This apparatus, whether for diesels or gasoline engines, permits condensation and retention on board of the water vapor present in the exhaust gases; thus, 100% of the weight of fuel burned may be converted to water, and so, the ship may avoid the penalty of flying light with its nose down by reason of burning up the weight of fuel.

Stuart C. Plummer (Goodyear Aircraft Corp.): With modern aids to navigation, it is not necessary to fly through a thunderhead, and such practice would be undesirable from the standpoint of passenger comfort. From time to time, gusts of approximate thunderhead proportions will be unavoidable. Many years of study have been devoted to the determination of atmospheric gust structure, and these data have been applied to a hypothetical airship framework, with the result that reasonably accurate design load factors are now available for the design of a new modern airship. This, coupled with the advancements made in structural design will result in a high degree of flight safety. A saving in structural weight will be reflected in an overall increase in operating efficiency. A further appreciable increase in efficiency will be made possible by improving the aerodynamics of the hull and empennage, and by the removal of many of the appendages apparent on airships of previous design.

In the interest of improving propulsive efficiency, much experimental and theoretical effort has gone into the problem connected with locating a propeller at the stern. It appears that the 15% increase in propulsive efficiency mentioned in this paper is a reasonable estimate of the gain to be realized from this improvement. I believe that gains in excess of this amount are ultimately obtainable.

Mr. Knowles: With respect to the chairman's inquiry on the subject of thunderstorms, I would like merely to add to what Capt. Knox has said by pointing out that the answer appears to lie in the mastery of a thunderstorm or frontal-approach technique. Many such experiences have been successfully handled by our own crew of blimp pilots, and they are, of course, part of the published records of the Graf Zeppelin and the Hindenburg operations. The basic elements in this technique are to place the airship in a good condition of equilibrium and to approach the point of minimum disturbance at reasonably low altitude and slow speed. In overocean operations the navigator has some opportunity to select the point of frontal passage, or even perhaps to avoid it entirely. In operations of the Graf Zeppelin, however, due to French military restrictions on the size of the belt which the German airship could utilize in crossing French territory on its way to its home port, the German airships were repeatedly forced to fly through narrow mountain passes in which intensive thunderstorms were present. That no damage to the ships occurred is a testimonial to the fact that proper operating technique will solve this problem.

Question: Doesn't the fact that all the rigid airships we have built have been lost by accident prove that they are impractical?

Comdr. MacCoull: There are two ways of looking at such facts, one is statistical, the other analytical. From the statistical angle the record does look bad, but no engineer would attempt to extrapolate statistical data from the past into the future, unless he has no other information. America has built a total of only three rigid airships. True, these ships were ultimately lost in accidents occurring during their operations as military craft. The loss of these ships can be charged to the penalty of progress. In each case, important knowledge was gained which advanced the art another step. Much of our American progress in heavier-than-air can be traced through a long series of trials and failures, reaching success only after much disappointment, trial, and error. Where would we be today had we been willing to accept as the ultimate the dismal performance of the early multi-engined airplane?

The way to look at such a question, as engineers well know, is from the analytical angle. If the cause of any failure can be learned and corrected, we may bring success from failures, as is done every day in any research laboratory. It is my own belief that technical reasons for most of the rigid airship losses are sufficiently well understood among airship men, so that they should be prevented in future airships.

The operational record established by the Graf Zeppelin and Hindenburg stand as conclusive proof that airships can be built and operated successfully under all conditions with a high degree of regularity, safety, and schedule-keeping ability. The use of our exclusively American helium gas rules out the fire hazard, which caused the loss of the Hindenburg.

Obviously, anything that moves presents a hazard, and no medium of transportation can offer absolute guarantees against mishap, any more than such guarantees can be given for a man's safety in an automobile – but the probability of such an accident is accepted as too small to interfere with our daily enjoyment of the family automobile. There does not now appear to be any insurmountable engineering obstacle to prevent the construction and safe operation of large rigid airships.

R. H. Upson (H. J. Heinz Co.): Comdr. MacCoull's paper is something that we have much needed, giving as it does fundamental facts without prejudice for either heavier-than-air or lighter-than-air. His bases of comparison, the delivery factor and ultimate range, are generally sound, and his demonstration of the optimum fuel-load proportion most useful. Nevertheless, the attention that he gives to lighter-than-air makes his paper notable in comparison with other recent treatments of the air cargo problem.

From his figures, it is hard to justify practical airship transportation in competition with airplanes on a purely commercial basis. For example, in Table 3 his modernized Macon is shown to have an ultimate range of 10,200 miles at 70 mph, which is more than any airplane has made to date. On the other hand, Fig. 1 shows that a modernized airplane with an L/D ratio of 21, at the assumed weight ratio of 2, can cover the same distance; and Fig. 2 shows that the required drag characteristics can be attained with a parasite drag, exclusive of wing, equal to about 0.9 the wing profile drag, at an optimum speed 2¼ times that of the airship. It still remains that airships can negotiate longer nonstop distances than airplanes, and always will; but it happens that the earth on which we live is finite in size; and offers, for the most part, commercial nonstop routes that are well within the economical range of heavier-than-air potentialities.

The military, particularly naval, problem is quite a different one, however. Comdr. MacCoull has mentioned some of the special conditions which then exist. Still further, the airship as a carrier of bombing and fighting airplanes has possibilities of the utmost importance; so much so that it may be well worth while, in guarding future Pearl Harbors, to give governmental encouragement to commercial airships sufficient to assure their military availability in time of need.

Still another possibility exists; that of improving the airship itself,

concluded on page 485

ALTITUDE VA

by W. H. CURTIS
and R. R. CURTIS
Thompson Products, Inc.

AN investigation of vapor formation in aircraft fuel systems at altitude led the authors to a number of conclusions, some of which are:

Operation of a centrifugal fuel-booster pump properly installed will reduce vapor potential of fuel on the discharge side of the booster.

Use of an electric-motor-driven fuel pump at any point in the fuel line between the fuel tank and the engine fuel pump has no effect upon the altitude at which vapor may appear in the carburetor, neither does it reduce its volume.

The VP/*L* samples, in general, confirm data obtained by direct observation. Refinements in procedure and additional tests should permit the development of evaluation factors that will increase the usefulness of this technique.

The reduction of vapor by centrifugal booster operation has been confirmed by flight test.

■

THE AUTHORS: W. H. CURTIS (M '41), president of Curtis Pump Co., an affiliate of Thompson Products, Inc., is father of the Curtis-Curtis engineering team. He has been engaged since 1929 in research, development and manufacturing activities in pumps and meters for the petroleum industry. For the past four years he has been West Coast resident engineer for Thompson Products. R. R. CURTIS (M '41), son, is executive vice-president of Curtis Pump Co. A graduate from the University of Cincinnati, Mr. Curtis has been associated successively with Glenn L. Martin Co., and Romec Pump Co. He is now specializing in research and development on aircraft pumps.

FOR some time there has been a need for a method of simulating vapor formations in fuel systems so that visual examination could be directed to any condition at any point in the system, and, as a correlated procedure, a means of quickly determining by test the vapor-forming properties, or vapor potential, of the fuel under substantially identical conditions. Thus, visual observation of vapor in a correctly simulated system would be confirmed by a vapor-potential test of the fuel itself, both being focused upon the same point in the fuel system. The object of this series of tests was to explore the possibility of developing these procedures.

The forming of vapor in a simulated fuel system proved to be a comparatively simple matter. The determination of the vapor potential, or the amount of vapor that would form under a given set of conditions and persist for sufficient time to cause faulty functioning of some unit of the fuel system, presented a new problem in that it was necessary to base the test upon the assumption that equilibrium between fuel and vapor was absent, this being one of the conditions present in a closed system where both temperature and pressure are in a state of change. We therefore adopted a new technique, and this is a report of the first attempt to develop its applicability. Considerable work remains to be done before it will have been evaluated to the point where it can be of practical value. However, the preliminary results appear to justify the opinion that it holds much promise of becoming a quick and accurate method of not only confirming the observed behavior of a given fuel in a simulated fuel system, but an equally valuable means of making quick comparisons between different fuels to determine their vapor-forming tendencies, or, as we have called it, vapor potential.

[This paper was presented at the SAE National Aircraft Engineering & Production Meeting, Los Angeles, Calif., Oct. 1, 1943.

■ Equipment

A diagram of the test equipment arrangement is shown in Fig. 1. Further details are shown in Figs. 2, 3, 4, 5, and 6.

■ Procedure

In general, the plan adopted provided for the circulation of the fuel in the simulated system at a rate of flow comparable with the requirements of many airplane engines, while imposing a rate of climb on the system as a whole, and a temperature rise on the fuel as it passed from the fuel tank to the zone marked "5-psi carburetor." (This represents the metered chamber in high-pressure carburetors. The zone marked "9-psi carburetor" represents the unmetered chamber). The fuel was then passed through a refrigerated heat exchanger to remove the added heat. Thus, it was returned to the fuel tank at the original tank fuel temperature.

Standard conditions for all test runs were:

1. Fresh, specification AN-F-28 fuel, amendment No. 1, grade 130, 30 gal each run.
2. 2000 fpm rate of climb.
3. 900 lb per hr fuel flow rate.
4. Temperature rise imposed on fuel between fuel tank and carburetor zones.
5. Pressure drop imposed in two controlled stages as follows:

   15 psi to 9 psi in zone marked "9-psi carburetor."
   9 psi to 5 psi in zone marked "5-psi carburetor."
6. The engine fuel pump was operated continuously.
7. The zero point for all pressures, except vapor-potential sampling gages, is the pressure of the ambient atmosphere at the altitude noted.

At increments of 4000-ft altitude, beginning with sea level, vapor-potential samples were drawn. The start of the draw was timed a few seconds before the altitude was reached so that the period required to complete the draw would straddle the altitude to which it was referenced. At no time was the climb interrupted.

The sampling procedure was planned as follows and, with few exceptions, appears to be satisfactory and capable of standardization.

VAPOR FORMATION in Aircraft Fuel Systems

1. All gages were recalibrated against a mercury manometer to read in inches of mercury absolute.

2. The bottles, with gages attached, were evacuated to within 100 microns of absolute zero.

3. Only those containers checked for vacuum holding were used.

4. All evacuated containers were held in a water bath, having a temperature approximately the same as the fuel being sampled, until they were used. See Fig. 4.

5. The draw was made through stopcocks having a 1-mm bore. See Fig. 5.

6. All draws were made to the absolute pressure in the zone being studied. *In this case all samples were drawn to the absolute pressure existing in the 5-psi carburetor zone.*

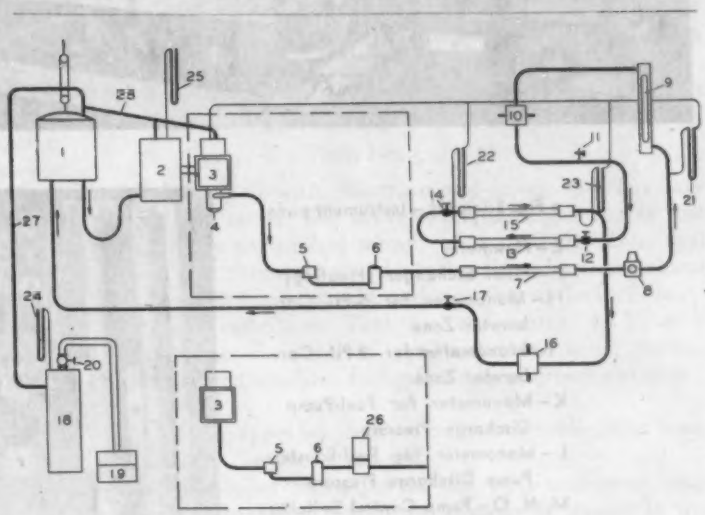
7. The draw was continuous until the gage hand passed the required pressure by 1-2 in. of mercury. Following the first shut-off, the gage hand moved to a lower pressure by reason of absorption and condensation of excess vapor formed during the draw. A series of adjusting draws was then made until the gage hand remained stationary at the required pressure for approximately 2-3 sec. This was considered to be the end point.

8. All samples were drawn from the fuel line at a point carrying full carburetor pressure at the maximum imposed temperature.

9. No effort, beyond preheating the evacuated sample bottle, was made to control the temperature of the sample.

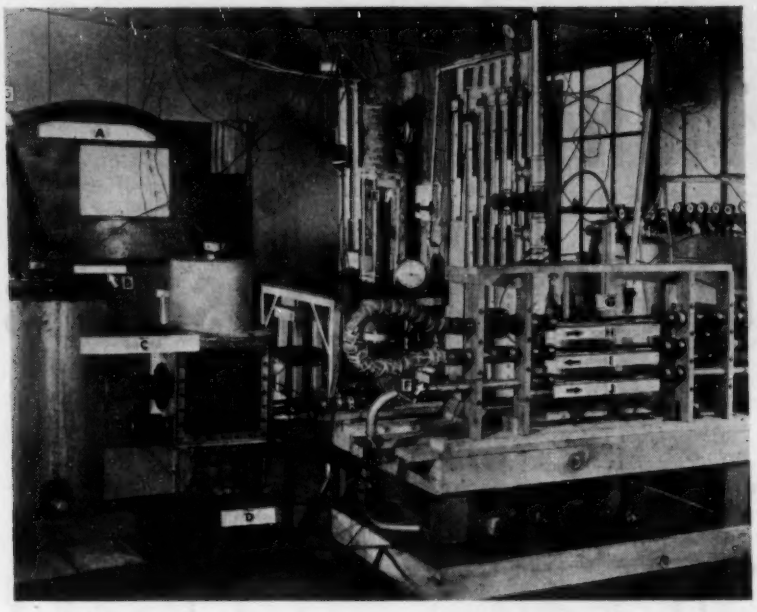
From this procedure it will be seen that while drawing the sample, conditions of pressure drop and shock or turbulence are imposed that cause the formation of an excess of vapor. The major portion of this excess is rapidly absorbed or condensed. That which persists for a 2-3 sec period at the absolute pressure existing in the part of the system under examination is considered to be the vapor potential. Equilibrium is not present, for, as the sample stands for a longer period, additional vapor is slowly absorbed and the gage pressure will drop. But, vapor that will persist for as little as 2 sec, can travel a considerable distance in a fuel system and thereby reach a point where it may cause faulty operation. This is the condition that is dealt with here.

Moreover, the method automatically takes into account the physical changes that take place in the fuel during a climb to altitude as they may be affected by both rate of



■ Fig. 1 – Diagram of equipment arrangement

1 – Fuel leveling tank – arranged to be raised or lowered to adjust fuel level in (3)
2 – Controlled-level fuel tank
3 – Box-like extension of (2) with three transparent sides to permit view of action inside tank
4 – Centrifugal fuel-booster pump
5 – Selector cock
6 – Strainer
7 – Transparent-section fuel line – marked "fuel pump suction" in photographs
8 – G-9 engine fuel pump
9 – Flowmeter
10 – Heat exchanger to raise fuel temperature – uses hot water
11 – Stopcock for drawing vapor-potential samples
12 – Valve for controlling rate of fuel flow
13 – Transparent-section fuel line – marked "9-psi carburetor" in photographs
14 – Valve for controlling pressure in (13)
15 – Transparent-section fuel line – marked "5-psi carburetor" in photographs
16 – Heat exchanger to lower fuel temperature – uses refrigeration
17 – Valve for controlling pressure in (15)
18 – Altitude tank – altitude of airplane determined by pressure in this tank
19 – Vacuum pump
20 – Valve for controlling rate of climb
21 – Manometer – fuel-pump discharge pressure
22 – Manometer – first carburetor pressure
23 – Manometer – second carburetor pressure
24 – Manometer – airplane altitude
25 – Manometer – equivalent altitude or pressure within fuel tank
26 – Electric-motor-driven G-9 fuel pump used in run No. 3
27 – Vent line – fuel tank system to altitude tank
28 – Vent line – pressure balancing

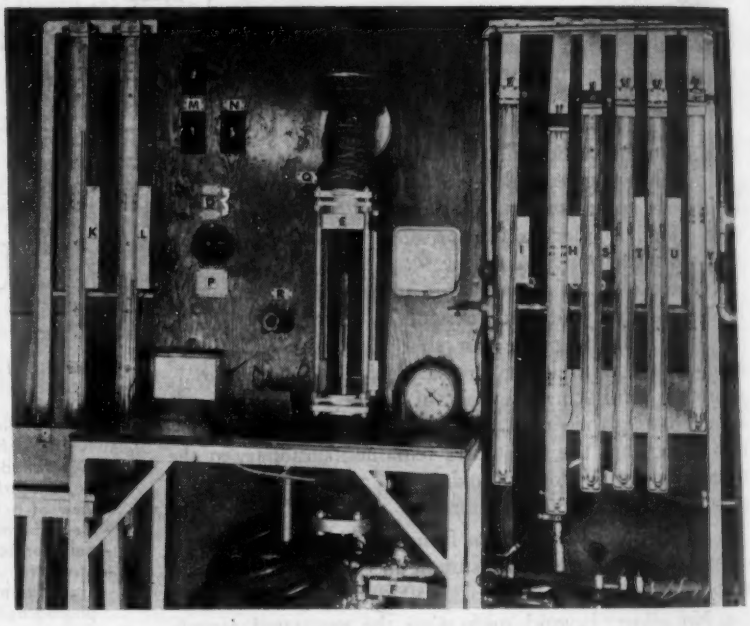


■ Fig. 2 (left) – General view of equipment

A – Vent Lines
B – Refrigerated Heat Exchanger (Cooling)
C – Fuel Tank
D – Centrifugal Fuel Booster
E – Flowmeter
F – Heat Exchanger (Heating)
G – Altitude Sign
H – 5-psi Carburetor Zone
I – 9-psi Carburetor Zone
J – Fuel Pump Suction

■ Fig. 3 (right) – Instrument panel

E – Flowmeter
F – Heat Exchanger (Heating)
H – Manometer for 5-Psi Carburetor Zone
I – Manometer for 9-Psi Carburetor Zone
K – Manometer for Fuel-Pump Discharge Pressure
L – Manometer for Fuel-Booster-Pump Discharge Pressure
M, N, O – Pump Control Switches
P – Potentiometer Switch
Q – Voltage Control
R – Rate-of-Fuel-Flow Control Valve
S – Manometer – Not Used
T – Manometer – Altitude in Fuel Tanks
U – Manometer – Altitude in Altitude Tank
V – Manometer – Rate of Climb



climb and the altitude at time of sampling.

Since the sample bottle is substantially free of air at the start of the draw, any space not occupied by liquid when the draw is completed must represent fuel vapor and air released from the fuel. The total vapor thus formed has been termed "vapor potential," symbol "VP," to distinguish over the symbol V commonly used to designate total vapor in equilibrium.

The ratio of the volume of this vapor to the volume of the fuel in the sample is indicated by the notation VP/L. Its value is usually written as a decimal. Algebraically it is obtained by the equation:

$$\frac{VP}{L} = \frac{T\text{-}L}{L}$$

where

T = Total volume of sample container, ml
L = Volume of liquid in sample, ml

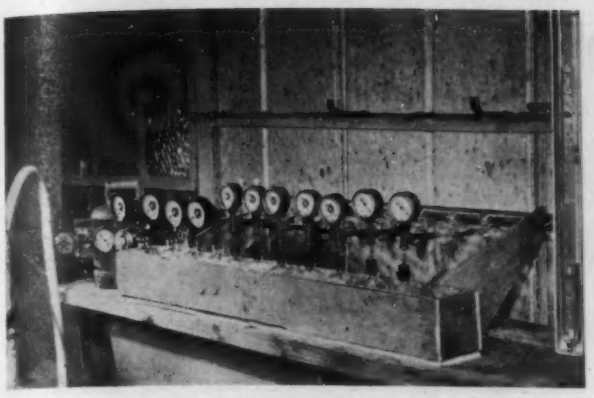
■ Test Runs

A total of eight test runs were made. Three were rejected because of errors that were considered sufficient to invalidate the results. A photographic record of the five valid runs was made on a moving picture film. It presents all of the details in much clearer form than is possible in a written description. However, for those who may not have an opportunity to see the film, the following synopsis will be of interest.

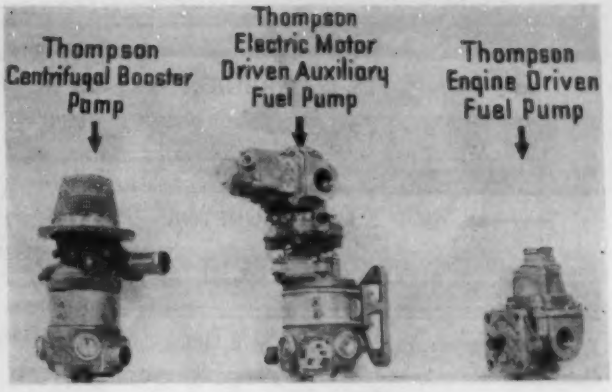
Run No. 1

Pump equipment: Centrifugal booster pump on fuel tank; G-9 engine fuel pump – 2100 rpm.

Specific conditions: Tank fuel temperature 85 F; 20 F temperature rise – fuel tank to 5-psi carburetor zone; centrifugal booster pump not operated until vapor appeared in 5-psi carburetor zone.



■ Fig. 4 – Sample evacuating stand



■ Fig. 6 – Pumps used

Notes:

1. Vapor appeared in "fuel pump suction" at 4000-ft altitude.
2. Vapor appeared in "5-psi carburetor" at 11,400-ft altitude.
3. The centrifugal booster was started at about 12,000-ft altitude but was stopped when no reduction in vapor was noted in "5-psi carburetor" after 23 sec.
4. It was later discovered that we had not operated the centrifugal booster for sufficient time.
5. The system failed at 14,200-ft altitude by loss of fuel pressure.

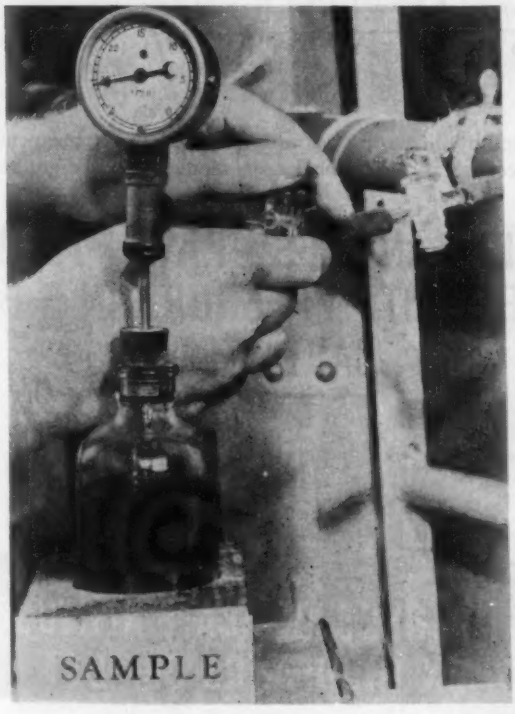
Run No. 2

Pump equipment: Centrifugal booster on fuel tank; G-9 engine fuel pump – 2100 rpm.

Specific conditions: Tank fuel temperature 85 F; 20 F temperature rise – fuel tank to 5-psi carburetor zone; centrifugal booster operated continuously.

Notes: No vapor made its appearance at any time in the three observation zones except for a very brief interval at 16,000-ft altitude. This was later found to have been caused by an error in pressure control.

No indications of failure were observed during the climb to 40,000-ft altitude.



■ Fig. 5 – Drawing sample

Runs No. 3 and 3R

Pump equipment: Electric-motor-driven G-9 auxiliary fuel pump in fuel line between fuel tank and engine fuel pump; G-9 engine fuel pump – 2100 rpm; the centrifugal pump was removed from the fuel tank and a conventional strainer was fitted over the tank opening to the fuel line.

Specific conditions: Tank fuel temperature 85 F; 20 F temperature rise – fuel to 5-psi carburetor zone; electric-motor-driven auxiliary fuel pump operated continuously.

Notes:

1. Vapor appeared in the 5-psi carburetor zone at 8000-ft altitude. See Fig. 7, run 3*R*.
2. The vapor formation in the 5-psi carburetor zone increased in volume until an altitude of 24,400 ft was reached, at which point vapor suddenly appeared in the fuel-pump suction and the system failed by loss of fuel pressure. See Fig. 8.

Run No. 4

A repeat of run No. 1 except for longer period of centrifugal booster operation to note effect on vapor in 5-psi carburetor zone.

Notes:

1. Vapor appeared in the fuel-pump suction at 4000-ft altitude.
2. Vapor appeared in 5-psi carburetor zone at 10,000-ft altitude.
3. The centrifugal booster was started at 10,500-ft altitude and remained in operation until the 5-psi zone was cleared of vapor.
4. The time required to remove all vapor was 126 sec. The altitude at the time was 14,700 ft.

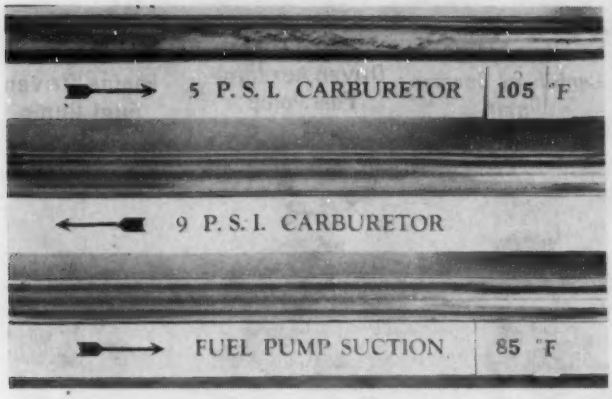
Run No. 5

Pump equipment: Centrifugal booster pump on fuel tank; G-9 engine fuel pump – 2100 rpm.

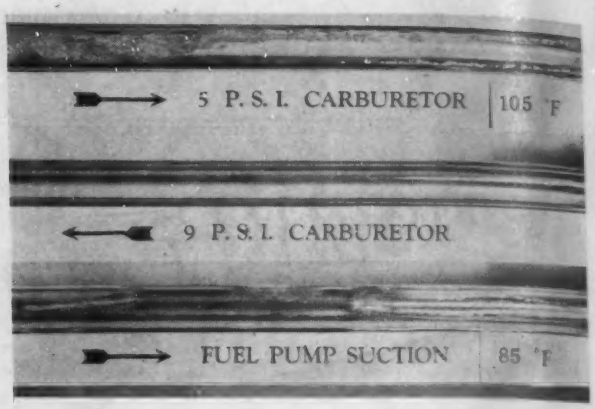
Specific conditions: Tank fuel temperature 110 F; 10 F temperature rise – tank to 5-psi carburetor; centrifugal booster operated continuously.

Notes:

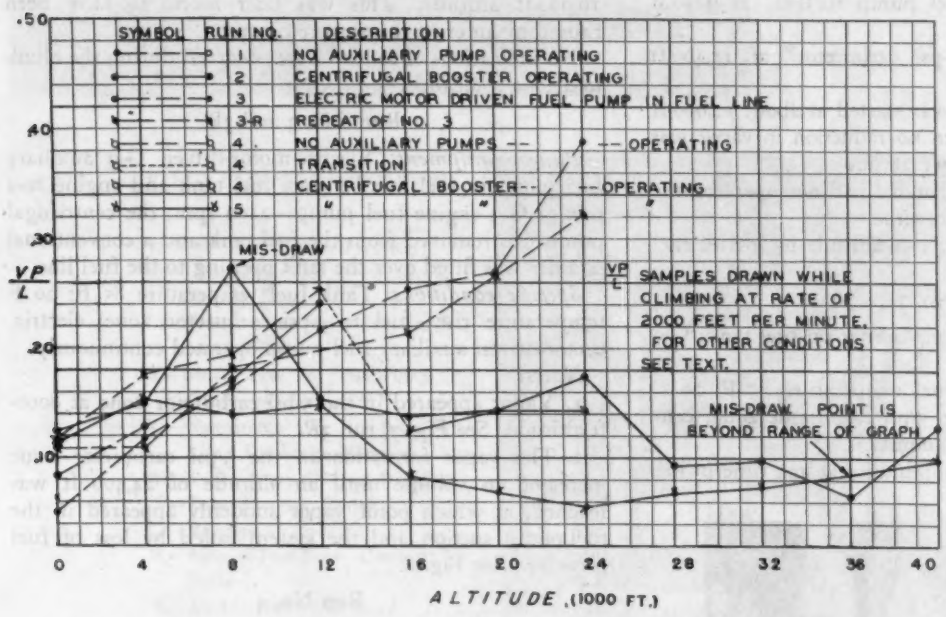
1. Difficulty was experienced in controlling temperature rise after passing the altitude of 16,000 ft because of the drop of tank fuel temperature caused by fuel boiling.



■ Fig. 7 – First vapor in run No. 3R – 8000-ft altitude



■ Fig. 8 – Vapor condition at failure – run No. 3R



■ Fig. 9 – Combination of results

2. No vapor was observed in any of the three observation zones at any time during the climb to 40,000 ft.

3. There were no indications of failure.

■ Vapor-Potential Samples

Vapor-potential samples were drawn during each of the five test runs and the resulting VP/*L* measured. The combined results appear in Fig. 9. Individual samples as drawn are shown in Figs. 10-14, inclusive.

■ Conclusion

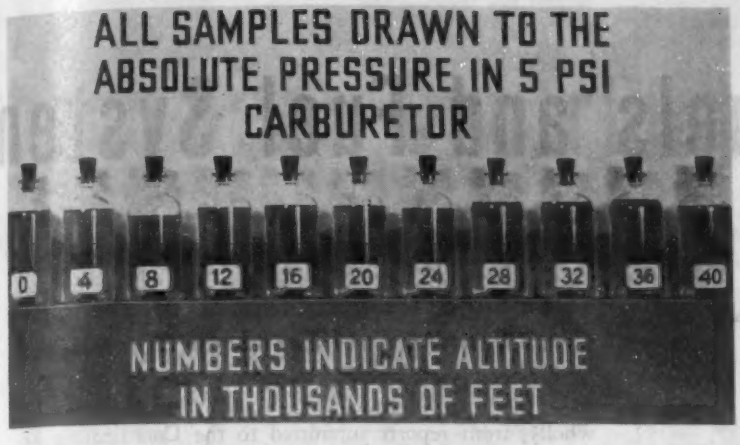
The operation of a centrifugal fuel-booster pump properly installed will reduce the vapor potential of the fuel in the system on the discharge side of the booster.

The use of an electric-motor-driven fuel pump at any point in the fuel line between fuel tank and engine fuel pump has no effect upon the altitude at which vapor appears in the carburetor, neither does it reduce its volume.

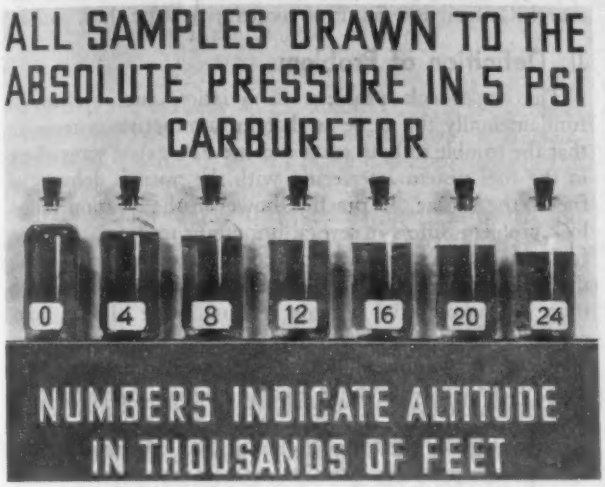
The effect of centrifugal booster operation in run No. 4 is striking in its ability to eliminate vapor once having been formed notwithstanding there having been no pause in the rate of climb.

The VP/*L* samples, in general, confirm the data obtained by direct observation of the system. Refinements in procedure and additional tests should permit the develop-

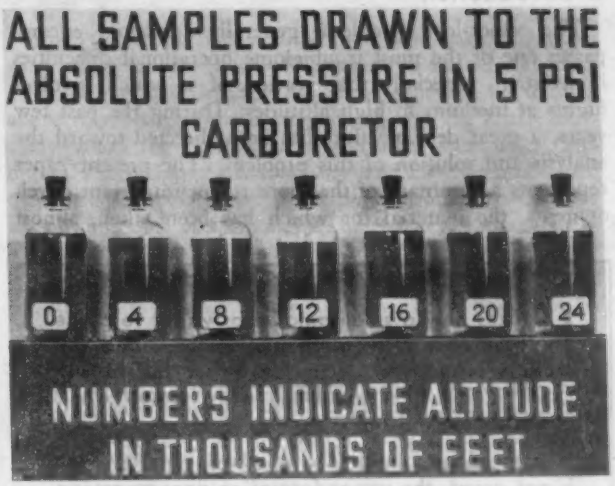
■ Fig. 10 – Run No. 1 – VP/L samples


ALL SAMPLES DRAWN TO THE ABSOLUTE PRESSURE IN 5 PSI CARBURETOR
0 4 8 12 16 20 24 28 32 36 40
NUMBERS INDICATE ALTITUDE IN THOUSANDS OF FEET

■ Fig. 11 – Run No. 2 – VP/L samples


ALL SAMPLES DRAWN TO THE ABSOLUTE PRESSURE IN 5 PSI CARBURETOR
0 4 8 12 16 20 24
NUMBERS INDICATE ALTITUDE IN THOUSANDS OF FEET

■ Fig. 12 – Run No. 3 – VP/L samples


ALL SAMPLES DRAWN TO THE ABSOLUTE PRESSURE IN 5 PSI CARBURETOR
0 4 8 12 16 20 24
NUMBERS INDICATE ALTITUDE IN THOUSANDS OF FEET

■ Fig. 13 – Run No. 4 – VP/L samples

ment of evaluation factors that will increase the usefulness of this technique.

The possibility of vapor in detrimental quantities being formed in fuel systems at sea level is suggested by the VP/*L* samples.

The reduction of vapor by centrifugal booster operation has been confirmed by flight test.

■ Acknowledgment

We wish to acknowledge the cooperation of C. D. Curtis, powerplant design engineer, Vega Aircraft Corp., and F. G. Bollo, research engineer, Shell Development Corp.

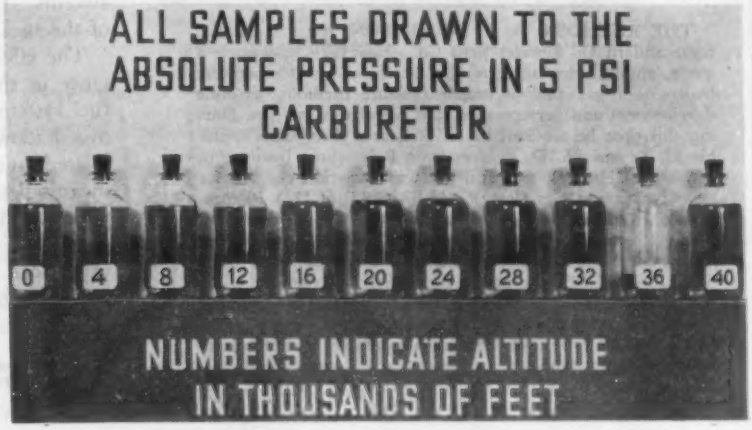
References

1. "Seventh Progress Report – Evaluation of *V/L* Ratio," Dec. 15, 1941, by the Group Working on Vapor-Lock Projects of the CFR Aviation Fuels Division.

See this report for a definition of *V/L* ratio (in equilibrium) as distinguished from the VP/*L* ratio (not in equilibrium) used in this paper.

2. "Fuel Feed at High Altitude," by W. H. Curtis and R. R. Curtis, SAE Transactions, Vol. 50, August, 1942, pp. 321-337.

See this article for more complete description of laboratory simulation methods for aircraft fuel systems.


ALL SAMPLES DRAWN TO THE ABSOLUTE PRESSURE IN 5 PSI CARBURETOR
0 4 8 12 16 20 24 28 32 36 40
NUMBERS INDICATE ALTITUDE IN THOUSANDS OF FEET

■ Fig. 14 – Run No. 5 – VP/L samples

Studies of Fuels and Fuel System Ph

by A. E. ROBERTSON* and R. ALBRIGHT**

(Compiled from a Report of the CRC Coordinating Fuel Research Committee)

I. Introduction

The vapor-lock problem, generally speaking, encompasses one of the most troublesome operational difficulties in aviation, especially that portion of it which involves flights at medium to high altitudes. During the past few years, a great deal of effort has been directed toward the analysis and solution of this problem. The present paper represents a summary of the more recent important developments, the material for which has been taken, almost wholly, from reports submitted to the Coordinating Research Council as a result of cooperative vapor-lock studies.

IT takes only a small amount of vapor to cause vapor lock in aviation fuel systems, and this trouble may even occur far below the boiling point of the fuel, especially when dissolved air is present in the fuel.

Although the solubility of air in aviation fuel is not great, the vapor formation accompanying its evolution may be out of all proportion to the amount present.

The equilibrium amount of vapor that a gasoline will form for given initial and final conditions of temperature and pressure may be predicted by means of an equation presented herein by the authors.

With a knowledge of the temperature and pressure characteristics of a given fuel system, the vapor-liquid conditions existing throughout the system can be estimated in those cases where the initial condition of the fuel with respect to air saturation is known.

■

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II. Definition of Problem

The vapor-lock problem in aviation fuel systems is fundamentally the same as that in automotive systems, in that the trouble in both cases is caused by vapor somewhere in the fuel system interfering with the normal delivery of fuel to the engine. In practice, however, the aviation vapor-lock problem differs in several important respects. Aviation fuel systems are infinitely more complicated, with the result that they are, in general, more critical to vapor formation than are automotive systems. Thus, while simple automotive systems will usually handle from 20 to 40 volumes of vapor per volume of liquid V/L, aviation systems may vapor lock when the vapor to liquid volume ratio is unity, or less. In addition, the fact that the aviation fuel system is called upon to operate at high altitudes and resulting low atmospheric pressures greatly increases the vapor-forming tendency of the fuel.

III. Fuel Characteristics Related to Vapor Lock

The only fuel characteristic which is important from the standpoint of vapor lock is its vapor-forming tendency. In the case of automotive fuel systems, a combination of the vapor pressure plus the character of the front end of the ASTM distillation curve is commonly used as the criterion of vapor-lock tendency. The case of aviation fuel systems is different in that the amount of vapor required to cause trouble is relatively so small, that the influence of the front end of the distillation curve can be completely ignored. On the other hand, there is one factor which can ordinarily be ignored in the case of automotive systems, but which assumes great importance in the case of aviation systems. This is the effect of dissolved air on the tendency of the fuel to form vapor.

The effect of air dissolved in the fuel is of such importance in the consideration of vapor formation in aviation fuel systems of low vapor-handling capacity that it almost overshadows vapor pressure. The two are actually interrelated, and any consideration of one without the other is meaningless from the standpoint of analyzing aviation vapor lock. It follows, therefore, that from the fuel standpoint, the principal considerations in a study of aviation vapor lock are: (a) the vapor pressure of the fuel, (b) the potential solubility of air in the fuel, and (c) the amount of air actually dissolved in the fuel.

[This paper was presented at the SAE War Engineering – Annual Meeting, Detroit, Mich., Jan. 14, 1944.]

* Standard Oil Development Co.
** Socony-Vacuum Oil Co.

Phenomena Contributing to AVIATION VAPOR LOCK

A. The Solubility of Air in Gasoline – The solubility of a gas in a liquid is, in general, proportional to the partial pressure of the gas at the liquid surface, or:

$$S = KP_p \qquad (1)$$

where S is the amount of gas absorbed, K is the proportionality factor, or absorption coefficient which is characteristic of the liquid, and P_p is the partial pressure of the gas at the liquid surface.

It is important to note that the amount of air which will dissolve in gasoline decreases as the vapor pressure increases. This is due to the fact that the partial pressure of air at the liquid surface is the total pressure minus the vapor pressure. Thus, a gasoline which is saturated with air at one temperature and pressure will give up air if the temperature is raised, not because of any change in the absorption coefficient K, but because the vapor pressure of the gasoline increases, causing a decrease in the partial pressure of air at the liquid surface.

Since air is a mixture of gases and not a pure substance, its absorption coefficient varies somewhat, depending on conditions, and hence cannot be considered a true constant. The variations, however, are not large enough to be of significance for the present report.

1. *The Prediction of Air Solubility Coefficients* – The solubility of air in a number of gasolines was investigated rather thoroughly by Bridgeman and Aldrich,[1] who were able to correlate the absorption coefficient K with the density of the gasoline according to the following relationship:

$$K = 0.2116\, d(1 - 1.125d) \qquad (2)$$

where K represents the volume of air (corrected to 32 F and 760 mm pressure) which will be absorbed by 100 volumes of gasoline (corrected to 60/60 F) for each mm partial pressure of air, and d is the density of the gasoline at 60 F.

Since the Bridgeman-Aldrich data were based on motor gasolines of a type quite different from modern aviation gasolines, it was desirable to know whether or not the above relationship would be applicable to the latter, especially when they contained appreciable amounts of aromatics. Accordingly, air solubility data have been obtained on a number of modern aviation gasolines containing various amounts of aromatics. These data are presented in Table 1.

The data in Table 1 indicate that the Bridgeman-Aldrich equation is satisfactory for estimating the solubility of air in regular aviation gasolines containing up to 20% aromatics. For higher aromatic contents, the predicted values are low, but not excessively so up to 40% aromatics. Since the aromatic content of modern aviation gasolines is limited to about 20% by a heating value specification, it is concluded that the Bridgeman-Aldrich equation will satisfactorily predict their air absorption coefficients.

Bridgeman and Aldrich concluded from their data that in the temperature range between −20 and 40 C temperature has no effect on the absorption coefficient K. On the other hand, the data from company A in Table 1 indicate that temperature does affect the K value. Since other literature data indicate that the effect of temperature may be either positive or negative, depending on the hydrocarbon compound in question, it may be that the opposing trends happened to cancel one another in the case of the Bridgeman-Aldrich data. Since the data indicating that temperature has an effect on K are sketchy, it has been assumed to be independent of temperature in the present paper.

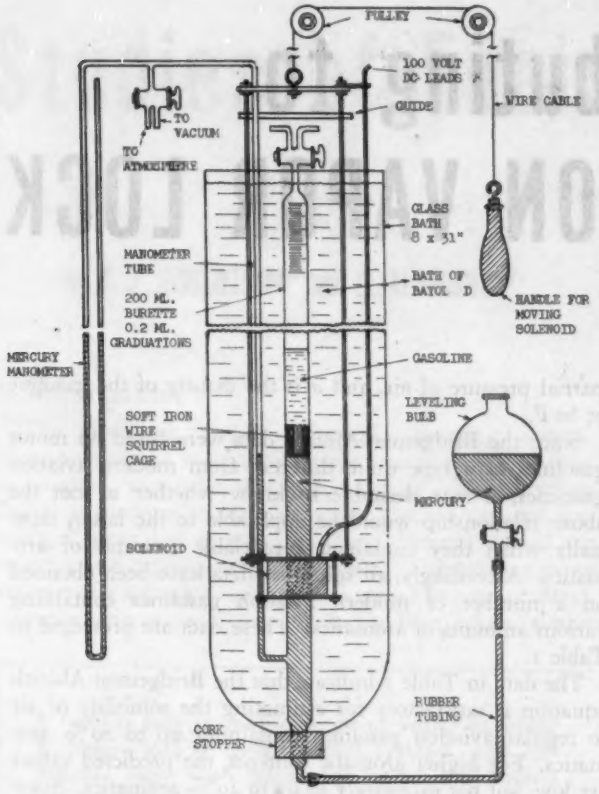
2. *The Experimental Determination of the Solubility of Air in Fuel* – The solubility of air in gasoline may be determined experimentally in a number of ways. The Aviation Fuels Division, in cooperation with the Motor

[1] See *SAE Journal*, Vol. 27, July, 1930, pp. 93-106: "The Properties of Gasolines with Reference to Vapor Lock," by O. C. Bridgeman and E. W. Aldrich.

Table 1 – Solubility of Air in Aviation Gasolines

Fuel Identification	Nature of Fuel	Source of Data	Gravity, deg API	Temperature of Test, F	*K* Values*		
					Found	Bridgeman and Aldrich Predicted	From Literature
L-5151A	Paraffinic	Company A	67.8	77	0.0297	0.0302	.....
PPF-813	40% Aromatics in L-5151A	Company A	51.4	33	0.0212	0.0211	.....
PPF-813	40% Aromatics in L-5151A	Company A	51.4	80	0.0239	0.0211	.....
PPF-813J	40% Aromatics in Paraffinic Base	Company B	51.4	95	0.0236	0.0211	.....
50% PPF-813J in Virgin Aviation Gasoline	20% Aromatics in Paraffinic Base	Company B	57.8	100	0.0268	0.0265	.....
Virgin Aviation Gasoline	Paraffinic	Company B	64.3	100	0.0292	0.0284	.....
C. P. Benzene	Aromatic	Company B	28.6	100	0.0157	0.0009	0.0155

*K is the pressure coefficient of air absorption. When multiplied by the partial pressure of air in mm of mercury, it gives the volume of air (corrected to 32 F and 760 mm pressure) which will dissolve in 100 volumes of gasoline (corrected to 60 F/60 F).



■ Fig. 1 – Apparatus used to determine solubility of air in gasoline

Fuels Division of the Coordinating Fuels Research Committee, is in the process of setting up a standardized apparatus and procedure for studying this and other properties of fuels. While the details of the method are not standardized as yet, it is identical in principle with the method used in obtaining the data reported by company B in Table 1. This method is briefly described.

The apparatus used in obtaining air solubility data is shown schematically in Fig. 1. It consists of a calibrated burette, with mercury sealing liquid, in a constant-temperature bath. The temperature control devices are omitted from Fig. 1 for the sake of simplicity. The burette has an internal plunger in the form of an iron wire "squirrel cage" which can be moved up and down inside the burette by means of an electromagnet, which is moved up and down outside the burette. This plunger is used to obtain rapid equilibrium between the vapor and liquid phases.

The manometer arrangement shown in Fig. 1 was arranged to facilitate the reading of pressures. A suitably mounted cathetometer is recommended to replace the arrangement shown.

To obtain air solubility data on a gasoline, the sample containing dissolved air is introduced into the burette and confined over the mercury with zero vapor space at atmospheric pressure. The pressure on the sample is then reduced, by suitable manipulation of the mercury leveling bulb, until a vapor space is formed. The sample is then agitated by means of the plunger, until constant readings of volume and pressure indicate that equilibrium has been established. The pressure at the gasoline liquid surface is then determined by noting the difference between the mercury levels in the burette and the leveling bulb, and correcting for the head of liquid gasoline.

The following equation may be used for computing the pressure at the gasoline liquid surface:

p = Barometric pressure — difference in level between the mercury in the burette and that in the leveling bulb + the height of the column of gasoline in the burette × density of gasoline/13.6

The volumes occupied by the liquid gasoline and the vapor are noted and the vapor to liquid volume ratio, or V/L, calculated. Similar sets of readings are now taken at two other pressures, the maximum vapor volume being in no case more than half that of the gasoline volume. The three readings are taken at constant temperature.

The solubility coefficient of air in the gasoline, the vapor pressure of the gasoline at the test temperature, and the amount of air actually dissolved in the gasoline may now be calculated. The equation for the solubility of air in gasoline may be written as follows:

$$S = K\,(P - P_g) \tag{4}$$

where S = Volumes of air measured at 32 F and 760 mm of mercury dissolved in 100 volumes of gasoline measured at 60 F

P = Total pressure on surface of liquid gasoline, mm of mercury

P_g = Vapor pressure of liquid gasoline, mm of mercury

K = Solubility coefficient.

The amount of air that will come out of solution as the pressure is lowered is the difference between the amount of air in the sample originally and the amount soluble in the sample at the reduced pressure. This may be expressed as follows:

$$\Delta S = S_1 - S_2 \tag{5}$$

where

ΔS = Amount of air in the vapor space

S_1 = Amount of air in the original sample

S_2 = Amount of air soluble in the gasoline at the reduced pressure

For any set of test data where equilibrium is attained, S_2 of equation (5) is the same as S of equation (4). Substituting:

$$\Delta S = S_1 - K\,(P - P_g) \tag{6}$$

Since the air that comes out of solution will be saturated with gasoline vapor, the volume which the air and vapor will occupy under the test conditions will be:

$$\frac{V}{L} = [S_1 - K\,(P - P_g)] \times \frac{760}{P - P_g} \times \frac{t + 460}{492} \times \frac{1}{100\,[1 + 0.0006\,(t - 60)]} \tag{7}$$

where

t = Test temperature

$\frac{V}{L}$ = Observed vapor to liquid volume ratio

(In previous publications, V/L has been defined as the vapor volume at test temperature and test pressure divided by the original liquid volume at 32 F. In the present work, however, V/L is defined as the vapor volume at test temperature and pressure divided by the actual liquid volume at test temperature.)

An inspection of equation (7) shows that there are three unknowns, when a set of test data are substituted, these are S, P_g, and K. By substituting the three sets of test data in this equation, three equations will be obtained. By the simultaneous solution of these equations, the three unknowns may be evaluated.

The use of equation (7) involves the following assumptions:

(a) That the solubility coefficient K is constant and independent of temperature.

(b) That the vapor pressure of the gasoline does not vary with V/L. (This is approximately true for the small V/L values involved.)

B. The Effect of Dissolved Air and Vapor Pressure on the Vapor-Forming Tendency of Aviation Fuels – While the amount of air which will dissolve in gasoline is small, seldom being greater than 20 volumes of air per 100 volumes of liquid, it is very important from the standpoint of aviation vapor lock. This is true because there are indications that extremely small amounts of vapor in some parts of the fuel system may cause irregular engine operation and, in addition, the dissolved air on being disengaged may carry with it large volumes of vapor, so that it exerts an influence out of all proportion to the amount present.

The effect of dissolved air on the vapor-forming tendency of a gasoline may likewise be investigated experimentally by the use of an apparatus such as that shown in Fig. 1. It has been found, however, that by the use of an equation similar in form to equation (7), the vapor which a given gasoline will form can be calculated with considerable precision for any desired condition of air saturation and desorption. For instance, if a gasoline has been saturated with air at a known temperature and pressure and the temperature is raised and the pressure is held constant, the amount of vapor which will be evolved can be computed from the following equation:

$$\frac{V}{L} = K(P_{g_2} - P_{g_1}) \frac{760}{P - P_{g_2}} \times \frac{t_2 + 460}{492} \times \frac{1}{100\,[1 + 0.0006\,(t_2 - 60)]} \qquad (8)$$

where the subscripts 1 and 2 indicate values at saturation and test conditions respectively, while the remainder of the symbols have the same significance as in previous equations.

The relative amounts of air and vapor in the vapor space may readily be calculated from the relationship:

$$\text{Per Cent of Air} = 100 \times P_1 - P_{g_2}/P.$$

If a gasoline has been saturated with air at a known temperature and pressure and the pressure is lowered while the temperature is held constant, the amount of vapor which will be evolved can be calculated from the following equation:

$$\frac{V}{L} = K(P_1 - P_2) \frac{760}{P_2 - P_g} \times \frac{t + 460}{492} \times \frac{1}{100\,[1 + 0.0006\,(t - 60)]} \qquad (9)$$

The relative amounts of air and vapor in the vapor space may be calculated from the relationship: Per Cent of Air $= 100 \times P_2 - P_g/P_2$.

If a gasoline is saturated with air at a known temperature and pressure and both the temperature and pressure are changed, then the amount of vapor which will be formed may be predicted by use of the following equation:

$$\frac{V}{L} = K\,(P_1 - P_2 + P_{g_2} - P_{g_1}) \frac{760}{P_2 - P_{g_2}} \times \frac{t_2 + 460}{492} \times \frac{1}{100\,[1 + 0.0006\,(t - 60)]} \qquad (10)$$

The relative amounts of air and vapor in the vapor space may be calculated from the relationship:

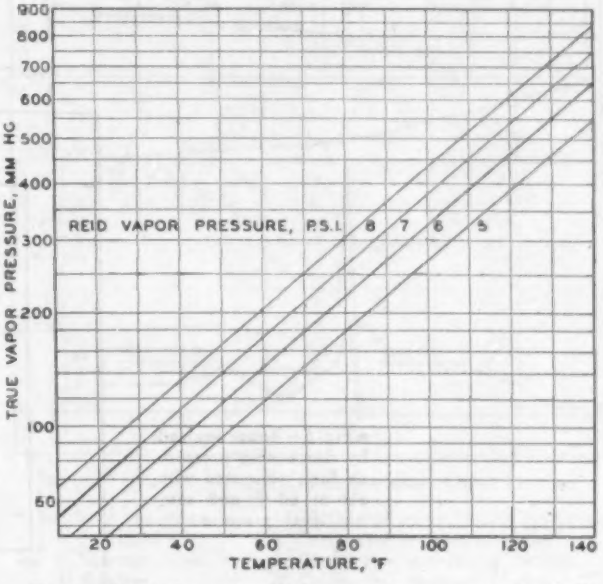
$$\text{Per Cent of Air} = 100 \times P_2 - P_{g_2}/P_2.$$

In applying equations (8), (9), and (10), K values may be calculated by means of equation (2). Where it is necessary to predict vapor pressures of gasolines at temperatures other than that at which the vapor pressure determination was made, this may be done with sufficient precision by the use of a vapor-pressure extrapolation chart, such as that shown in Fig. 2.

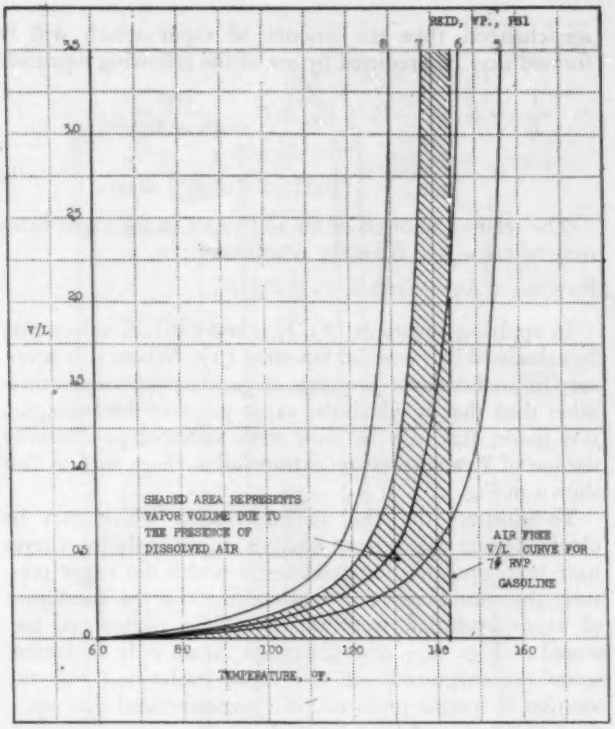
To illustrate the kind of information which may be obtained from this type of analysis, vapor-evolution curves have been prepared for gasolines in which the vapor pressure, the conditions of air saturation, and the conditions of vapor evolution were varied. These curves are presented in Figs. 3, 4, and 5, and are meant only to demonstrate the magnitude of the vapor evolution. For the solution of specific problems, it is recommended that equations (8), (9), and (10) be used.

The effect of increasing temperature on the vapor evolved by gasolines of 3 to 7 psi Rvp which have been saturated with air at 60 F and sea-level pressure, is presented in Fig. 3. These curves were calculated by the use of equation 8. To demonstrate the vapor which is formed because of the presence of dissolved air, the air-free V/L curve for the 7-psi Rvp gasoline has been included. The cross-hatched area between the two curves represents air and vapor which is formed by the air-saturated sample in excess of that which would be formed by the air-free sample.

The effect of varying the temperature of air saturation on the vapor evolved by a 7-psi Rvp gasoline when the



■ Fig. 2 – Chart for extrapolating vapor pressures of aviation gasolines


REID, VP., PSI
V/L
SHADED AREA REPRESENTS VAPOR VOLUME DUE TO THE PRESENCE OF DISSOLVED AIR
AIR FREE V/L CURVE FOR 7# RVP GASOLINE
TEMPERATURE, °F.

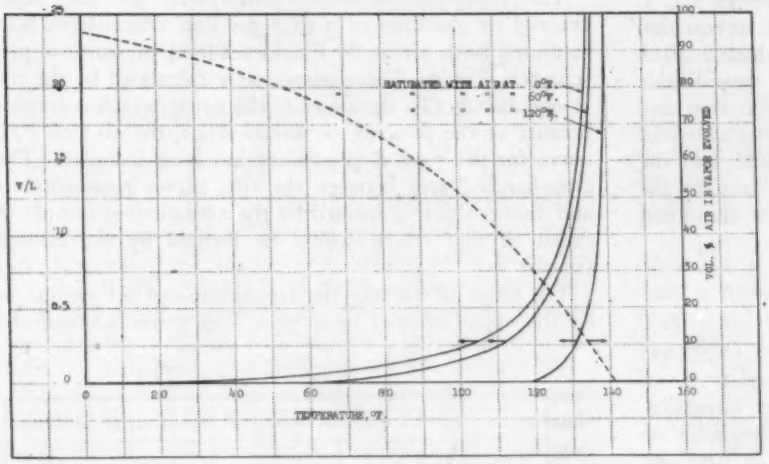
■ Fig. 3 – Vapor evolved on raising temperature of gasoline saturated with air at 60 F

temperature is raised, is illustrated in Fig. 4. A curve showing the amount of air in the vapor phase is included in this figure.

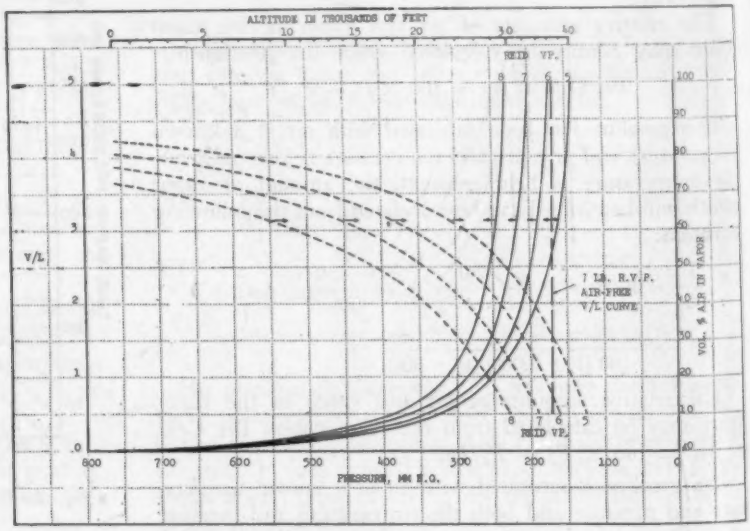
The vapor evolved on decreasing the pressure on 5- to 8-psi Rvp gasolines, which have been saturated with air at 60 F and sea-level pressure, is shown in Fig. 5. Curves showing the concentration of air in the vapor phase, and 7-psi Rvp air-free V/L values are included in this figure.

C. *Nomograph for Estimating V/L Values* – In order to facilitate the estimation of V/L values for 7-psi aviation gasoline, the nomograph presented in Fig. 6 has been prepared. This nomograph is based on equation (10), but the V/L values have been adjusted to compensate for the normal decrease in vapor pressure with increasing V/L. The error caused by this effect becomes appreciable at V/L values above 1.5, and can only be corrected by an arduous process of trial and error. This nomograph is particularly useful for estimating V/L values in fuel systems where both temperatures and pressures are changing.

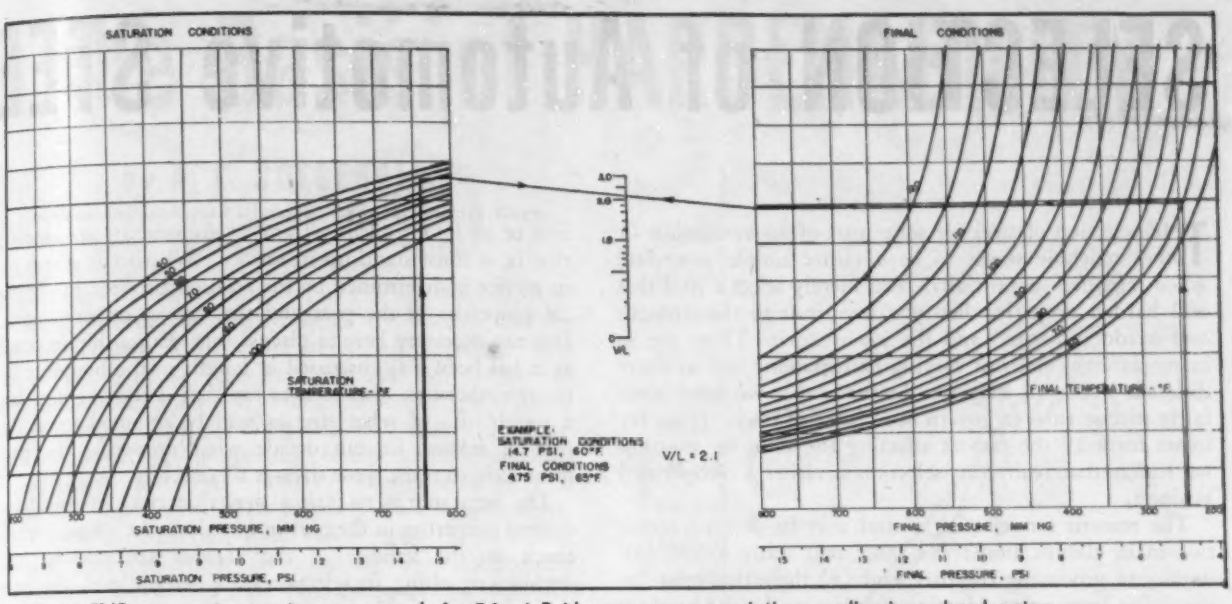
The values obtained from this nomograph will agree, within plus or minus 3%, with those obtained from equation (10), after the latter values have been corrected for the normal change of vapor pressure with V/L.



■ Fig. 4 – Effect of varying temperature of air saturation on vapor evolved by 7-psi gasoline



■ Fig. 5 – Vapor evolved by decreasing pressure on fuels saturated with air at 60 F and sea-level pressure



■ Fig. 6 – V/L-pressure-temperature nomograph for 7.0-psi Reid vapor pressure aviation gasoline in a closed system

IV. Estimation of Vapor Formation

The application of the V/L relationships developed above to actual fuel systems can best be illustrated by means of an example. A simplified aviation fuel system is shown in Fig. 7. The pressures and temperatures assumed to exist throughout this system at sea level and at 18,000-ft altitude are included in this figure. The fuel charged to the airplane tank is assumed to be a 7-psi Rvp gasoline saturated with air at sea-level pressure and at 60 F. It is further assumed that no change in the fuel-tank temperature takes place due to changing outside air temperature.

For sea-level operation, the fuel saturation conditions are, therefore, 60 F, and 14.7 psi absolute. The final conditions are the fuel pressures and temperatures at any point in the system indicated in Fig. 7. Amounts of vapor formed in the fuel system for the conditions as determined by use of the V/L nomograph (Fig. 6) are also included in Fig. 7. It can be seen that the greatest amount of vapor forms at the fuel-pump inlet, but even here it is too small to be of significance (approximately 0.1 V/L).

In predicting the vapor formation in the fuel system of a plane operating at 18,000-ft altitude, the initial condition of the fuel is not completely predictable. This is due to the fact that the fuel in the tank, if insufficiently agitated, will not evolve air fast enough to maintain equilibrium between the vapor and liquid phases as the pressure is reduced, and a condition of supersaturation results. For this reason in one case it has been assumed that no vapor is involved during the climb, a condition which may exist in fuel tanks having no agitation, and representing an extreme case of supersaturation. In another case, it has been assumed that no supersaturation existed, a condition which corresponds to the continuous equilibrium evolution of air and incidental vapor as the pressure decreases during the climb.

The amounts of vapor which would be involved in the fuel system at 18,000 ft under these two extreme conditions of air saturation are indicated in Fig. 7. These data show that the greatest amount of vapor is formed at the fuel-pump inlet and corresponds to V/L ratios of 3 and 1.1 for maximum and minimum supersaturation, respectively. It is interesting to note that, with maximum supersaturation at 18,000 ft, a slight amount of vapor forms in the carburetor metered section, transfer line, and nozzle valve (approximately 0.1 V/L).

The foregoing example is intended merely to serve as a

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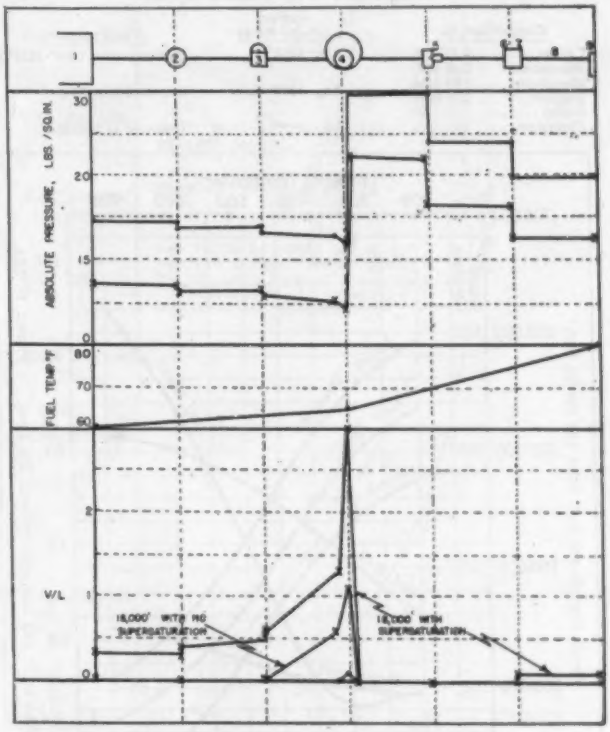


■ Fig. 7 – Fuel system V/L-pressure-temperature diagram

● Sea-level operation
X Operation at 18,000-ft altitude

1. Fuel tank
2. Selector valve
3. Strainer
4. Engine fuel pump
5 Carburetor strainer chamber
6. Carburetor unmetered section
7. Carburetor metered section
8. Transfer line
9. Nozzle valve

SELECTION of Automotive STEEL

THE selection of steel for some part of an automobile or other machine seems to be a rather simple procedure when regarded superficially. You merely select a steel that will harden properly, then you temper it to the strength and hardness desired and the job is done. There are so many different steels of varying hardenability and so many different sizes and shapes to be made that we need some fairly precise rules to govern our choice of steels. If we try to set forth all the factors affecting the ultimate selection we realize that really this selection is rather a complicated business.

The reasons for selecting a steel may be divided under two main classifications: (1) those that relate to the economics of producing the part, and (2) those that relate to suitability from an engineering standpoint. Under heading number one come such things as: A. Annealing characteristics. B. Machinability. C. Ease of manufacture at the mill. D. Ease of control within quality and composition limits. E. Relation to other steels in use, and F. Cost.

Regardless of economic aspects, the steel selected must first of all be satisfactory from an engineering standpoint; that is, it must stand up in service. Whether it stands up in service is determined by the relation between the physical properties of the part and the stresses imposed on it. It is not necessary here to discuss this relationship in detail as it has been ably discussed at length in the literature by many authorities. Suffice it to say that if the engineer has a rough idea of what stresses will be imposed upon the part in service, he can decide what properties the part must have to resist those stresses successfully.

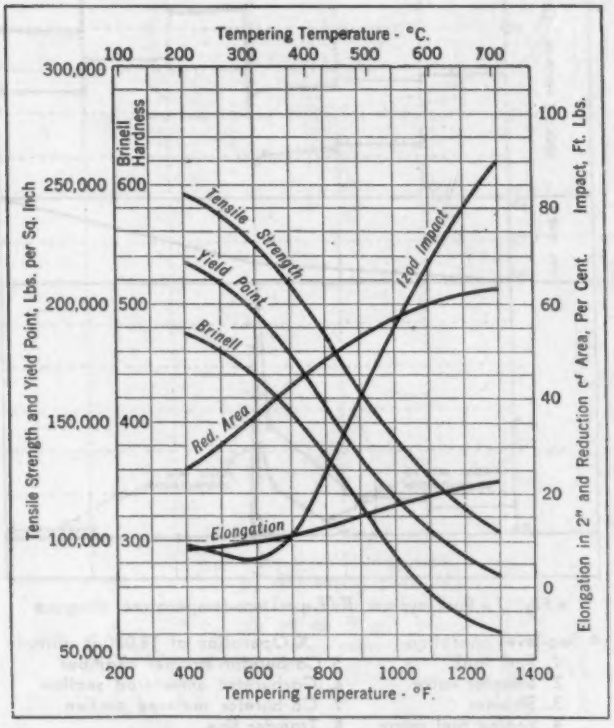
The next step is to pick a steel that can develop those desired properties in the part in question. As a final overall check on the validity of the various assumptions and decisions resulting in selection of a certain steel, the part is service tested. Then, and only then, can the success of the part be determined.

In selecting a steel on the basis of its potential physical properties, it used to be customary to refer to physical property charts. All steel producers and alloy companies publish such physical property charts of the type shown in Fig. 1, representing the effect of tempering temperature after hardening on tensile strength, yield point, reduction of area, per cent elongation and hardness. The data for these charts were obtained by testing tensile test specimens cut from 1-in. diameter bars that had been hardened and subjected to various tempering treatments. Some steels harden only part way through a 1-in. diameter bar. The outside ¼ in. may harden fully but the center ½ in. may be only partially hardened; therefore, the physical properties in a test bar cut from this center ½ in. section would not represent the properties obtained in the properly hardened outer ¼ in. layer. Physical property charts that have been prepared from data obtained on incompletely hardened test bars can be detected by the low hardness values shown for low tempering temperatures. For example, a 0.35% carbon alloy steel should harden to at least 48 Rockwell C to be fully hardened, so the hardness after tempering at 400 F should be that high since 400 F will not reduce the original hardness. When physical property charts for a 0.35% carbon steel show 40 Rockwell

[This paper was presented at a meeting of the SAE Metropolitan Section, New York City, June 1, 1944.]

Nickel-Chromium Steel, S.A.E. No. 3140, Oil Quenched
(In small sizes, ½" to 1½" diameter or thickness)

Composition, %		Approx. Critical Range		Heat Treatment
Carbon	0.35-0.45	Ac_1	1355° F.	Quenched from 1475°-1525° F.
Manganese	0.60-0.90			
Phosphorus	0.04 Max.			
Sulphur	0.05 Max.	Ac_2	1380° F.	Quenched into Oil
Nickel	1.00-1.50			
Chromium	0.45-0.75	Ac_3	1395° F.	Tempered as Indicated



■ Fig. 1 – Typical physical property chart

WHEN the first paper on the end-quenched hardenability test bar was presented only seven years ago, there was considerable doubt and surprise expressed about the table included, which showed a large variation in hardenability of carburized 4615 steel.

Today, the reasons for most of this variation are understood and are taken as a matter of course. Measures must be taken in utilization of steel to compensate for this variable. As a result, the ability to obtain a uniform product is much greater than it was before the question was understood.

Discussed here by Mr. Boegehold are the prin-

on the Basis of HARDENABILITY

by A. L. BOEGEHOLD

Research Laboratories Division, General Motors Corp.

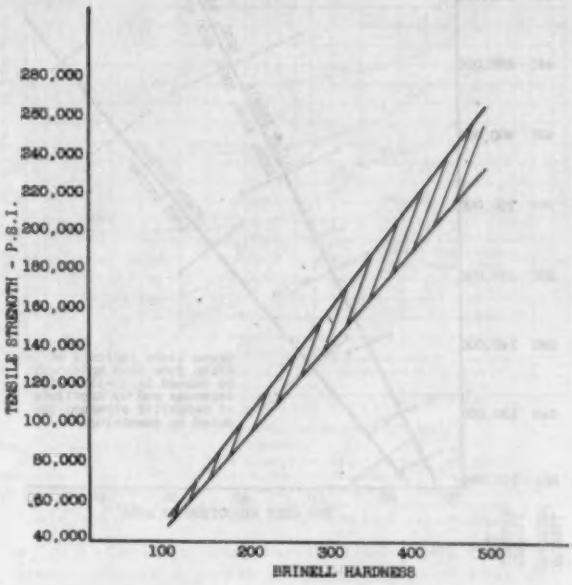
C or thereabouts after a 400 F draw, as some do, it is evidence of incomplete hardening of the test bar and the properties shown in that chart are of doubtful value. The use of inexact data of this kind, obtained without careful regard for the as-quenched hardness of test bars, was partly responsible for the supposition that each steel had a combination of physical properties peculiar to itself and different from other steels. Janitsky[1] showed that physical properties of different steels could be quite closely correlated with hardness and that the composition had little effect on this correlation. From this we have reasoned that it was not necessary to have physical property charts like Fig. 1 for every different type steel since one set of charts would show the relationship for all steels between hardness and: 1. Tensile strength. 2. Yield point. 3. Reduction of area, and 4. Tempering temperature.

A set of such charts was prepared by the writer for the General Motors Metallurgical Committee for use in the GM Standards Book. The data used for preparing these charts were taken from charts similar to Fig. 1 published by Bethlehem Steel Co., Republic Steel Co., U. S. Steel Corp., Climax Molybdenum Co., Timken Steel & Tube Co., International Nickel Co., and from unpublished data from the writer's files. These charts were adopted by the SAE and now appear in the SAE Handbook. More recently, W. Patton[2] has prepared similar charts by collecting physical property information from a number of metallurgical laboratories. The charts prepared from these data show good agreement with the SAE charts. The SAE charts are shown in Figs. 2, 3, 4, and 5.

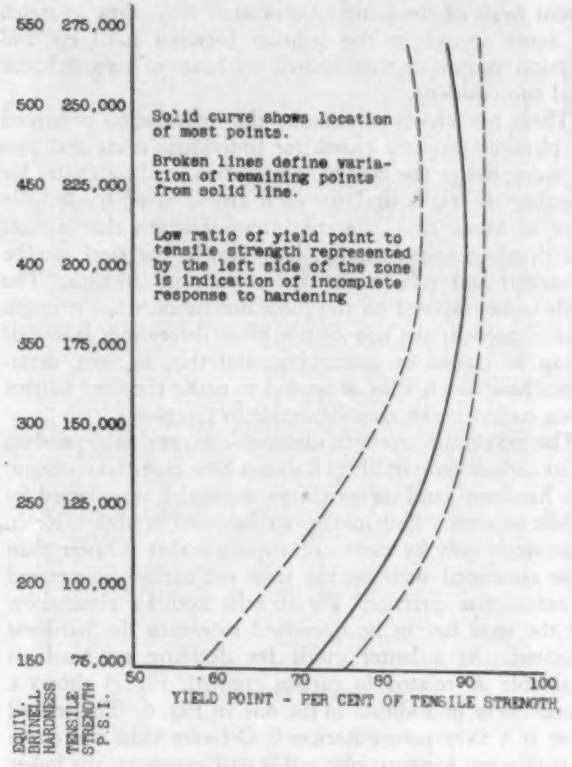
The variation of properties within the bands shown in these charts is not the result of variation in composition. The properties of any composition steel may occur toward the upper or lower boundaries of the band, depending

[1] See "The Marked Similarity in Tensile Properties of Several Heat-Treated SAE Steels," by E. J. Janitsky and M. Baeyertz; ASM Metals Handbook, 1939, pp. 515-518.

[2] See *Metal Progress*, Vol. 43, May, 1943, pp. 726-733: "Mechanical Properties of NE, SAE, and Other Hardened Steels," by W. G. Patton.



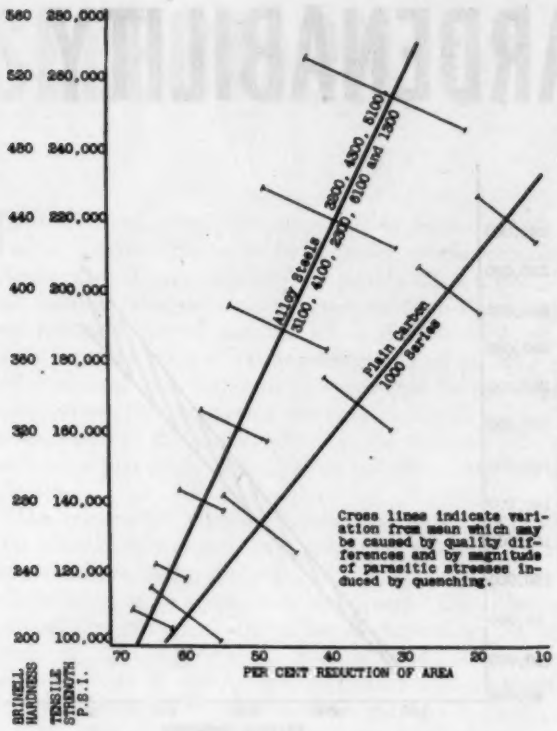
■ Fig. 2 – Relation between hardness and tensile strength – summary of data on carbon and alloy steels including series 1000, 1300, 3100, 3200, 4100, 2300, 4600, 5100, and 6100 in hardened and tempered, as-rolled, annealed, and normalized conditions



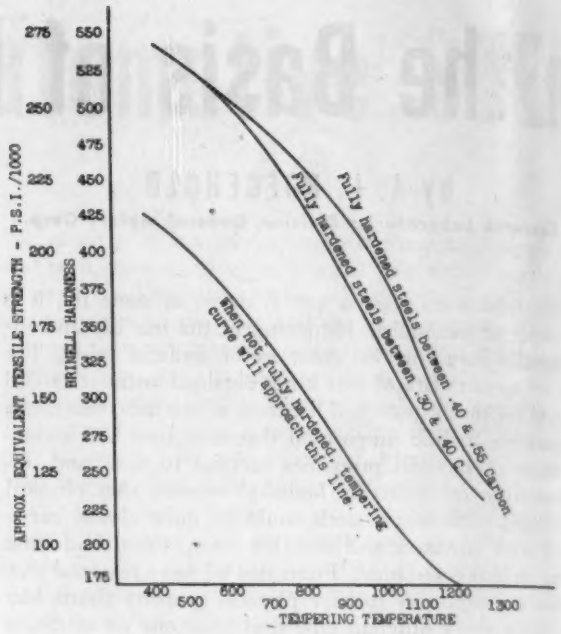
■ Fig. 3 – Relation of tensile strength and hardness with yield point – for 0.30 to 0.50% carbon content of series 1000, X-1300, 1300, 5100, 4100, 4300, 4600, 3100, 3200, 2300, and 6100 steels in the quenched and tempered condition

...ples of selecting steel on the basis of harden-...bility. The author further indicates some cus-...mary procedures as to hardness requirements ... the as-hardened condition to ensure a de-...rable structure after tempering.

THE AUTHOR: A. L. BOEGEHOLD, who is head of ... metallurgy department of General Motors Research ...boratories, is a member of a special committee of the Na...nal Academy of Sciences asked by the former Office of ...oduction Management to investigate and report on possible ...bstitutes for various metals vital to defense. He joined ...f Research Division in 1920 and five years later was as...gned to his present position. Mr. Boegehold, who was ...aduated from Cornell with an M. E. degree in 1915, was ...arded the J. H. Whiting Gold Medal from the American ...undrymen's Association two years ago.


BRINELL HARDNESS
TENSILE STRENGTH P.S.I.
PER CENT REDUCTION OF AREA
Alloy Steels 3100, 4100, 2330, 6100 and 1300
Plain Carbon 1000 Series
Cross lines indicate variation from mean which may be caused by quality differences and by magnitude of parasitic stresses induced by quenching.

■ Fig. 4 – Relation of tensile strength and hardness with reduction of area


APPROX. EQUIVALENT TENSILE STRENGTH - P.S.I./1000
BRINELL HARDNESS
Fully hardened steels between .30 & .40 C
Fully hardened steels between .40 & .55 Carbon
When not fully hardened, tempering curve will approach this line.
TEMPERING TEMPERATURE

■ Fig. 5 – Relation between hardness and tempering temperature – summary of data on carbon and alloy steels including series 1000, 1300, 3100, 3200, 4100, 2300, 4600, 5100, and 6100

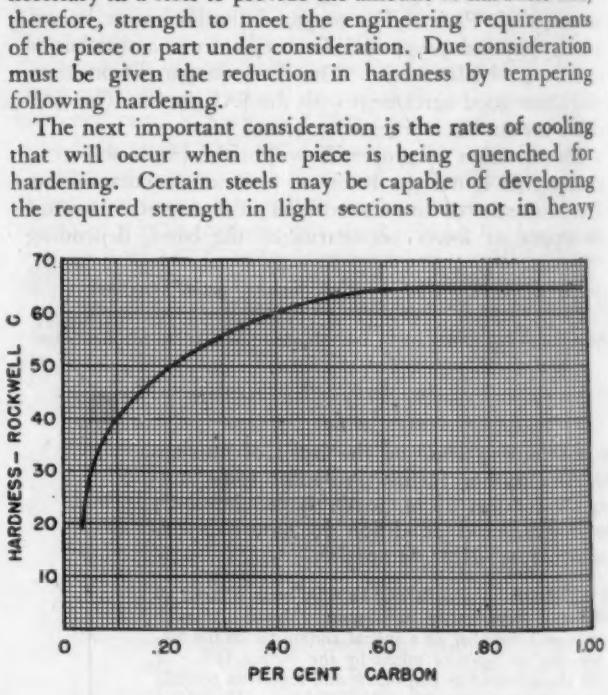
upon the quality of the steel, which is a consequence of the details of procedure in making the steel; that is, different heats of the same type of steel may show as much or more spread in the relation between hardness and physical properties than shown by heats of two different steel compositions.

These few charts summarize the information presented by physical property charts for individual steels and perhaps emphasize the lack of information in these charts for selecting the right steel for each article to be made. We have to know first, the maximum strength that a steel can develop, and secondly, how slowly the steel can be quenched and still obtain that maximum strength. The loads to be imposed on the piece determine what strength steel is needed; the size of the piece determines how fast it can be cooled by quenching, and this, in turn, determines how much alloy is needed to make the steel harden when cooled at the rate obtainable in the piece.

The maximum strength obtainable in steel is dependent on its carbon content. Fig. 6 shows how maximum obtainable hardness (and its corollary – strength) are affected by carbon content. This maximum hardness is obtainable in some steels only by means of a quench that is faster than those associated with section sizes ordinarily encountered in automotive practice. Fig. 6 tells nothing about how fast the steel has to be quenched to obtain the hardness indicated. As a better guide for deciding on hardness obtainable as related to carbon content, Fig. 7 shows a second curve in addition to the one in Fig. 6. The second curve is 5 to 7 points Rockwell C lower than the curve for maximum hardness obtainable and represents the lower limit of hardness obtained when the steel is quenched at 600 F per sec, a cooling speed commonly encountered in the processing of automotive parts, and also, as will be seen later, the cooling rate at the 1/16 in. point on the Jominy hardenability test bar. Hardenability curves for 1400 heats of steel collected by the joint SAE-AISI Hardenability Committee were studied to determine this curve. A third curve represents the lower limit for plain carbon steels at this same cooling rate.

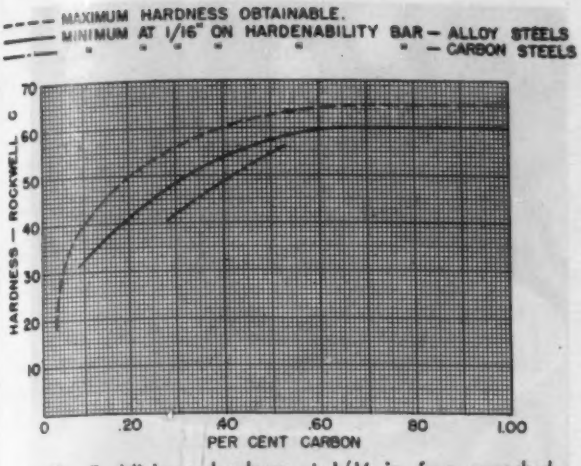
This chart may be used to judge the amount of carbon necessary in a steel to provide the amount of hardness and, therefore, strength to meet the engineering requirements of the piece or part under consideration. Due consideration must be given the reduction in hardness by tempering following hardening.

The next important consideration is the rates of cooling that will occur when the piece is being quenched for hardening. Certain steels may be capable of developing the required strength in light sections but not in heavy



■ Fig. 6 – Maximum hardness obtainable in plain carbon or low-alloy steel[3]

[3] According to ASM Transactions, Vol. 26, March, 1938, p. 14: "Quantitative Hardenability; Proposed Standard Test," by J. L. Burns, T. L. Moore, and R. S. Archer.



■ Fig. 7 – Minimum hardness at 1/16 in. from quenched end of Jominy bar (curves represent minimum values for 90% of 1400 heats; values for 10% of heats fall below curves)

sections. As already mentioned, that cooling rate below which a steel fails to harden depends upon the alloy content and is the means by which hardenability of the steel is classified or graded.

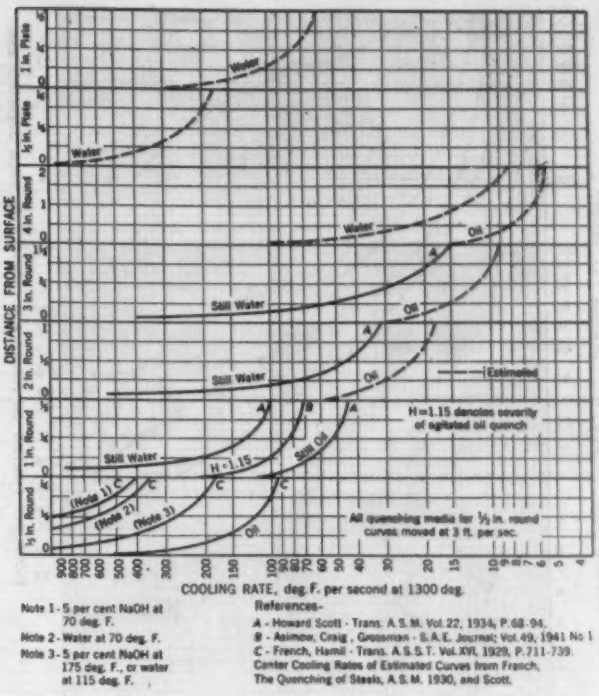
Experienced metallurgists know from long practice about what kind of steel is required to obtain adequate hardening in a certain sized piece. This experience factor, however, is now being reduced to an exact science by the process of collecting information on cooling rates.

Most of the presently available information on cooling rates in different size sections when quenched in oil or water is shown in Fig. 8, which was prepared for the SAE Hardenability Test Committee and now appears in the SAE Standards Book as a recommended procedure for hardenability testing. Another form of this information is shown in Figs. 9 and 10. This method of plotting permits more accurate determination of cooling rates in sizes in between those shown in Fig. 8.

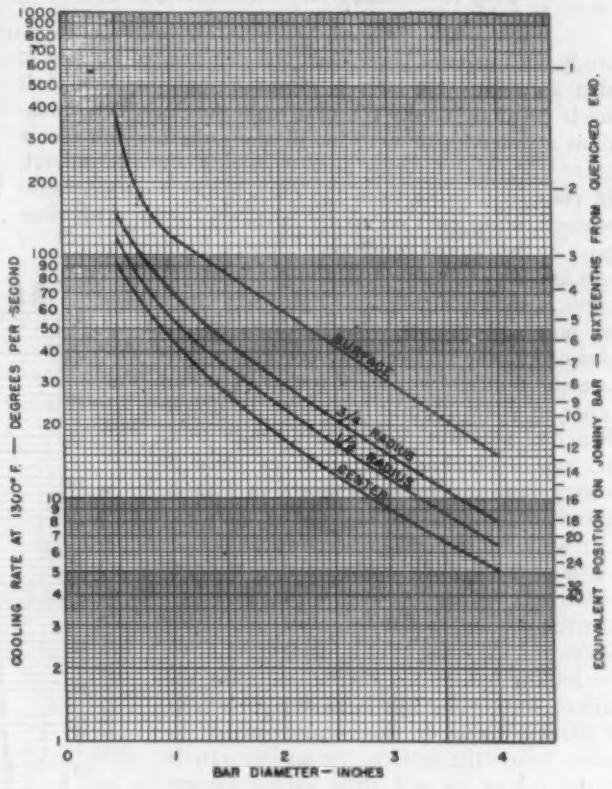
In these charts is depicted the manner in which cooling rates change with distance below the surface of the object quenched. The speed of cooling decreases rapidly from the surface to the center of a bar and is slower at corresponding locations as the bar size increases. The rate of cooling also differs, depending on whether oil or water is used as the quenching medium. We must know, therefore, how every steel reacts when subjected to each one of these cooling rates. This knowledge will permit us to predict how far below the surface of each size bar any steel will harden properly and also what hardnesses will occur at each cooling rate. The range of cooling rates with which we deal ordinarily varies from 10 deg per sec to 1000 deg per sec. The cooling rates at the surface of water-quenched rounds less than 2 in. in diameter are faster than 1000 deg per sec measured at 1300 F. Cooling rates at the center of oil-quenched 3-in. rounds are as low as 10 deg per sec.

To study the hardness produced in steels by quenching at rates of cooling within this wide range, we make a test called the Jominy hardenability test.[4, 5]

This test is made on a cylinder of steel 3 in. long and 1 in. in diameter. It is heated to the proper temperature for hardening and is then quenched on one end only by


DISTANCE FROM SURFACE
1 in. Plate
½ in. Plate
4 in. Round
3 in. Round
2 in. Round
1 in. Round
½ in. Round
Water
Oil
Still Water
Still Oil
Estimated
H=1.15 denotes severity of agitated oil quench
All quenching media for ½ in. round curves moved at 3 ft. per sec.
COOLING RATE, deg. F. per second at 1300 deg.
Note 1 - 5 per cent NaOH at 70 deg. F.
Note 2 - Water at 70 deg. F.
Note 3 - 5 per cent NaOH at 175 deg. F., or water at 115 deg. F.
References-
A - Howard Scott - Trans. A.S.M. Vol. 22, 1934, P. 68-94.
B - Asimow, Craig, Grossman - S.A.E. Journal; Vol. 49, 1941 No. 1
C - French, Hamil - Trans. A.S.S.T. Vol. XVI, 1929, P. 711-739.
Center Cooling Rates of Estimated Curves from French, The Quenching of Steels, A.S.M. 1930, and Scott.

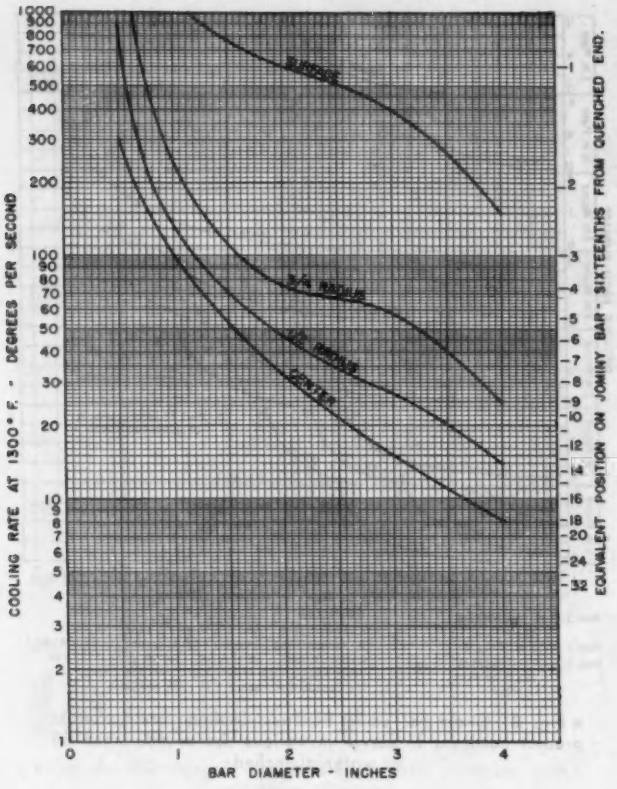
■ Fig. 8 – Chart for using hardness-cooling rate curve to predict hardness *U* curves in various size rounds when oil or water quenched


COOLING RATE AT 1300° F. — DEGREES PER SECOND
EQUIVALENT POSITION ON JOMINY BAR — SIXTEENTHS FROM QUENCHED END
BAR DIAMETER — INCHES
SURFACE
3/4 RADIUS
1/2 RADIUS
CENTER

■ Fig. 9 – Cooling rates for still-oil quench

a stream of water in the manner shown in Fig. 11. This method of quenching produces cooling rates varying from 600 deg per sec at 1/16 in. from the quenched end down to 4 deg per sec at the upper end. We do not need to

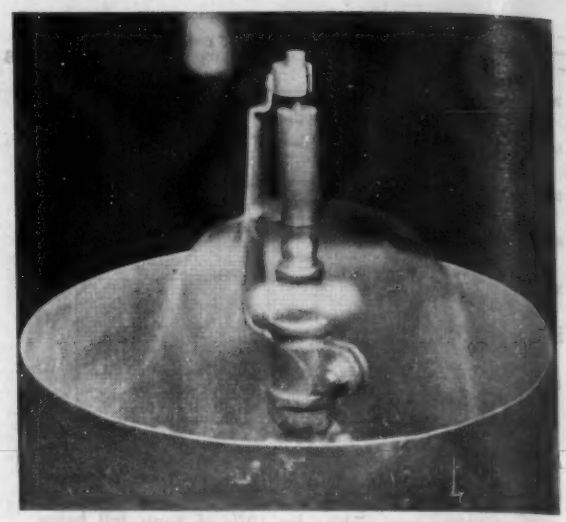
[4] See ASM Transactions, Vol. 26, June, 1938, pp. 574-599 + disc. 599-606: "Hardenability Test for Carburizing Steel," by W. E. Jominy and A. L. Boegehold.
[5] See "SAE Handbook," 1944, pp. 314-324.


COOLING RATE AT 1300° F. - DEGREES PER SECOND
EQUIVALENT POSITION ON JOMINY BAR - SIXTEENTHS FROM QUENCHED END.
SURFACE
1/4 RADIUS
1/2 RADIUS
CENTER
BAR DIAMETER - INCHES

■ Fig. 10 – Cooling rates for still-water quench

study cooling rates faster than 600 deg per sec because at that speed full hardness is obtained with practically all steels currently used. We next make hardness determinations at intervals of 1/16 in. all along the bar and plot a curve of hardness versus distance from the quenched end. In place of distance from the quenched end we may substitute cooling rates occurring at each 1/16 in. This then gives us the desired connection between hardness and cooling rate. A typical hardenability curve is shown in Fig. 12. From this curve we may now predict what hardness can be obtained at the surface and between the surface and center of any sized round quenched either in water or oil. It becomes a simple matter of substituting hardness values from the hardness-cooling rate curve in place of cooling rates in the curves shown in Figs. 8, 9, and 10.

If we are interested in selecting a steel for a 2-in. round shaft to be hardened by quenching in oil, we refer to Figs. 8 or 9 and find that the cooling rate at the center is 18 deg per sec, or the same as at about 11/16 in. on the Jominy bar. At ½ the radius below the surface, the cooling rate is 24 deg per sec, or at about 9/16 in. on the Jominy bar. At ¼ radius below the surface, the cooling rate is 32 deg per sec, or at 8/16 in. on the Jominy bar. At the surface the cooling rate is 60 deg per sec, or at 4½ sixteenths on the Jominy bar. The hardness at these points on the curve shown in Fig. 12 may be substituted to show that a 2-in. round of 5130 steel quenched in oil from 1675 F would have



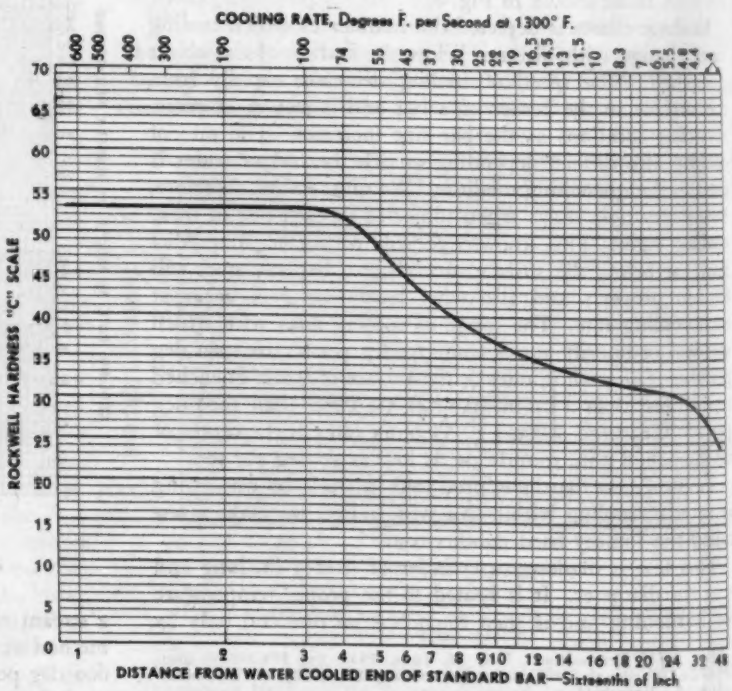
■ Fig. 11 – Manner of quenching for Jominy hardenability test

hardnesses throughout the cross-section as shown in Table 1.

Table 1 – Hardenability of 2-In. Diameter Bar of 5130 Steel (Quenched in Oil)

Location in 2-In. Diameter Bar	Cooling Rate in Oil-Quenched 2-In. Diameter Bar, deg per sec	Corresponding Locations on J Bar	Hardness—Rockwell C for 5130 from Curve in Fig. 12
Center	18	11/16	36
½ radius below surface	24	9/16	38
¼ radius below surface	32	8/16	41
Surface	60	4½/16	50

If the same bar were water quenched, the corresponding cooling rates and hardnesses would be as shown in Table 2.


COOLING RATE, Degrees F. per Second at 1300° F.
ROCKWELL HARDNESS "C" SCALE
DISTANCE FROM WATER COOLED END OF STANDARD BAR—Sixteenths of Inch

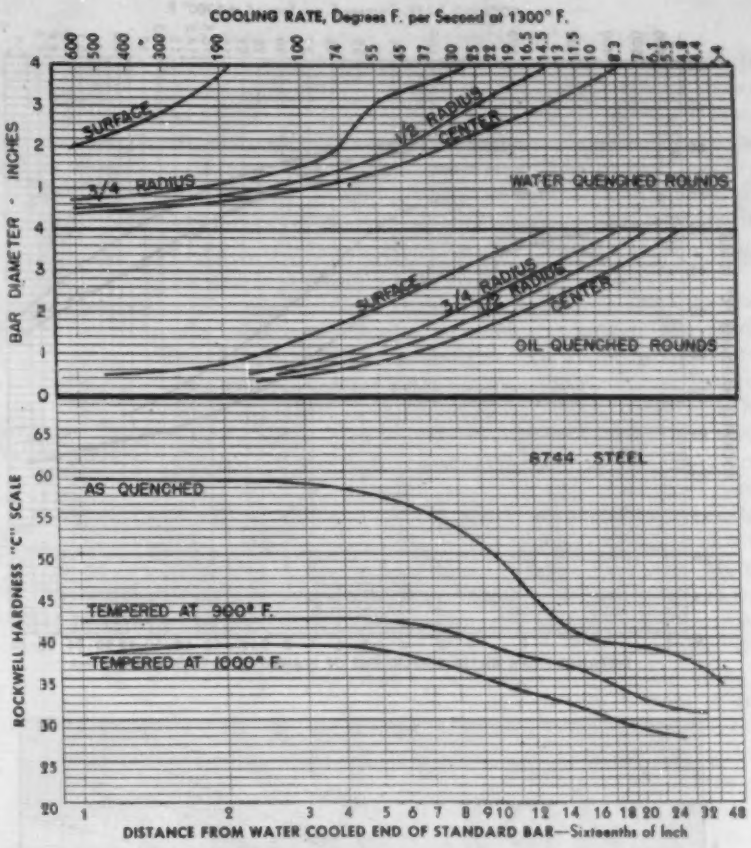
■ Fig. 12 – Hardenability of SAE 5130 heated to 1675 F and end quenched

Table 2 – Hardenability of 2-In. Diameter Bar of 5130 Steel (Quenched in Water)

Location in 2-In. Diameter Bar	Cooling Rate in Water-Quenched 2-In. Diameter Bar, deg per sec	Corresponding Locations on J Bar	Hardness—Rockwell C for 5130 from Curve in Fig. 12
Center	27	7½/16	39
¼ radius below surface	48	6/16	45
¼ radius below surface	70	4/16	52
Surface	000	1/16	53

Whether either one of these hardness patterns is considered suitable depends on what kind of loads are going to be imposed and on the views of the person making the decision.

Some metallurgists believe that for best physical properties after tempering, to say 35 to 40 Rockwell C, a hardness range suitable for an axle shaft, the steel must first be hardened above Rockwell C 49 at the surface and above 44 Rockwell C at the center. Others accept any as-quenched hardness at the center so long as the center comes within the hardness limits required after tempering. The center hardness could then be as low as 39 to 40 Rockwell C to stay within 35 to 40 range after tempering. It is not proposed to argue this point in this paper because the facts are not available to prove one side or the other correct. For example, we have no fatigue test results on partially hardened and tempered steel to compare with fatigue tests on fully hardened and tempered steel of the same hardness and composition.

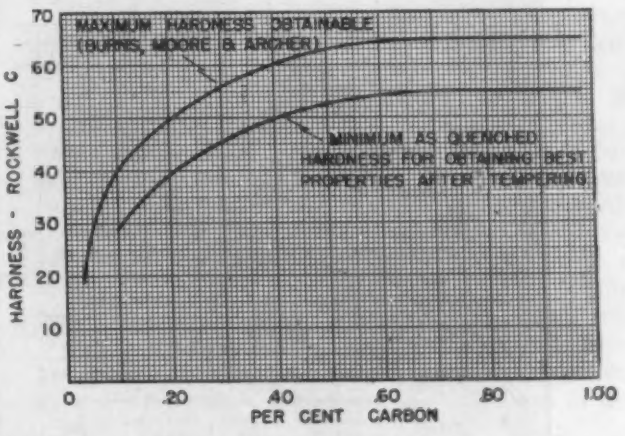

COOLING RATE, Degrees F. per Second at 1300° F.
600 500 400 300 190 100 74 55 45 37 30 25 22 19 16.5 14.5 13 11.5 10 8.3 7.6 7.1 6.5 5.9 5.4 4.9
BAR DIAMETER - INCHES
SURFACE
1/2 RADIUS
CENTER
3/4 RADIUS
WATER QUENCHED ROUNDS
SURFACE
3/4 RADIUS
1/2 RADIUS
CENTER
OIL QUENCHED ROUNDS
8744 STEEL
AS QUENCHED
TEMPERED AT 900° F.
TEMPERED AT 1000° F.
ROCKWELL HARDNESS "C" SCALE
DISTANCE FROM WATER COOLED END OF STANDARD BAR—Sixteenths of Inch

■ Fig. 13 – Hardness in round bars determined from hardenability curve

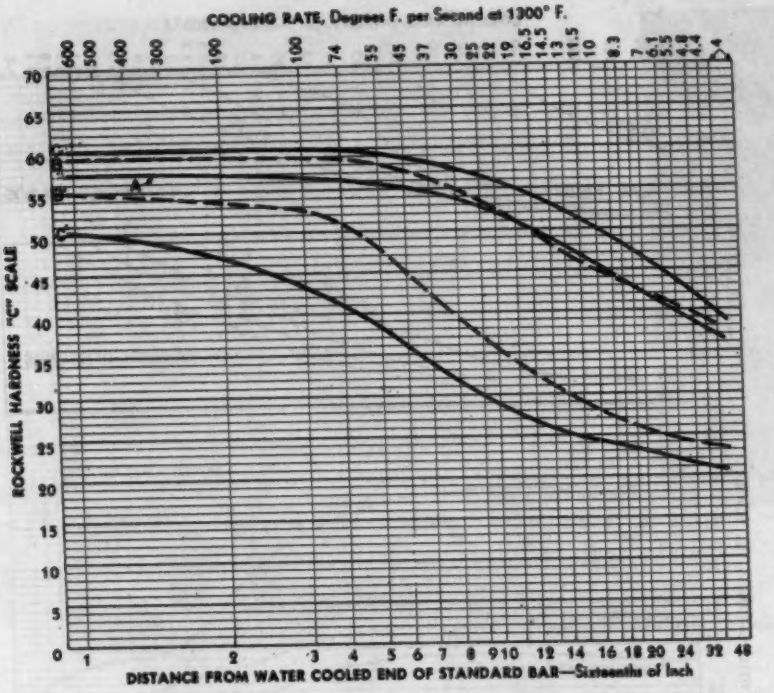
If we belong to the school that is satisfied with partial hardening throughout the section and we wanted a hardness after tempering of 35 to 40 Rockwell C, we would consider 5130 steel satisfactory for the part. By tempering at about 900 F the part of the bar that hardened to around 50 Rockwell C would be softened to 40 Rockwell C and the center that was hardened to 39 Rockwell C would temper to about 35 Rockwell C and the whole bar section would be within the desired hardness limits. The greater hardness decrease obtained during tempering fully hardened steel than occurs in partially hardened steel is shown in Fig. 13, giving the as-quenched hardenability curve for 8744 steel and the hardness curves obtained by tempering at 900 F and 1000 F. That part of the bar that hardened to 59 Rockwell C, when tempered at 900 F, softened to 42 Rockwell C, a drop of 17 points. When the quenched hardness was 50 Rockwell C, 900 F tempering only reduced the hardness 11 points to 39 Rockwell C. When the quenched hardness was 40 Rockwell C, 900 F only reduced the hardness 4 points to 36 Rockwell C.

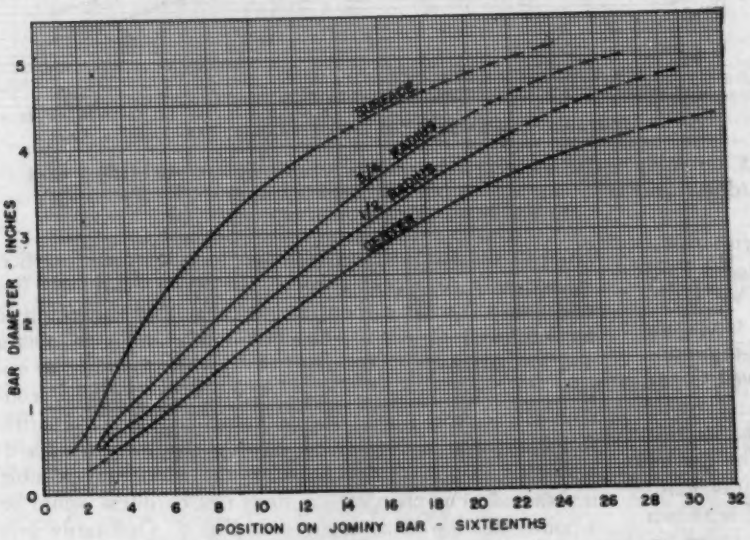
Metallurgists who require minimum 44 Rockwell C at the center and those who require full hardening throughout the section, or at least to ¾ radius below the surface would reject 5130 steel for this part. When we refer to "full hardening" we are again talking in terms on which all metallurgists are not on common ground.

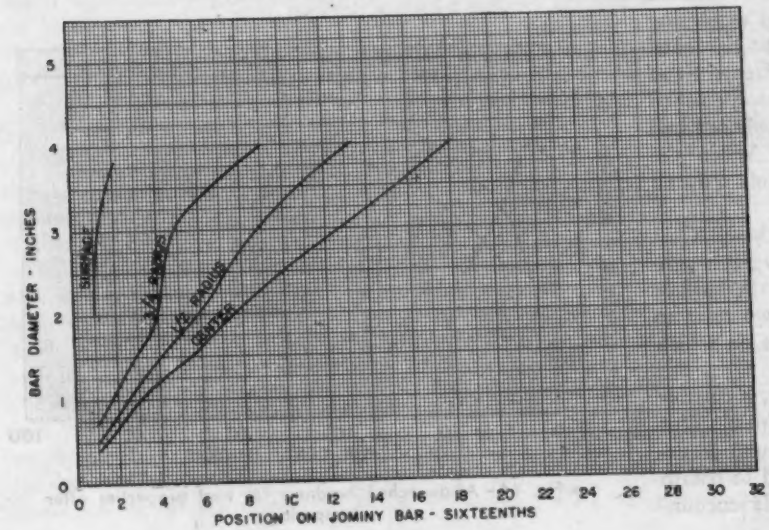
Each steel, depending upon its carbon content, is capable of being hardened to a certain maximum hardness. This maximum hardness determined by carbon content is depicted in Fig. 6. The quenching speed required to obtain this maximum hardness is faster than ordinarily encountered in the size sections common to automotive practice, so we have to accept a somewhat lower hardness. For example, a 0.40% carbon steel is capable of hardening to 60 Rockwell C when quenched fast enough to obtain 100% martensite. The speed of quench necessary to obtain this maximum hardness depends on the alloy content of the steel, therefore, its hardenability. Ordinarily, full hardening is considered accomplished when a hardness of 5 to 7 Rockwell C below the maximum possible is obtained. For plain carbon steels this hardness would be 5 points still lower, as depicted in Fig. 7. Ordinarily it is


MAXIMUM HARDNESS OBTAINABLE (BURNS, MOORE & ARCHER)
MINIMUM AS QUENCHED HARDNESS FOR OBTAINING BEST PROPERTIES AFTER TEMPERING
HARDNESS - ROCKWELL C
PER CENT CARBON

■ Fig. 14 – As-quenched hardness for best properties after tempering


COOLING RATE, Degrees F. per Second at 1300° F.
600 500 400 300 200 100 74 55 45 37 30 25 22 19 16.5 14.5 13 11.5 10 8.3 7 6.1 5.5 4.9 4.4 3.4
ROCKWELL HARDNESS "C" SCALE
DISTANCE FROM WATER COOLED END OF STANDARD BAR—Sixteenths of Inch





considered that lower hardness than full hardening, as just defined, will yield properties, when tempered, that are equal to those resulting from tempering after full hardening.

If we draw a curve in Fig. 14 at a distance of 10 points Rockwell C below the curve showing maximum hardness obtainable, we will show the hardness for each carbon content that may be considered as the acceptable hardness in the quenched condition for obtaining suitable properties after tempering. Fig. 14 shows that 50 Rockwell C is acceptable as full hardening for a steel containing 0.40% carbon. If, therefore, we wish our 2-in. diameter shaft to harden to 50 Rockwell C we should use a 0.40% carbon steel with enough hardenability so that 50 Rockwell C is obtained at a cooling rate of 18 deg per sec (the cooling rate at the center of a 2-in. round quenched in oil). Such a steel would have to harden above 50 Rockwell C to a distance of 12/16 in. from the end of a Jominy end-quench hardenability bar. The hardenability curve of such a steel would be as shown in Fig. 13. This curve is typical of a number of steels, such as 3140, 3240, 4140, 4340, and 8740, but all heats within the chemical specification limits of these steels would not have the required hardenability to fulfill the requirement of 50 Rockwell C at the center of a 2-in. diameter shaft oil quenched.

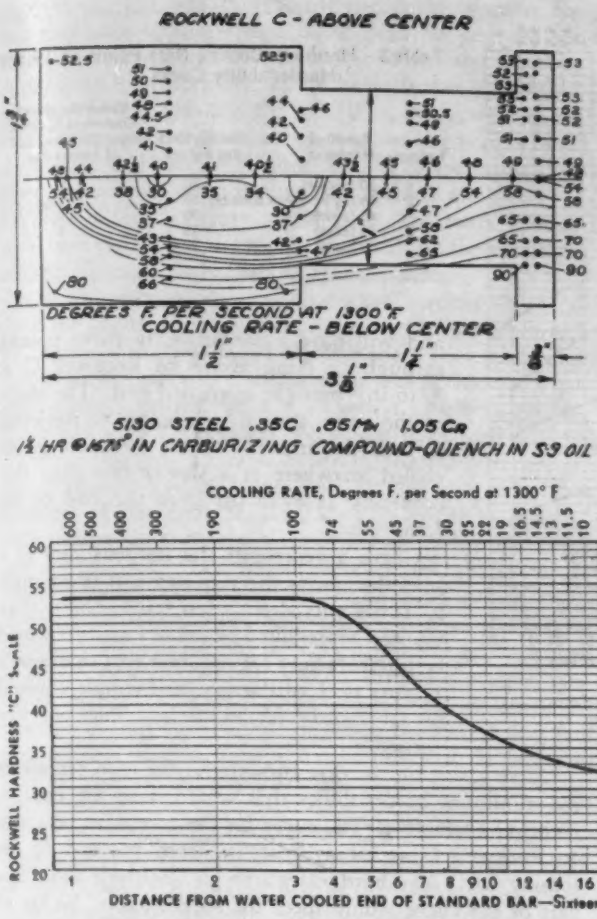
Fig. 15 shows how 8740 may vary in hardenability as the chemical composition varies within permissible limits. Hardenability is low when the alloy content is at the low specification limit and highest when the composition is on the high side of specification. Very few steels have composition limits that would always ensure a minimum of 50 Rockwell C at the center of a 2-in. diameter shaft quenched in oil. 9540 is such a steel which has a minimum hardenability above 12/16 in. at Rockwell C 50. Another steel that will meet this requirement is modified 9262 containing 0.25% molybdenum.

There is a great deal of 4340, 3240, 4140, 8740, and steels of similar hardenability used in highly stressed members of 2 in. or more in diameter. The obvious conclusion is that full hardening at the center is not necessary. Whatever hardness is desired in the as-quenched condition, Figs. 8, 9, and 10 can

■ Fig. 15 (top) – Hardenability curves: A – curve for steel that will harden to 50 Rockwell C at center of 2-in. round, oil-quenched; B – expected limits of 94% of heats of 8740 and 8640 steel; C – extreme limits for 8740 and 8640 in 168 heats

■ Fig. 16 (middle) – Correlation of identical cooling rates in Jominy bar and oil-quenched rounds

■ Fig. 17 (left) – Correlation of identical cooling rates in Jominy bar and water-quenched rounds


ROCKWELL C - ABOVE CENTER
DEGREES F. PER SECOND AT 1300°F
COOLING RATE - BELOW CENTER
5130 STEEL .35C .85Mn 1.05Cr
1½ HR @1675° IN CARBURIZING COMPOUND-QUENCH IN 5.9 OIL
COOLING RATE, Degrees F. per Second at 1300° F
ROCKWELL HARDNESS "C" Scale
DISTANCE FROM WATER COOLED END OF STANDARD BAR—Sixteenths of Inch

■ Fig. 18 – Hardenability of SAE 5130 – heated to 1675 F, and quenched

be used to determine the hardenability limits for the steel that will meet the requirements. For those that dislike logarithmic charts, the same information is presented with linear coordinates in Figs. 16 and 17.

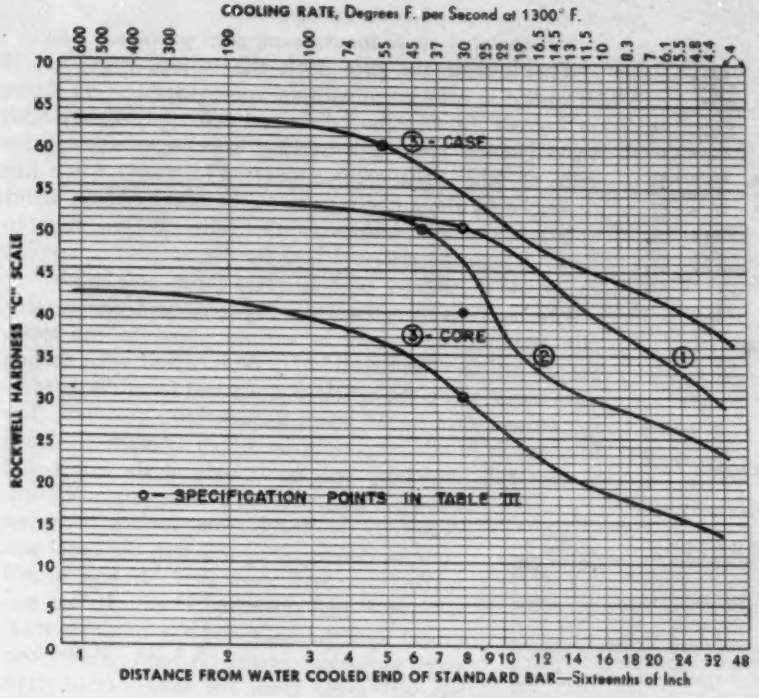
The question of what is the proper hardness to specify and what steel should be specified to obtain that hardness will be discussed again later in the paper. At the moment, there is more to be said about selecting a steel to meet a desired hardness requirement in an article of irregular shape. Cooling rates in various size rounds and plates as shown in Figs. 8, 9, and 10 have been measured by means of thermocouples, so that where the article to be made is represented fairly closely by one of these simple shapes, then the published cooling rates can be used to determine what kind of steel to select. Irregularly shaped objects, however, are frequently of such shape that cooling rates cannot be estimated or calculated closely enough for guidance in selecting the proper steel. There are two ways of determining these cooling rates. One way is to attach thermocouples to the article at points of importance stress-wise and actually measure the temperature change during quenching from the hardening temperature. This method is cumbersome and requires equipment and technique that few plants have. The other method can be readily carried out at any plant because it involves only the determination of hardnesses throughout the article after quenching from the hardening temperature and then substitution of cooling rates corresponding to those hardnesses picked from a hardenability curve for the same steel previously used.

To illustrate the procedure, the various steps will be described in connection with the spool-shaped object shown in Fig. 18. The method described will apply to any other shaped object such as a steering knuckle, a transmission gear, or other vehicle component. This method was first described by the author[6] prior to the adoption and standardization of the Jominy hardenability test in 1941 by SAE. It is being repeated here for the purpose of revising the charts to conform with standard forms now in use.

The object is to determine what steels will produce certain required hardnesses at various points in the spool-shaped article. The first thing to do is to find what cooling rates occur when the article is quenched from the hardening temperature in the selected quenching medium. The first step is to select a steel from which to make the spool so that full hardening will occur only at the edges and corners, thus ensuring a wide range of hardnesses in the spool from surface to center after hardening. For this test, the spool was made from SAE 5130 steel. It was heated to 1675 F and quenched in oil. It was sectioned longitudinally and the hardness measured at various points. A hardenability test bar was made from the same bar of 5130 steel and also heated to 1675 F prior to end quenching. The hardnesses on cross-section of the spool after hardening and the hardenability curve for 5130 steel are shown in Fig. 18. The cooling rates corresponding to the hardnesses shown on the cross-section of the spool were picked from the hardenability curve and located on the lower half of the spool cross-section in Fig. 18 in the same positions as the corresponding hardnesses. Points of equal cooling rates are connected by lines, thus forming contours of equal cooling rates. We now know the cooling rates that occur when the spool is quenched from 1675 F in oil. If we wish to know the cooling rates when quenched from some other temperature or in some other quenching medium, the process may be repeated, being sure each time that the spool and the hardenability test bar are both heated to the same temperature and that both have the same treatment, if any, prior to the hardening treatment. Any difference in the composition or treatment between the hardenability bar and the object being investigated will produce erroneous results.

Having determined the cooling rates in the spool when oil quenched from 1675 F, we can tell from the hardenability curve of any other steel what hardness that steel would produce in the spool if quenched from 1675 F. If we know what hardness is required in the spool, it becomes a simple matter to select the steel which will produce that hardness in the spool. The cooling rates occurring in the spool when it is quenched, coupled with certain hardnesses required at known cooling-rate locations, provide a series of hardness-cooling rate pairs which will define a hardness-cooling rate curve. The steel that satisfactorily meets the hardness requirements for the spool then will have a Jominy hardenability curve having the same hardness-cooling rate pairs. To illustrate this principle, let us assume certain requirements as to hardness in the

[6] See SAE Transactions, Vol. 49, July, 1941, pp. 266-276: "Use of Hardenability Tests for Selection and Specification of Automotive Steels," by A. L. Boegehold.


COOLING RATE, Degrees F. per Second at 1300° F.
600 500 400 300 190 100 74 55 45 37 30 25 22 19 16.5 14.5 13 11.5 10 8.3 7 6.1 5.3 4.8 4.4 4
ROCKWELL HARDNESS "C" SCALE
70 65 60 55 50 45 40 35 30 25 20 15 10 5 0
① - CASE
③ - CORE
○ - SPECIFICATION POINTS IN TABLE III
1 2 3 4 5 6 7 8 9 10 12 14 16 18 20 24 32 48
DISTANCE FROM WATER COOLED END OF STANDARD BAR—Sixteenths of Inch

■ Fig. 19 – Hardenability curves that will meet example specifications 1, 2, and 3

Table 3 – Hardness-Cooling Rate Points on Jominy Hardenability Curve

Example	Specified Hardness	Cooling Rate, deg per sec	Sixteenths of Inch Equivalent Distance from Quenched End of Jominy Bar
1	50 (center)	30	8
2	50 (¼ in. below surface)	40	8½
	40 (center)	30	8
3	60 (case)	58	6
	30 (core)	30	8

spool. We will consider three examples. In the first two examples the part is to be subjected to stresses which will require in the quenched and tempered part a hardness of 32 to 36 Rockwell C.

Those who insist on obtaining substantially full hardening throughout the part, prior to tempering, will require a hardness of 50 Rockwell C minimum in the as-quenched condition. Those who will accept a lower internal hardness, so long as it is higher than the hardness desired after tempering, might specify 40 Rockwell C minimum throughout and 50 Rockwell C minimum to a depth of ¼ in. below the surface.

In the third example we can assume the requirements of a part to be case carburized in which 60 Rockwell C is required to a depth of 0.025 in. and 30 Rockwell C is required as the minimum core hardness. The hardness requirements in these three examples then are as follows:

Example 1. 50 Rockwell C minimum throughout.

Example 2. 50 Rockwell C minimum ¼ in. below surface.
40 Rockwell C minimum throughout.

Example 3. 60 Rockwell C minimum 0.025 in. below the surface.
30 Rockwell C minimum throughout.

These hardness requirements, coupled with the cooling rates known for the locations in the spool at which hardness is specified, give us hardness-cooling rate points on the Jominy hardenability curve as shown in Table 3.

With this information we can draw curves in Fig. 19 that will represent steels that can meet the requirements of examples 1, 2, and 3.

Curve 1 meets the requirement of example 1 and represents the minimum hardenability that will be satisfactory. Any steel having greater hardenability will be satisfactory and will have a hardenability curve passing through or lying above 50 Rockwell C at 8/16 in. from the quenched end. The shape of the curve after it falls below 50 Rockwell C does not matter because the spool is not cooled anywhere at a slower rate than that occurring at 8/16 in. from the end of the Jominy bar.

Curve 2 represents the hardenability of a steel that meets the requirements of example 2. It has only slightly less hardenability than the steel represented by curve 1 and yet would be unsatisfactory for requirements in example 1. Any steel satisfactory for example 1 would be satisfactory for example 2, but not vice versa.

Curve 3, one for the case and one for the core, represents the minimum hardenability that would meet the requirements of example 3. The curve for the core must lie above 30 Rockwell C as far as 8/16 in. from the end of the Jominy bar. Any hardness above 30 Rockwell C in this part of the curve would meet the specification. So far as the specification given in the example goes, the steel represented by curve 1 would be satisfactory. In actual practice, however, additional requirements would have to be imposed to limit the hardness that would occur at cooling rates faster than 30 deg per sec. This is done by specifying the carbon content limits, and will be discussed more in detail later in the paper.

Curve 3 for the carburized case meets the requirement that at least 60 Rockwell C is obtained as low as 58 deg per sec cooling speed, which occurs at 0.025 in. below the surface of the spool where the 1-in. diameter section joins the 1¾ in. diameter part of the spool. At all other places on the spool the cooling rate at 0.025 in. below the surface is faster than 58 deg per sec.

■ Precautions for Determining Cooling Rate

In the selection of steel to be used for determining cooling rates in articles to be heat-treated, the object to be explored for cooling rates should be made from a steel of lower hardenability than would normally be used for the part so that full hardness is only obtained at edges and corners where the cooling rate is fastest. The rest of the object will vary considerably in hardness, depending on the cooling rate. If the object is made of a steel of high hardenability, the cooling rates will not be revealed because maximum hardness will be obtained throughout, since all cooling rates result in the same hardness for a high-hardenability steel. To illustrate, refer to the spool-shaped object shown in Fig. 18. When made of 5130 and oil quenched from 1675 F, the hardness varied from 40 to

53 Rockwell C. According to the hardenability curve for this steel any cooling speed faster than 90 deg per sec results in 53 Rockwell C – no harder. Where the hardness in the spool is 53 Rockwell, all we know is that the cooling rate was at least 90 deg per sec. It is obvious from the shape of the piece that those regions near the periphery of the flanged end that are 53 Rockwell C cooled faster than places nearer the axis of the piece, which also were 53 Rockwell C hard. How much faster the outer portion cooled we cannot tell with 5130 steel because the hardness-cooling rate curve is flat above 90 deg per sec.

A high-hardenability steel like 4340 hardens to about the same hardness at all cooling speeds above 20 deg per sec, so that the spool-shaped object made from 4340 would harden to 53 Rockwell C throughout and no idea of what cooling rates occurred in the spool would have been gained. If we had used a 3115 steel we would have been able to determine the cooling rates better in the flanged part of the spool that cooled faster than 90 deg per sec. The hardness of one heat of 3115 steel on which a hardenability curve is available, decreases steadily beginning at 400 deg per sec. Care should be exercised, therefore, in selecting the steel to be used for determining cooling rates in objects so that the steel will harden only at the fastest cooling portions of the object being investigated. By means of charts showing cooling rates in various diameter bars shown in Figs. 8, 9, and 10 an estimate can be made of expected cooling rates in the object to be studied and the steel selected should have a hardenability such that the slanting part of the hardenability curve occurs at those cooling rates.

For objects equivalent to a 1-in. round quenched in water and to a ½-in. round quenched in oil, a steel should be used having the slanting part of the hardenability curve in the first three-sixteenths from the quenched end. 1040 or 1035 would be suitable materials for this purpose. For water-quenched objects equivalent to 1 to 3 in. in diameter, the slanting part of the curve should be between 1 and 12/16 in. from the quenched end. 1045 would be satisfactory for this purpose.

■ Testing Accuracy of Hardness Predictions

When the method described in this paper has been used to determine a steel that will quench to certain required hardness values in some object, the natural procedure is to make the object of the selected steel to check the accuracy of the prediction. All conditions that influence cooling rates must be maintained in the check test the same as they were when cooling rates in the object were first determined. Also, the steel selected for the check test should be from the same bar as that from which the hardenability test bar was made and the heat-treatment of the object when made from the selected steel must be the same as was used for the hardenability bar on which the prediction was based. Any violation of this principle will result in failure to obtain the hardnesses predicted in the material finally selected for making the article being studied.

Table 4 gives a list of factors that affect the rate of cooling and that affect hardenability of the steel. It is these factors that have to be kept the same in the process of proving out the accuracy of hardness predictions.

■ Hardenability Limits

It is seldom practical to specify a steel on the basis of a minimum-hardenability curve without supplementing the specification with chemical composition limits. The use of a minimum-hardenability curve along with a chemical composition specification provides definite limits on maximum hardenability, as will be seen later in discussion of selection of steels. In some cases it is desirable to place more restricting limits on hardenability by specifying a minimum- and maximum-hardenability curve. This is more often a necessity when the article is to be carburized and is of such shape that considerable change in section, therefore in cooling rates, occurs in the article. For example 4 then let us take a gear such as a transmission gear. The cooling rates may be determined by the process illustrated by means of the spool. Let it be assumed that the important locations in which hardness must be controlled are within the limits shown in Table 5.

Table 4 – Factors Affecting Cooling Rate and Hardenability

Factors Affecting Cooling Rate of Object Quenched	Factors Affecting Hardenability of Steel
1. Kind of quenching medium	1. Heating temperature (grain-size effect)
2. Temperature of quenching medium	2. Prior treatment (effect on carbides)
3. Circulating speed of quenching medium	3. Heating rate (effect on carbides)
4. Size and shape of object	4. Composition
5. Quantity and kind of scale on object	5. Grain size
6. Temperature of object quenched	6. Unknown composition factors

Table 5 – Locations Within Which Hardness Must Be Controlled

Hardness Control Location	Cooling Rate at Control Location, deg per sec	Equivalent Distance on J Bar, sixteenths	Rockwell C Hardness Range Desired
0.080 in. below surface of tooth	200	2	35 – 45
Center at base of tooth	80	4	30 – 40
Center of hub	50	5½	25 – 35

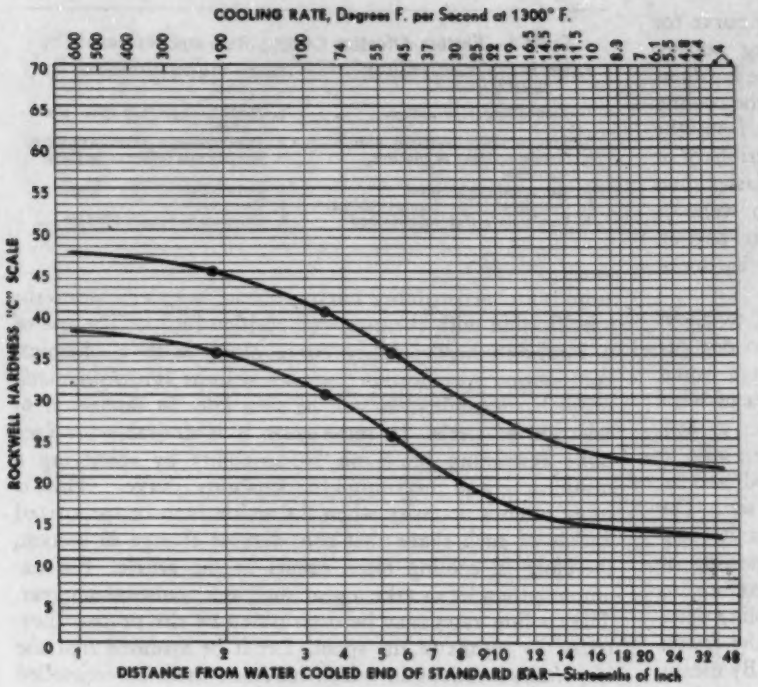
These requirements give us three hardness-cooling rate pairs or hardness-sixteenths pairs which define two limiting curves, as shown in Fig. 20. These two curves could be described in a specification in two different ways. Either by specifying a hardness range at three different points on the Jominy bar or by specifying a hardness range at the 1/16-in. point and a maximum and minimum distance from the quenched end at a reference hardness number. The reference hardness number is determined by the carbon content range of the steel as is the hardness range at 1/16 in. from the quenched end. The first method would express the curves:

35 to 45 Rockwell C at 2/16 in.
30 to 40 Rockwell C at 4/16 in.
25 to 35 Rockwell C at 5½/16 in.

The second method would express the curves:

35 to 45 Rockwell C at 1/16 in.
4/16 in. to 8/16 in. at J-30 Rockwell C

At the moment, industry is in favor of the latter method for expressing these curves. The carbon range required to meet the specification of 35 to 45 Rockwell at J-1/16 in. as determined from curves in Fig. 7, would be 0.125 to 0.15% carbon but this is not a commercially practical limit. It is only the occasional heat that hardens at the 1/16 in. point up to the maximum hardness represented by the upper curve. Practically, the carbon limits can be held at the conventional 7- or 8-point range and still come within the 10-point hardness spread. Where more precise control is required it becomes necessary to select heats to come within the prescribed limits.


COOLING RATE, Degrees F. per Second at 1300° F.
ROCKWELL HARDNESS "C" SCALE
DISTANCE FROM WATER COOLED END OF STANDARD BAR—Sixteenths of Inch

■ Fig. 20 – Hardenability limits for example No. 4 – 37 to 47 Rockwell C at 1/16 in.; 4/16 to 8/16 at J-30 Rockwell C
o specification points

The reference hardness numbers to be used in specifying points on hardenability curves have been determined from a study of the hardenability curves for 1400 heats collected by the SAE-AISI Hardenability Committee. These reference hardness numbers as decided upon by the SAE War Engineering Board Iron & Steel Committee are shown in Table 6.

Table 6 – Hardness Reference Numbers for Specifying Hardenability Limits

Carbon Range	Hardness Reference Number – Rockwell C	
	Alloy Steel	Plain Carbon Steel
0.08 – 0.12	25	..
0.13 – 0.17	25	..
0.18 – 0.22	30	25
0.23 – 0.27	35	30
0.28 – 0.32	40	35
0.33 – 0.37	45	40
0.38 – 0.42	45	40
0.43 – 0.47	50	45
0.48 – 0.52	50	45
0.53 – 0.57	55	50
0.58 – 0.62	55	50

One more example of the determination of cooling rates and the selection of a suitable hardenability to meet the hardness requirements will be given and then the question of selecting steels to meet hardenability requirements will be discussed.

■ Cooling Rates in Transmission Gear

Fig. 21 shows a mechanical drawing of a rear unit center gear for the Hydramatic transmission. The gear and a Jominy test bar were both made from the same bar of SAE 5040 steel. The hardenability curve obtained from the Jominy bar provided the relation between cooling rate and hardness necessary for determining cooling rates in the gear. The gear made from 5040 steel was heated in activated salt at 1525 F and quenched in oil of 80- to 100-sec viscosity at 100 F. The hardness traverse from outside to center was made and is shown in Fig. 22. Alongside the hardness figures are shown the cooling rates picked from the hardenability curve for 5040 steel shown in Fig. 23. Also shown in Fig. 23 are hardenability curves for 5145 and 5030, one with higher alloy and carbon content than 5040 and the other with lower carbon than 5040 steel.

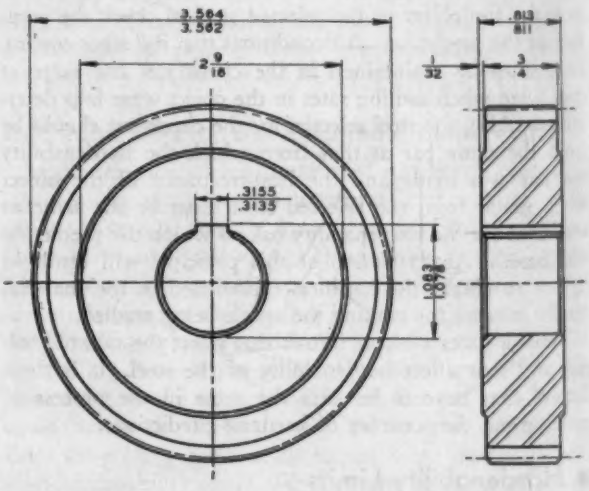
Table 7 shows the cooling rates determined for the gear, with hardnesses at those cooling rates for the steels.

From the hardness information in Table 7 we conclude that 5040 steel having the hardenability of the particular heat investigated gives just about the hardness distribution desired for this particular gear. Very little tempering is necessary to avoid brittleness; therefore, the hard surface layer from the activated salt bath is maintained full hard. X-5145, however, results in so much higher hardness that a higher tempering would be necessary to reduce the hardness and brittleness, thus sacrificing some of the surface hardness obtained from carburizing in the activated bath. 5030, on the other hand, would probably be unsatisfactory because of too low hardness and strength in the gear teeth.

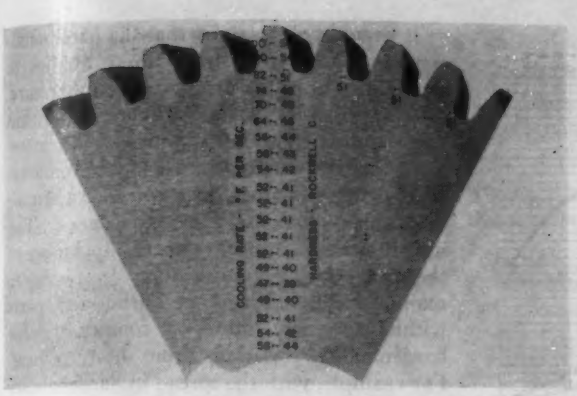
Table 7 – Cooling Rates and Hardnesses

Cooling Rate deg per sec at 1300 F	X-5145*	5040	5030*
100	57.5	54.0	46.0
82	57.0	52.0	42.0
74	56.0	49.0	39.0
70	56.0	47.5	38.0
64	55.0	45.0	36.0
58	54.0	43.0	34.0
56	53.5	42.5	33.5
54	53.0	42.0	33.0
52	52.0	41.0	32.0
49	51.5	40.0	31.5
47	50.5	39.0	30.5

* Predicted from hardenability curve and cooling rates estimated by substitution in 5040 gear.



■ Fig. 21 – Rear unit center gear for Hydra-matic transmission



■ Fig. 22 – Hardness traverse on gear section

■ Selection and Specification of Steel

We have discussed the process of establishing the degree of hardenability required in the steel used to achieve certain desired hardness values by heat-treating some object of random size and shape. Having decided what hardenability curve the steel should possess, the next step is to select the kind of steel that has the desired hardenability. When we examine the hardenability curves for different heats of the same chemical specification steel, we find that as the chemistry varies within the specified limits the hardenability varies also, increasing as the amount of carbon and alloying elements increase. We find then that each steel specification has a range of hardenability that can be represented by a band defined by a maximum hardenability curve and a minimum hardenability curve.

In example 1 we specified a minimum hardenability of 8/16 in. at J-50. The steel specified to meet that requirement then must have the low side of the hardenability band pass through 8/16 in. at 50 Rockwell C. Only a few steels would meet such a specification, among them are 3240, 8650, 4150, 9450, and 9540. The hardenability of these five steels will vary between the low and high limits shown in Table 8 on account of variation in chemical composition within the limits standard for each steel.

The hardenability bands defined by the limits in Tables 8, 9, and 10 were determined by means of the SAE-AISI survey of 2000 heats of steel. Fig. 24 shows the range or band of hardenability that can be expected in connection with 8650 and 8750 steel. Observe that the low side of the band passing through 8½ sixteenths at 50 Rockwell C slightly exceeds the specification for example 1.

Table 8 – Hardenability Bands

Steel	Sixteenths at 50 Rockwell C Hardenability Range Min.	Max.
3240	9	32
8750 or 8650	8½	29½
9450	8	48
9540	24	48
4150	11	48

In example 2 the minimum hardenability specified was 6½ sixteenths at 50 Rockwell C. Only two steels on which hardenability bands are available in addition to those in Table 8 will meet this specification. See Table 9.

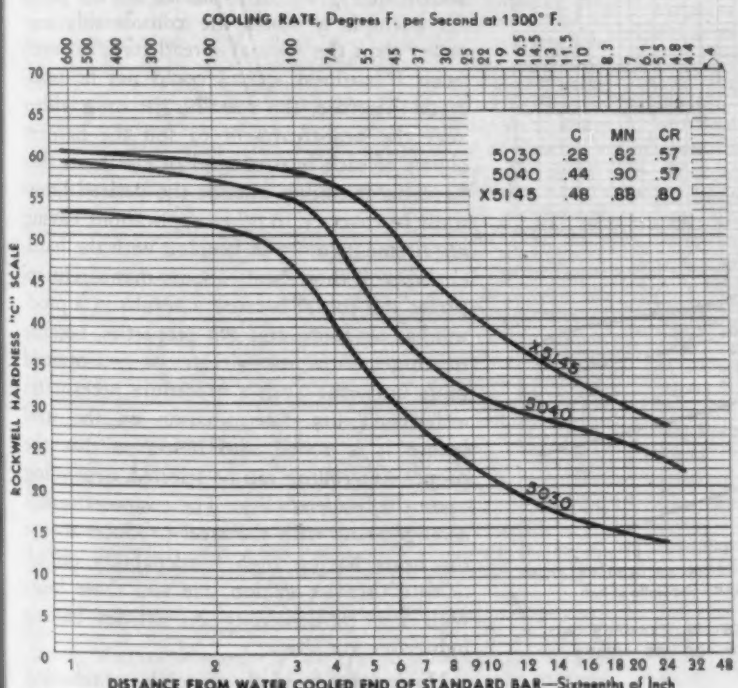
Table 9 – Hardenability Bands

Steel	Sixteenths at 50 Rockwell C Hardenability Range Min.	Max.
5150	7	14
8150	7	18

In example 3 the carburizing steel was required to have a minimum of 8/16 in. at J-30 Rockwell C.

The steels on which we have records of hardenability bands do not show any band minimum high enough to meet this specification. The nearest approach to meeting this requirement is 4820, which has a spread of from 7/16 in. to 32/16 in. at J-30 Rockwell C.

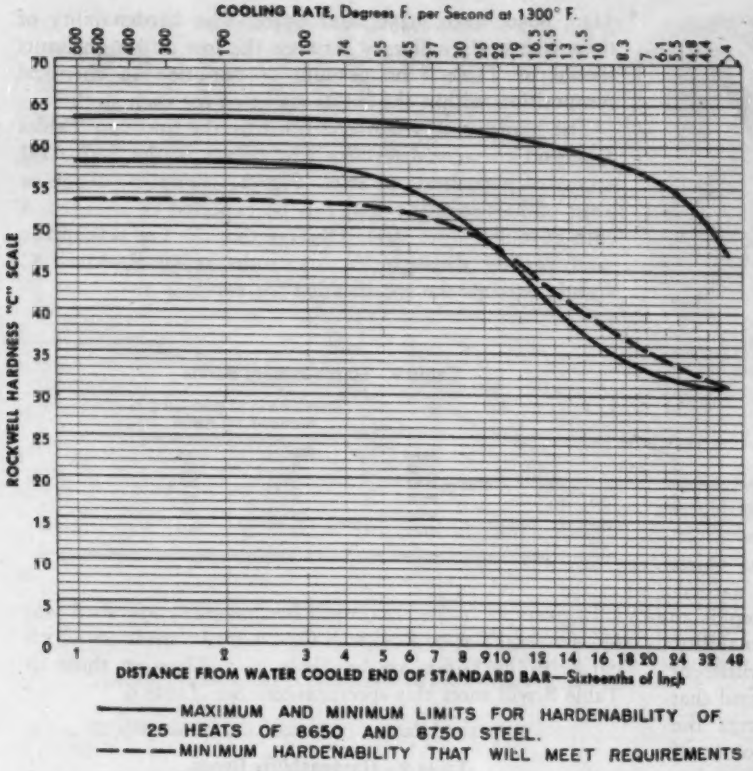
Example 4, which is typical of a frequently stipulated requirement, limits the hardenability to a maximum as well as a minimum. The hardenability limits are set for the purpose of maintaining the Rockwell hardness within a 10-point range at several points in the core of a carburized gear. None of the carburizing steels of standard chemical ranges will have all heats falling within the hardenability band specified, which was 4/16 to 8/16 in. at J-30 Rockwell C. Some typical hardenability bands resulting from the SAE-AISI hardenability committee work are shown in Table 10.



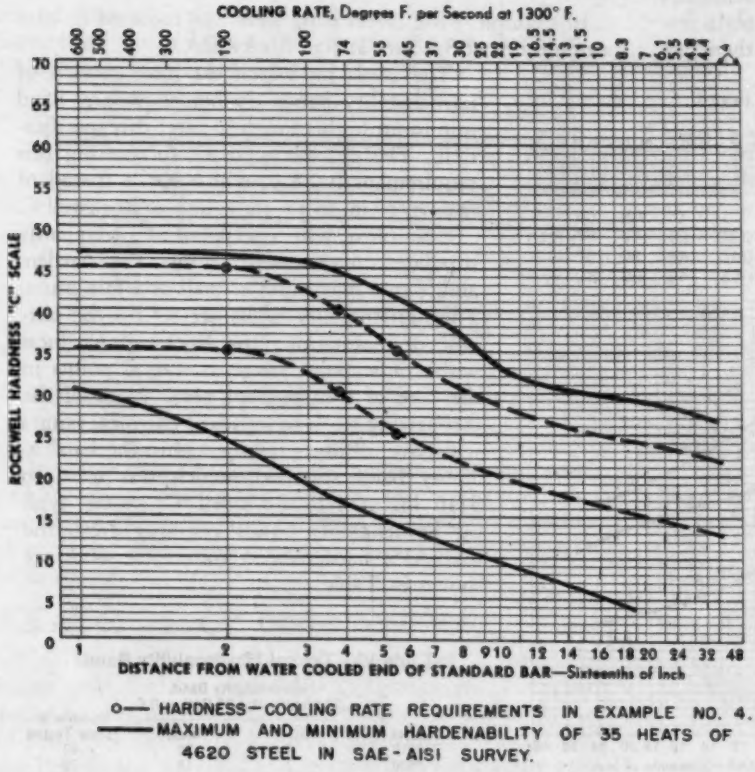
■ Fig. 23 – Hardenability of three gear steels

Table 10 – Typical Hardenability Bands

Steel	Hardenability Band, Sixteenths at J-30 Rockwell C Min.	Max.	Number of Heats Tested
3310	3	26	63
4620	1	14	35
4820	7	32	26
8720 and 8620	2	10	179


COOLING RATE, Degrees F. per Second at 1300° F.
600 500 400 300 190 100 74 55 45 37 30 25 22 19 16.5 14.5 13 11.5 10 8.3 7 6.1 5.5 4.8 4.4 4
ROCKWELL HARDNESS "C" SCALE
70 65 60 55 50 45 40 35 30 25 20 15 10 5 0
1 2 3 4 5 6 7 8 9 10 12 14 16 18 20 24 32 48
DISTANCE FROM WATER COOLED END OF STANDARD BAR—Sixteenths of Inch
MAXIMUM AND MINIMUM LIMITS FOR HARDENABILITY OF 25 HEATS OF 8650 AND 8750 STEEL.
MINIMUM HARDENABILITY THAT WILL MEET REQUIREMENTS

■ Fig. 24 – Hardenability curves that can be expected with 8650 and 8750 steels


COOLING RATE, Degrees F. per Second at 1300° F.
600 500 400 300 190 100 74 55 45 37 30 25 22 19 16.5 14.5 13 11.5 10 8.3 7 6.1 5.5 4.8 4.4 4
ROCKWELL HARDNESS "C" SCALE
70 65 60 55 50 45 40 35 30 25 20 15 10 5 0
1 2 3 4 5 6 7 8 9 10 12 14 16 18 20 24 32 48
DISTANCE FROM WATER COOLED END OF STANDARD BAR—Sixteenths of Inch
HARDNESS—COOLING RATE REQUIREMENTS IN EXAMPLE NO. 4.
MAXIMUM AND MINIMUM HARDENABILITY OF 35 HEATS OF 4620 STEEL IN SAE-AISI SURVEY.

■ Fig. 25 – Hardenability curves that can be expected with 4620 steel

It should be explained that the hardenability variation shown in Table 10 applies to quenching from the carburizing temperature. The hardenability spread encountered within the number of heats examined could be reduced by quenching from a lower temperature, as is sometimes done to refine the case and obtain maximum case hardness. The metallurgist has recourse to manipulation of the heat-treating temperatures in order to maintain the hardness of carburized parts within specified limits to compensate for hardenability variations from heat to heat. For example, with the spread in hardenability shown by 4620 steel in Table 10, quenching from the carburizing temperature would result in a hardness variation from 18 to 45 Rockwell C at any point in the quenched article having a cooling rate of 100 deg per sec.

It is interesting to note the complete change in appreciation of this variation in hardenability that has occurred in seven years. When the first paper on the end-quenched hardenability test bar was presented in 1937 there was considerable doubt and surprise expressed about the table included in the paper showing a large variation in hardenability of carburized 4615 steel. Today, the reasons for most of this variation are understood and are taken as a matter of course. Measures must be taken in utilization of steel to compensate for this variable. As a result, the ability to obtain a uniform product is much greater than it was before the question was understood. The upper and lower hardenability limits for 35 heats of 4620 are shown in Fig. 25. Also shown are the limits for example 4 which are considerably narrower than the normal spread for 4620 steel. Such a hardness spread could not be tolerated; therefore, the metallurgist must either vary the heat-treatment to suit the hardenability of each heat of steel or else heats that have hardenability outside the desired limits must be diverted to other applications having requirements more in keeping with the hardenability of those heats. Some manufacturers prefer this course because it results in a product having more uniform properties, because treating temperatures can be maintained more uniform. Such a procedure necessitates a hardenability determination on the steel before it is rolled into billets so that the proper size rolling can be selected, depending upon the hardenability. The customer makes arrangements with the steel producer to use the heats having high hardenability on articles of heavier section size and those heats with low hardenability on articles having lighter section size.

Having discussed the principles of selecting steel on the basis of hardenability and indicated some customary procedures as to hard-

ness requirements in the as-hardened condition to ensure a desirable structure after tempering, the subject should be rounded out with a table showing the kind of steel to use and the treatment to obtain any desired hardness or tensile strength condition in any size section up to 4 or 5 in. round or 3 in. thickness of plate. This question is receiving the attention of the SAE War Engineering Board Iron & Steel Committee.

We know the cooling rates from surface to center in each of the sections we want to include in the table, so if we can decide on as-quenched hardnesses at the surface, center, and half radius that we will accept, we can use these hardness-cooling rate pairs to construct the hardness-cooling rate curve which is, in fact, the hardenability curve of the steel that will produce the desired hardness in that size section. This work, however, is not yet completed but when it is, a great deal of time will be saved by the assistance it will give the engineer and metallurgist in selecting the proper steel.

Transoceanic Air Cargo

continued from page 459

and its methods of use, to an extent that will put it on a true commercial footing. One factor of expense that Comdr. MacCoull has neglected to mention is that chargeable to helium gas. As a matter of fact, the direct cost of helium is not ordinarily as large as is commonly supposed, but indirectly is a liability far greater than most of its proponents seem willing to admit. Not only is the unit lift less, but the gas volume factor from a practical standpoint is considerably lower with helium inflation than with hydrogen, due to the much greater importance of ensuring against any expansion loss of the valuable helium. For the same reason, water-recovery apparatus must be more complete with helium inflation; and yet, with a generally inflammable type of construction, the use of helium is admittedly necessary in order to avoid the known fire risk of hydrogen.

In discussing an airship design exclusively for cargo carrying, it should be possible to re-open the consideration of hydrogen gas, especially when there is available for further development a type of construction, the metalclad, which by credible theory and available tests is found to be entirely proof against fire from ordinary static discharge, and is also comparatively easy to protect against other possible fire risks. The advantages then obtainable are so large as almost at one stroke to put the commercial cargo airship on a paying basis. Here, the ultimate range, although of no direct significance, is still a good indication of general transport efficiency. The various improvements that could be made on the modernized Macon, from a range point of view, in conservative round numbers are then as follows: from increased unit lift and effective volume (9% and 10%, respectively, of the gross), 40% increase in ultimate range at constant speed and altitude; from the use of hydrogen as fuel, a further (factorial) increase of 20% is readily obtainable. If, in addition the ship is kept at static ceiling, giving it the benefit of increased altitude not normally available to the helium-filled ship, a further gain of about 40% is obtained, thereby making the total increase due to the gas change 135%. From the elimination of unnecessary compartments and the use of a lower fineness ratio than is feasible with a compartmented ship, the total gain can readily be still further increased. And still we are not through.

A hydrogen-filled airship, which progressively consumes its lifting gas, and which for special purposes can even discharge substantial quantities without great financial loss, has more flexible operating conditions. It is no longer exclusively a low-altitude craft, but can seek a level where unfavorable winds, icing conditions, and so on, may be avoided and conditions picked that are safer and less demanding on fuel.

Aerodynamically, also, it is at least possible that great improvements can be made by suitable construction, as already indicated by Comdr. MacCoull. In order to use laminar flow to much practical advantage, it will obviously be necessary to avoid the conventional polygonal or straight-sided construction because even small changes in the angle of attack, such as are continually occurring, will cause flow transition over the relatively sharp edges.

A realization of laminar flow over an airship hull to the extent already proved attainable for wings still awaits definite substantiation; but if, with proper hull design and stern propulsion, such flow can be maintained over an equivalent proportion of area, the gain will be great enough to justify almost any amount of surface refinement; for not only does the airship hull represent a considerably larger proportion of the total drag than the wing of the usual airplane, but it has a Reynolds number so vast as to increase greatly the numerical ratio between the turbulent and laminar types of drag. Scientifically considered, it is easy to show a reduction of overall drag for the entire airship to considerably less than half of the minimum otherwise possible. The effect on performance and general economy of operation can well be imagined.

Further performance improvement of no small magnitude can be obtained, as suggested, by taking on in the air a large airplane which has taken off independently, and which can incidentally be used for pickup and delivery of substantial cargo consignments at points other than the airship terminals.

Altogether, for a given size, it appears that present airship range, and to a similar order of magnitude the delivery factor, can be at least doubled, and probably tripled or more, if we are simply willing to do what it takes.

Comdr. MacCoull: As Mr. Upson says, the L/D value of a plane can be raised by providing a greater wing area for a given fuselage area. This is equivalent to reducing the parasitic drag ratio, as shown in Fig. 2 of the paper, and designers have striven constantly to this end. But larger wings entail larger tail and control surfaces with their additional drag so the gain in L/D may not be as great as expected. Also, the weight of larger wings and control surfaces will be greater. Thus while it is possible to build a plane with an L/D greater than 18, or a weight ratio as high as 2, the combination of both is quite another matter, and it is the combination which results in the range of the plane. However it is analyzed, no planes have yet publicly shown themselves to be capable of an ultimate range exceeding the 6800 miles used in this paper for comparison with the U.S.S. Macon built 10 years ago.

Mr. Upson's suggestions as to future improvements in airships are stimulating. I did not refer to hydrogen in this paper because the public at large has not, and would not, distinguish the difference between a casualty caused by burning hydrogen and an essential airship failure. I have been very conservative in attempting to show that a helium-filled ship built like the Hindenburg or larger, would today be a valuable commercial or military carrier, not equaled in long-range transport by any plane yet in service. Airplane designers are, of course, men with imagination and tell us how near they are to something still better, but there are also men, like Mr. Upson, who can foresee great airship improvements. It is such men who make progress, but let us not confuse ourselves by comparing what has been done in one field, with what is just ahead in another.

PREROTATION f

HIGHER landing speeds and bigger, and therefore heavier, landing wheels have made it desirable to start the wheels rotating before they hit the ground. Such a procedure would save tire treads and lengthen tire life.

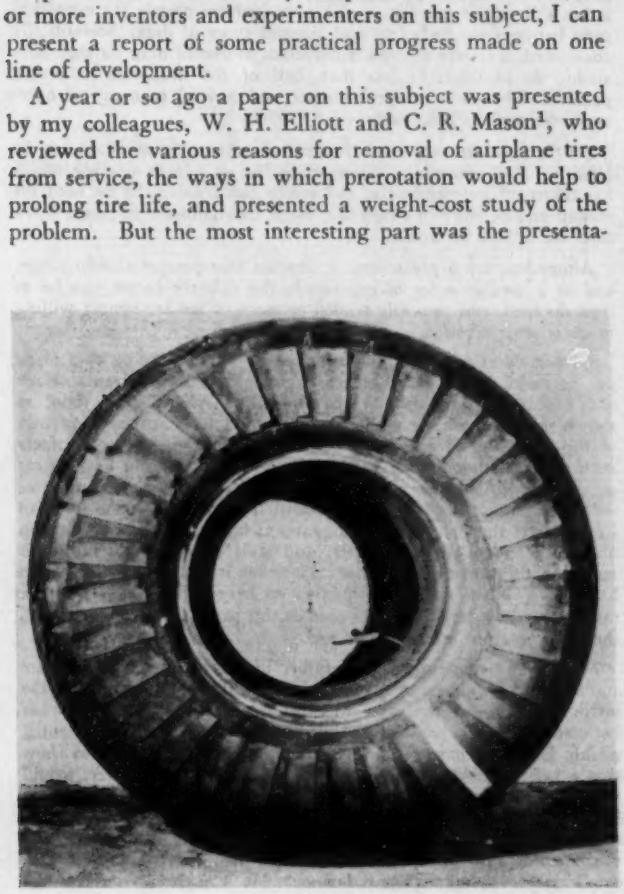
Mr. Schippel discusses here the latest developments on one method of accomplishing this, namely, by equipping the tires with flexible vanes that start the tires rotating as soon as the wheels are lowered for landing.

■

THE AUTHOR: HENRY F. SCHIPPEL (M '27) is manager of the aeronautical section, field engineering department, Tire Division, B. F. Goodrich Co. Mr. Schippel has had varied engineering experience, starting his career in Canada as a lecturer at McGill University, his alma mater, and progressing to research engineering for Canadian Consolidated Rubber Co. and Ames-Holden McCready. He joined B. F. Goodrich in 1925 on coming to the United States, and has been there since. Among the several technical papers which Mr. Schippel has written are: "Fabric Stresses in Pneumatic Tires," "Airplane Tires and Wheels," and "Rubber in Aviation."

FOR many years it has been the desire of airplane designers and operators to prerotate landing wheels so as to effect economies in design and maintenance. The mere physical operation of rotating a wheel is very simple matter indeed, but when additional weight is involved, the problem can become very complicated. Of the hundred or more inventors and experimenters on this subject, I can present a report of some practical progress made on one line of development.

A year or so ago a paper on this subject was presented by my colleagues, W. H. Elliott and C. R. Mason[1], who reviewed the various reasons for removal of airplane tires from service, the ways in which prerotation would help to prolong tire life, and presented a weight-cost study of the problem. But the most interesting part was the presentation of results obtained by Boeing Aircraft Co. on a pair of our 19.00-23 tires equipped with flexible vanes. (See Fig. 1.) Here at last was the first indication that we were on the right track. By the addition of only 5 lb of material to the airplane, we had succeeded in obtaining a prerotation of 68%. This figure represented the ratio of surface speed of the tire to indicated air speed at moment of contact with the ground. In level flight the tires would spin at only 58% of the ias, but the final portion of the landing approach is made with the airplane decelerating, and the wheel deceleration being less than that of the airplane, the resulting effectiveness was, therefore, much higher.



■ Fig. 1 – 19.00-23 tire equipped with flexible vanes

The report concluded that the vanes were generally satisfactory in producing prerotation before landing, that they reduced severe cutting of the tire due to sharp objects encountered during ground contact, but that if the wheel assemblies were not accurately balanced, they would produce resonant vibrations at the higher air speeds. The distinctive feature of the vane design was that they were flexible, normally lying flush with the surface of the tire, and only opening when the airstream impinged upon their leading edge. The vanes on both sides of the tire were so arranged that they were opened by the airstream on the lower half of the tire, but on the upper half they closed and caused little more drag than that of the normal tire surface. Thus, shielding of the upper half of the tire became much less important.

The next step in this development was a test by TWA on two 17.00-16 tires on two of their DC-3 planes. These tires had 25 vanes cemented on each sidewall. At an altitude of 2000 ft, several level test flights were made, the landing gear being extended at 150 mph. After one run of 142 sec, the vaned tire was revolving at 54 mph, or 41% of the terminal air speed of 132 mph, and on another run, 32 mph or 32% of the terminal speed of 100 mph. On an actual landing, the tire was revolving at only 17 mph when the last reading was taken over the end of the runway at an ias of 80 mph. This signified that the tire was decelerating faster than the airplane, which was believed partly due to the slight propeller power used during approach, the upper part of the tire being in the propeller slipstream. It was observed, however, that even with this somewhat low degree of prerotation, the tire screeched less than usual upon contact with the runway, and the drag force due to inertia of the wheel assembly was noticeably reduced. No conclusions were drawn from these tests, but the tires were left on the ships to study tread wear. It was later observed that because the tires on the DC-3 are not fully retracted, they slipped continually against the flight brakes which were not strong enough to resist the driving force of the vanes on the lower part of the tire.

[This paper was presented at the SAE National Aeronautic Meeting, New York City, April 7, 1944.]

[1] See "Prerotation of Landing Wheel Tires," by C. R. Mason and W. H. Elliott, presented at the SAE National Aeronautic Meeting, New York City, April 9, 1943.

f LANDING GEAR WHEELS

by HENRY F. SCHIPPEL
B. F. Goodrich Co.

These brakes wore grooves in the tire treads and the test was discontinued.

The next phase of development was a study of vane design and improvement in the method of attachment. It had been found on previous tests that the cemented vanes tended to strip off of the tire sidewall. Accordingly, on all vaned tires of later manufacture, the uncured vanes were attached to the uncured tire, and the whole assembly cured in the regular tire mold, adhesion of the movable part of the vane being prevented by holland cloth.

This change greatly improved the appearance of the finished product, as well as the streamlining, by providing recesses for the vanes to retract flush with the tire sidewalls.

Wind-tunnel tests were also made to determine the effect of varying the size of vanes and their angular position on the tire sidewall, and the stroboscope was used in conjunction with these tests to observe the vane action. As a result of this study, two pairs of 17.00-16 tires were made, one pair having radial molded vanes, and the other pair angular molded vanes. (See Fig. 2.) In both cases, 18 vanes were applied on each side of the tire. These tires were tested in the fall of 1943 by Eastern Air Lines, Inc., at Miami, Fla., and some very interesting results were obtained. Tire speeds were obtained by electric tachometer installation, as shown in Fig. 3. Test flights were made at constant air speeds to compare the efficiency of radial and angular vanes (Fig. 4), and it was found that while the angular vanes produced more rapid acceleration, the radial vaned tires reached higher terminal speeds. Further tests were then run with the air speed held constant at 150 mph for 1 min (Fig. 5) and then the air speed was reduced, simulating a landing approach. At the end of the 1-min, constant-speed run, the radial vanes again showed their superiority in speed by running 43% faster than the angular vanes, but on the decreasing ias part of the run, the angular vaned tires showed much less drop-off in speed. Test flights were made with the brakes applied to observe the vane action, and the results are shown diagrammatically in Fig. 6. On both the radial and angular vanes, it was noted that the vane at the lowest part of the wheel was completely closed. The vanes immediately ahead of and behind this one were 100% open and the next two vanes adjacent were partially open. This is no doubt caused by turbulence set up by the front vane, preventing the next vane behind from opening.

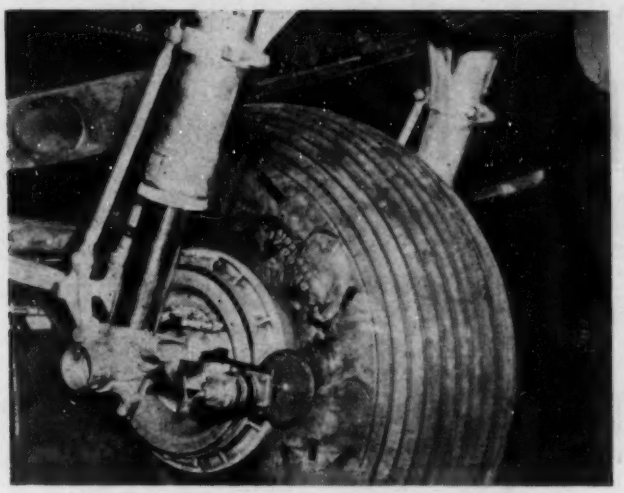
Consequently, it was believed that every alternate vane might be eliminated, and the same effectiveness obtained. Therefore, further constant ias tests were made on the angular vanes by cutting away one-half of the width of every alternate vane and also with these half-vanes removed completely, leaving nine full-size vanes. As a result of these preliminary tests, it was decided to run regular airplane landing tests, using 18 radial vanes, 18 angular vanes, and nine angular vanes, respectively. See Table 1.

You may find some inconsistencies in this table, but the figures do, in general, reflect for actual landings the comparisons already made in the steady-flight tests. It is interesting to study also the graphs of ias and tire speed for complete landings, such as those shown in Fig. 7 for the tires having nine angular vanes per side. These are the landings for which the highest prerotation speeds (80%) were achieved to date. Ground wind does not enter into this picture. Actually there was none, but had there been any, it would have directly reduced the gap between prerotation speed and actual airplane ground speed.

If I were asked to write a specification for a DC-3 landing procedure to secure the maximum degree of prerotation, I would say that the wheels should be lowered at an ias not less than 150 mph, that the landing glide should not exceed 2 min, and that the maximum deceleration should occur just before ground contact. No vibration trouble was experienced during all the tests, and it should be noted that these tires were statically balanced within 30 oz-in. as compared to the usual limit of 52.

Service tests are now being run by a number of different airlines, but at this time not enough data have been accumulated to enable us to make a comprehensive report. Most of our efforts to date have been directed along the line of development to obtain maximum prerotation effect, but we do have a few isolated tests completed which have shown very considerable tire life increase, as well as re-

■ Fig. 2 – 17.00-16 tire equipped with angular molded vanes



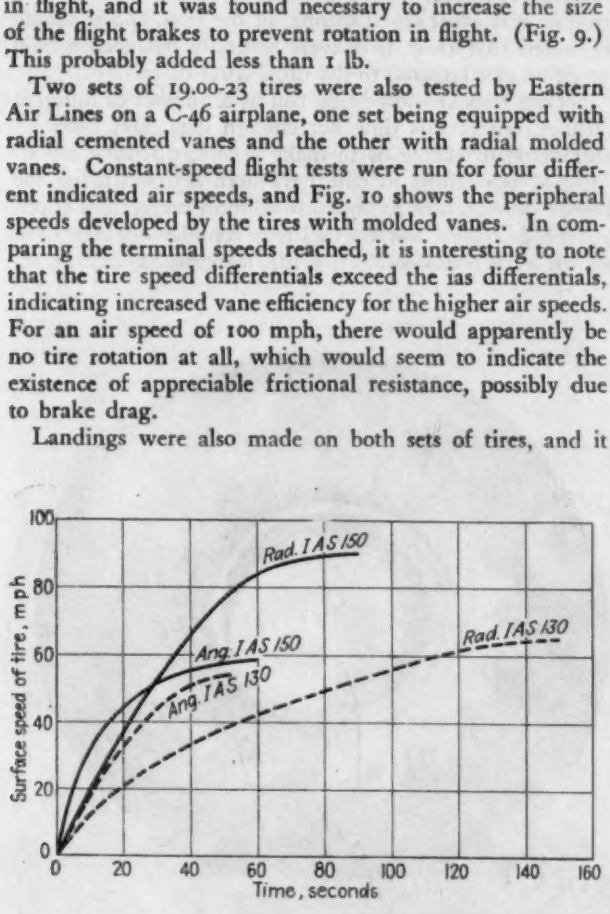
■ Fig. 3 – Electric tachometer installed to obtain tire speeds

duced wheel breakage and landing gear maintenance. A present-production, nine-vane tire is shown in Fig. 8.

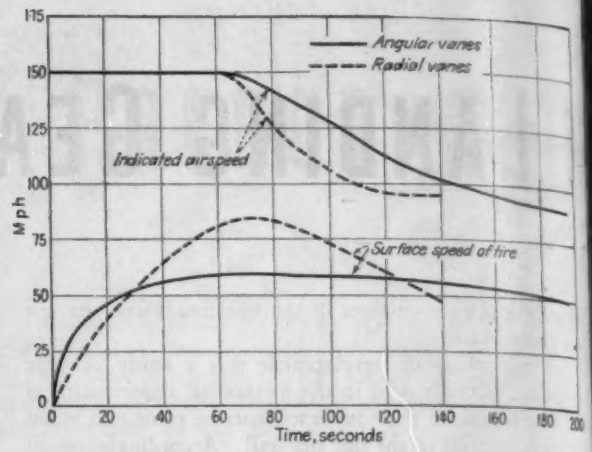
Regarding the very important matter of weight increase, it should be noted that the nine-vane construction increases the tire weight approximately 2 lb, which is less than the difference often found between two tires of the same make. On the DC-3 plane, the tires are not completely retracted in flight, and it was found necessary to increase the size of the flight brakes to prevent rotation in flight. (Fig. 9.) This probably added less than 1 lb.

Two sets of 19.00-23 tires were also tested by Eastern Air Lines on a C-46 airplane, one set being equipped with radial cemented vanes and the other with radial molded vanes. Constant-speed flight tests were run for four different indicated air speeds, and Fig. 10 shows the peripheral speeds developed by the tires with molded vanes. In comparing the terminal speeds reached, it is interesting to note that the tire speed differentials exceed the ias differentials, indicating increased vane efficiency for the higher air speeds. For an air speed of 100 mph, there would apparently be no tire rotation at all, which would seem to indicate the existence of appreciable frictional resistance, possibly due to brake drag.

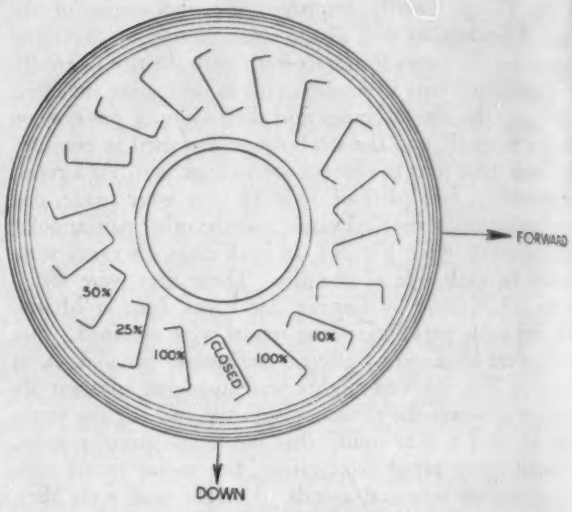
Landings were also made on both sets of tires, and it



■ Fig. 4 – DC-3 test flight at constant indicated air speed with 17.00-16 tires having radial and angular vanes (taken from report of Eastern Air Lines, Inc.)



■ Fig. 5 – DC-3 test flight with 17.00-16 tires having vanes (taken from report of Eastern Air Lines, Inc.)



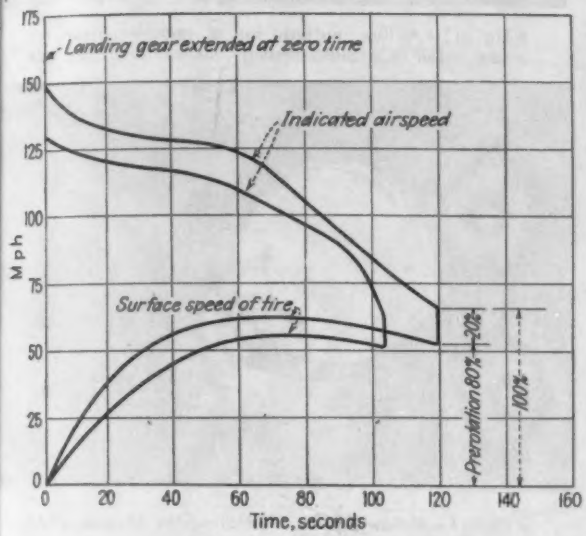
■ Fig. 6 – 17.00-16 tire with angular vanes

Figures shown on flaps are the approximate per cent opening – wheel stationary at 150 mph indicated air speed

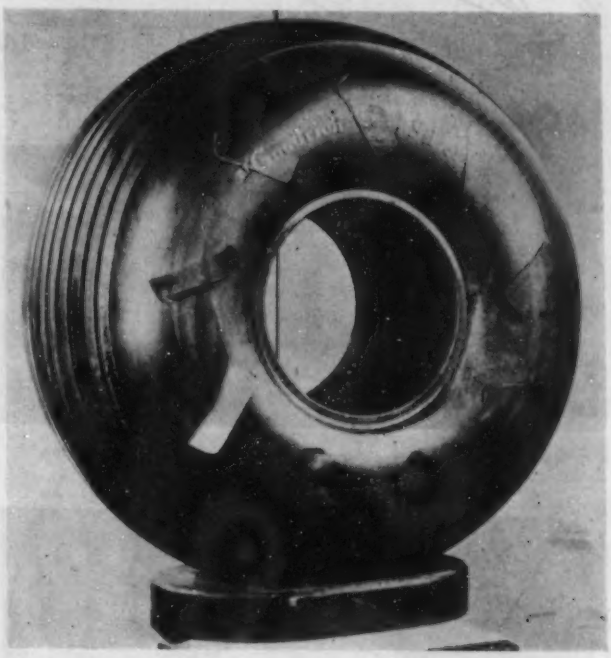
will be observed in Table 2 that in both cases, to achieve a minimum of 50% prerotation, it is necessary to lower the landing gear at an ias not less than 140 mph. For 150 ias, the molded vanes developed 58.7% prerotation and the cemented vanes 60.8%. This is believed to be due to the greater flexibility of the cemented vanes, these tires having had approximately 200 hr of service before the test. It was decided to determine if any vibration would be noticed at an increased wheel speed. Therefore, the landing gear was lowered on level flight at 150 mph and the air speed increased to 160. As the wheel reached a tire peripheral speed of 87 mph, slight vibration was noted, and when speed reached 98 mph, the vibration was excessive and brakes were applied to stop the wheel rotation. This vibration was, no doubt, caused by wheel assembly unbalance, as no precautions had been taken to balance them. However, these tire speeds were in excess of prerotational speeds that would be developed during normal landings, and it was, therefore, the general opinion that no trouble should be encountered in normal service operations.

Table 1 – Summary of Prerotation Test with 17.00-16 Tires with Molded Vanes (By Eastern Air Lines, Inc., on DC-3 Plane)

Vanes		Ias at Gear Extension, mph	Maximum Tire Speed, mph	Time to Reach Maximum Tire Speed, sec	Landing			
Number	Type				Tire Speed	Ias	Per Cent Prerotation	Time from Gear Extension to Ground Contact
18	Radial	140	55	46	43	80	51	82
18	Angular	130	52	70	44	70	63	112
18	Angular	140	54	50	49	69	71	74
18	Angular	150	64	54	52	75	69	110
9	Angular	130	55	70	51	64	80	103
9	Angular	150	62	72	52	65	80	120



■ Fig. 7 – DC-3 landings with 17.00-16 tires having vanes (taken from report of Eastern Air Lines, Inc.)



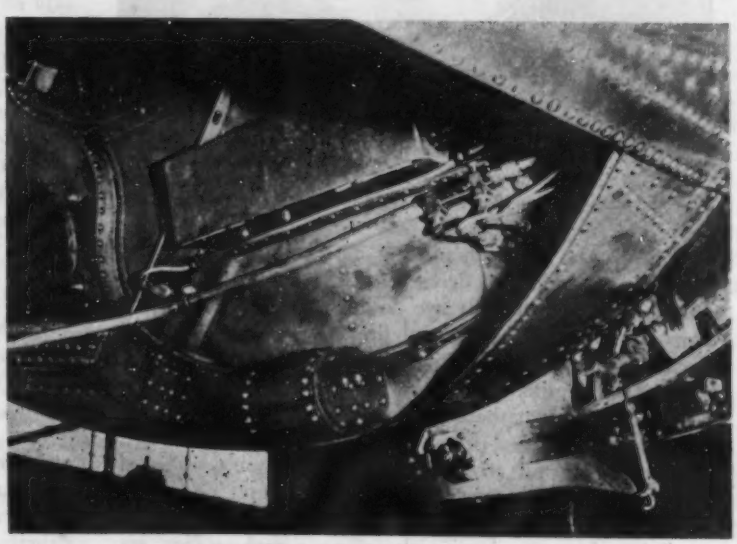
■ Fig. 8 – Nine-vane tire now being produced

Flight tests supplementing wind-tunnel tests were also made on 17.00-16 tires with vanes set at reverse angle to that now generally used. For a landing made with wheels extended at 145 ias, the prerotation was only 15%. Comparing this with 80% prerotation obtained with vanes set at the present adopted angle, it is interesting to speculate upon the cause of this wide difference. With the vanes set at angle as in Fig. 6, they are normal to direction of airflow in the trailing lower quadrant of the tire, whereas the vanes set at reverse angle are normal to the airflow in the leading lower quadrant.

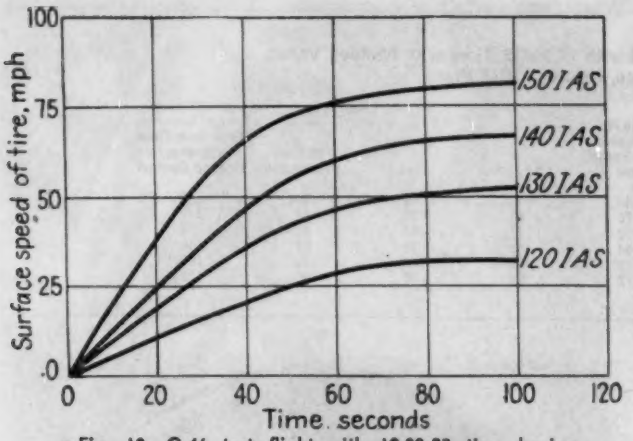
The explanation of the greater effectiveness of the airstream at the trailing half of the tire might be found from a study of the airflow pattern around the tire, and Figs. 11 and 12 show the airflow taken in planes passing through the center of a random model tested in a wind tunnel. Fig. 13 shows another airflow pattern, this one being for a smooth-contour tire model, taken in a plane through the center of the axle. Possibly also the Magnus effect, which is the flow distortion caused by a rotating body in an air-

Table 2 – Summary of Prerotation Test with 19.00-23 Tires with Radial Vanes (By Eastern Air Lines, Inc., on C-46 Plane)

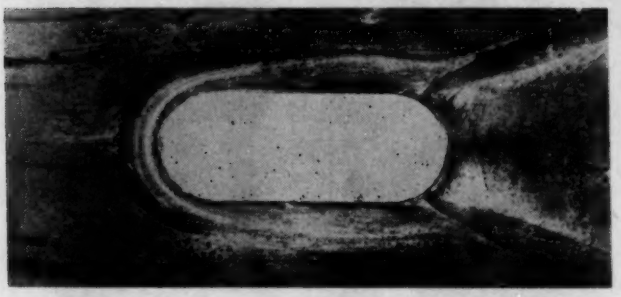
Ias When Gear Lowered	Prerotation, %	
	Molded Vanes	Cemented Vanes
150	58.7	60.8
140	54.7	57.4
130	41.6	49.4



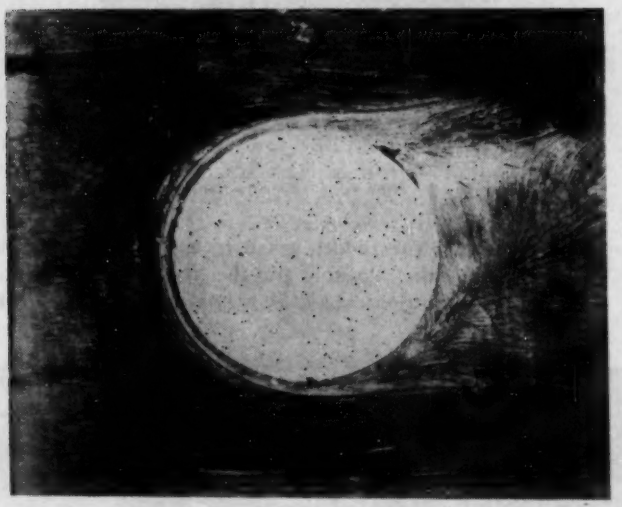
■ Fig. 9 – Flight brake used to prevent rotation in flight



■ Fig. 10 – C-46 test flight with 19.00-23 tires having vanes (taken from report of Eastern Air Lines, Inc.)



■ Fig. 11 – Airflow pattern around a tire taken in a plane passing through a model tested in a wind tunnel



■ Fig. 12 – Airflow pattern around a tire taken in a plane passing through a model tested in a wind tunnel

stream, may be involved, and Fig. 14 shows how this effect might cause prolonged driving force in the trailing lower quadrant. Unfortunately, we do not have any airflow patterns taken in any but central tire planes. Such patterns would be difficult to obtain, as the object upon which the pattern is deposited must not interfere with the airflow.

It is obvious that there are many variables affecting the performance of vaned tires, and for optimism performance, the vane design for any given size of tire should be based on the airflow characteristics of the airplanes on which that tire will be used. However, it is hoped that by determining the general principles of vane performance, we can establish a uniform design that will give generally

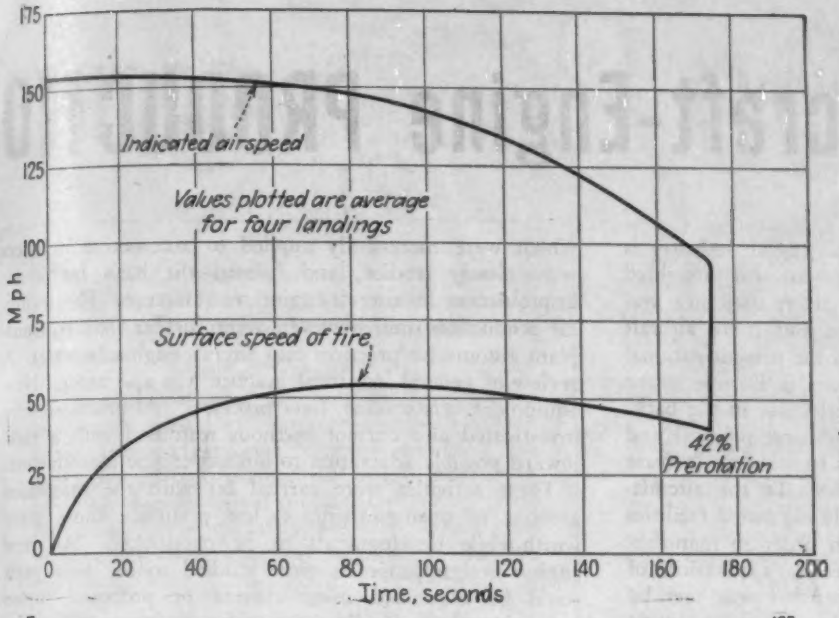
■ Fig. 13 – Airflow pattern for a smooth-contour tire model taken in a plane through the center of the axle

■ Fig. 14 – Airflow pattern illustrating the Magnus effect

satisfactory results. It will be necessary that the airplane designer be familiar with these various factors, so that he may provide the same advantageous flow of air around the tires as for the wings and control surfaces.

We also wish to mention the results of recent tests made by Consolidated Vultee Aircraft Corp. on the 56-in. smooth-contour, vaned tires on the B-24D airplane. For steady-flight condition, with landing gear extended at 120 mph, the tire prerotation was 38%, and at 187 mph it was 35%. With the landing gear retracted, due to the fact that one side of the tire was in the airstream on the lower surface of the wing, the prerotation was only 7 mph slower in each case. Quoting from the report, "There was no vibration or buffeting due to tire rotation during any of the test conditions, and the pilot stated that he could detect no gyroscopic effect during turns. During four landing procedures the average acceleration (Fig. 15) was from 0 to 53 mph in 80 sec. During this time, the indicated air speed varied between 155 and 150 mph. During the rest of the landing procedure, the air speed and the speed of the tires decelerated slowly until at contact with the ground, the air speed was 96 mph and the tire speed 40 mph, or 42% prerotation. The pilot commented that the landing shock was reduced considerably. It was observed that the smoke patch, caused by tires skidding upon contact with the ground, was reduced by two-thirds."

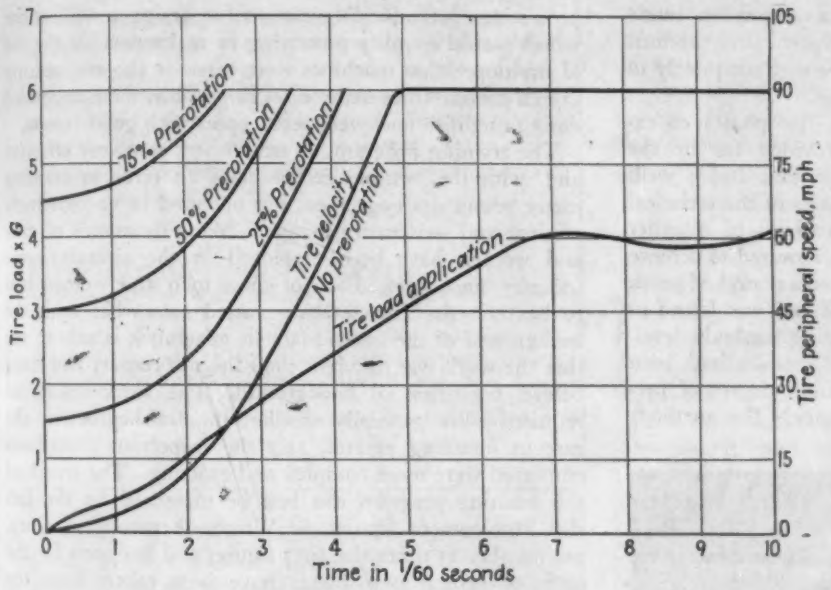
It is interesting to observe that while we have reached an effectiveness of 80% in the case of a DC-3 installation, the margin upon which our future efforts will be concentrated is fortunately not limited to the remaining 20% deficiency, for two reasons: First, the driving force on the vanes is concentrated at the tire sidewalls and, therefore,

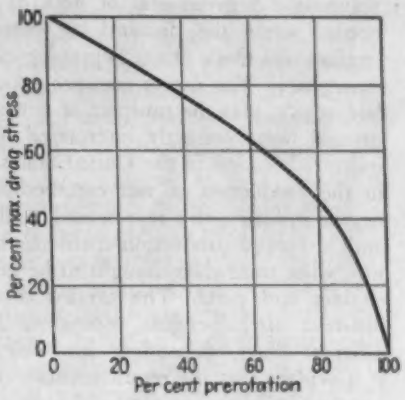


■ Fig. 15 (left) – B-24D landings with 56-in. tires having vanes (taken from report of Consolidated Vultee Aircraft Corp.)

■ Fig. 16 (lower left) – Model A-20 landing gear drop test

Solution for wheel acceleration-time; μ assumed = 0.55 and constant; tire rolling radius assumed constant; 44-in. smooth-contour tire





■ Fig. 17 – Model A-20 landing gear drop test

Solution for drag stress and prerotation; calculated from tire load application curve

at a radius smaller than that at which the tire speed is measured; and second, because the airplane speed is gradually reduced during the landing approach, the final tire speed includes a portion developed by inertia of the wheel assembly. But if 100% prerotation is not reached by this present method, to what degree will partial prerotation benefit the two factors of tire life and landing gear drag stress? The primary drag stress is generated by the frictional contact of the tire against the runway and, therefore, we may assume that the stress at any instant is equal to the product of vertical tire load and coefficient of friction.

For determination of the maximum value of this drag stress, I am indebted to K. W. Zahrt, of the Alighting Gear Unit of Matériel Command, Wright Field, for a practical procedure, as shown in Fig. 16. This method is based on data obtained from drop tests on the landing gear. The particular case is for the A-20 plane, which uses the 44-in. smooth-contour tire, and we start with the tire load application curve as shown on the chart. Assuming a constant coefficient of friction of 0.55 for the tire on the runway, and knowing the moment of inertia of the tire and wheel assembly, we can plot a curve of tire velocity-time, whose value at any time is proportional to the area under the load application curve up to that time. When this tire velocity curve reaches 90 mph, it cuts off horizontally because the tire has reached pull airplane ground speed and is no longer sliding. The drag stress at that instant is equal to the product of the coefficient μ and the tire load of 2.9g, or 1.6g. This is the condition for no tire prerotation. For 50% prerotation the drag stress is calculated at 70% of this value, or 1.12g, and Fig. 17 shows the complete curve of relationship between per cent maximum drag stress and per cent prerotation calculated on this basis. If we assume that tread wear follows the same trend, and there is good reason to assume it does, then tread wear, but only that part due to the landing slide, will be reduced 50% for a prerotation of 75%. But tires are not removed from service solely because the treads are worn out, but also because of cuts and other casualties, and as service tests already completed indicate general improvement from these other angles, we may reasonably expect substantial tire life increase for even moderate degrees of prerotation.

We will not venture an opinion as to how much prerotation is essential for any particular case, but we will say that this particular method appears to have a high degree of effectiveness on a weight basis, probably higher than can be expected for any other method so far considered, and therein lies its promise of accomplishing this very desirable improvement in airplane landing conditions.

Some Aircraft-Engine PRODUCTION ME

THE present position of the aircraft-engine industry is due to the exercise of sound judgment and farsighted planning. A great deal of both of these attributes was needed when the demand for more and more aircraft engines was made at the beginning of the present national emergency. The use of pressure politics in Europe in the late 1930's, plus the prospect of a world war in the background, were correctly interpreted by most political and industrial leaders in the United States to mean an increase in the production of war commodities. To the aircraft-engine industry, this represented vastly expanded facilities and improved production methods in order to manufacture what were then thought to be necessary quantities of engines and parts. The needed expansion was met by different aircraft-engine companies in almost as many different ways, depending on their particular product. It is evident that the requirements were met adequately in a most gratifying percentage of instances.

At the Wright Aeronautical Corp. the phases of expanded production facilities were provided for by the establishment of a technical production unit and a well-coordinated training program. The duties of the technical production unit were to study the problems of quantity production and recommend any action required to achieve increases. The automotive industry, over a period of years, had piled up a very satisfactory record and was found to have in operation the best mass-processing methods developed up to that time. For years automobiles had been produced in large quantities, with a high degree of precision being maintained where necessary. The methods which were successfully applied to automotive practices were closely studied, and formed the basis for much improvement in aircraft-engine manufacture. The technical production unit, however, went further than to transplant automotive practices into aircraft-engine factories. A review of general industrial practice was also made. New equipment was studied, new processes and practices were investigated and current methods restudied with a view toward possible adaptation to aircraft-engine manufacture.

■ Fig. 1 – Portion of cylinder-head department prior to changes affected by quantity-production demands – notice the large number of standard single-spindle drill presses

These activities were carried on with the maximum amount of open-mindedness, lest prejudice cause some worth-while developments to be overlooked. All new engine design proposals were studied to see how parts could be fabricated using current or proposed future processing methods. Suggestions for design modifications which would simplify processing or make possible the use of multioperation machines were made to the engineering design group. In as many cases as possible, these suggested design modifications were acted upon with good results.

The training program, as established, was very effective and, with the possible exception of an error in training many young draft-age men, has operated in an extremely efficient and satisfactory manner. Many thousands of men and women have been employed in the aircraft-engine industry since 1937. Each of these men and women had to be trained. This training varied somewhat with the background of the individual. In general, it is safe to say that the work was different than the newcomers had done before, regardless of background. The work tolerances required were generally smaller, the finishes better, the care in handling greater, and the inspection procedures employed were more complex and exacting. The worth of this training program can best be measured by the fact that employment figures at Wright Aeronautical Corp. are roughly 25 times the 1937 figure, and key men for the staffs of several new plants have been taken from the original nucleus. These expansion figures are somewhat distorted because no account is taken of vendor plant expansion and employment. The training of unskilled men and women and their ability to become in a short time members of a team producing aircraft engines is a tribute to the training program and the production methods employed.

The fact that quantity demands for engine parts were being made, and would continue to be made, is responsible for the machine tool developments that will be outlined in this paper. The need for large numbers of precision parts influenced the design and type of machine tool that would be needed to process those parts. The factor of a scarcity of skilled operators dictated that the maximum amount of automaticity and skill be built into the machine since it must make up for the lack of mechanical knowledge and training possessed by skilled operators. To face both a scarcity of skilled operators and a quantity demand for engines coincidentally, called for the use of more ingenuity than facing either of them separately, and resulted in the

[This paper was presented at the SAE War Matériel Meeting, Detroit, Mich., June 5, 1944.]

METHODS

by MARTIN M. HOLBEN
Wright Aeronautical Corp.

IN order to meet the need for tremendous quantities of aircraft engines quickly, aircraft-engine manufacturers had to develop many new techniques.

The author's company met the problem in two ways. First, it inaugurated a well-coordinated training program to make available the many new workers required; and, second, it established a technical production unit to study the problems of quantity production and then to recommend any action required to achieve increases.

How successfully this plan has worked can be judged from the accompanying story. Roughly 25 times as many people are now employed by the company as in 1937. Time, space, machinery, and skilled labor are all being used much more economically and efficiently because of the development of many automatic and multioperational machines.

■

THE AUTHOR: MARTIN M. HOLBEN (M '41) who, in the capacity of project engineer in the Production Model Division of Wright Aeronautical Corp., designs, tests and releases for production improvements found necessary by the Production and Service Divisions of the company. Since his graduation from the University of Michigan in 1934, Mr. Holben has been employed by Standard Oil Co. (Ind.), Borg-Warner Corp., Nash-Kelvinator Corp., Clark Railway Equipment Co., and Fairbanks, Morse & Co.

machine tool designing and processing methods shown and described in the following pages. That these tools and processes are filling the bill is attested to by the fact that they are in many instances now standardized throughout the industry. This fact alone is the biggest tribute that could be paid to the technical production unit, which is largely responsible for this record.

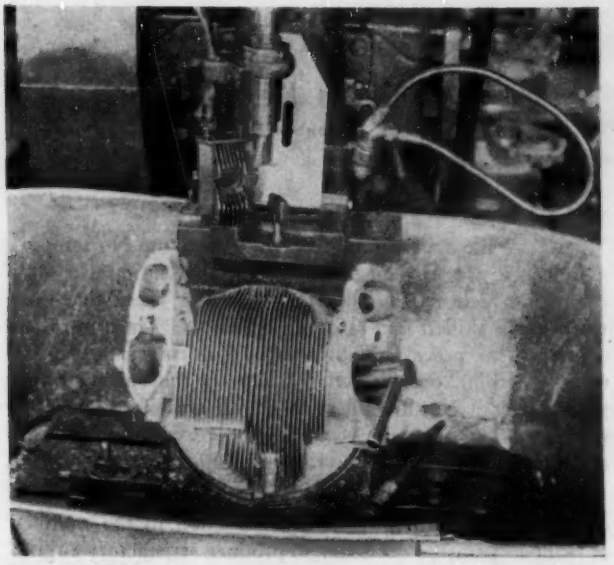
The efforts of the technical production unit also soon became evident in a material way. Semiproduction machinery was installed and began to produce aircraft-engine parts which had previously been turned out using standard production tools. Semiskilled labor began to produce parts which had formerly required skilled labor. Parts which had required much hand work, such as burring of sharp corners, blending of corner radii, hand buffing, and polishing, were processed on automatic equipment. These parts were equally as good as the parts previously produced, and the piece-to-piece variation was less than that experienced when using standard machine tools and skilled hand labor.

A specific instance where skilled hand labor was replaced is in the cam-o-lap automatic machine used to lap cam tracks. At one time the cam track was machine ground and polished, and then hand stoned to remove the "fuzz." With the help and cooperation of the Norton Co., a machine was developed to process the cam tracks in a manner which left no "fuzz." Each of these machines replaced 14 skilled hand-stoning operators. The completed cam track as processed now has a uniform surface finish of 13 micro-in. or less, where formerly the surface finish varied over a considerable range and was as high as 30 micro-in. This machine is finding universal application in the aircraft-engine industry for finishing cam tracks.

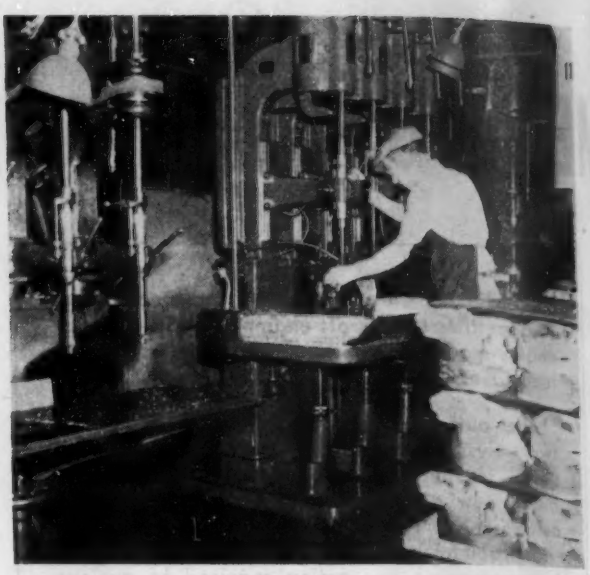
The success enjoyed with the use of semiproduction machine tools and the resulting reduction in the number of skilled operators necessary to produce satisfactory engine parts soon led to the procurement of special-purpose, quantity-production machine tools. In each instance where semiproduction or quantity-production machine tools were recommended for an application, it became increasingly desirable to reduce the amount of bench work required to process completely highly stressed engine parts. Every effort was directed toward manufacturing parts as completely machine finished as possible to minimize the human element which resulted in part-to-part variation.

A very interesting production development is represented in the processing of the Cyclone cylinder assembly. Some of the individual operations and the evolution of production methods over the past few years are briefly discussed in the following few paragraphs. A view of the cylinder-head department, in the early 1930's, is shown in Fig. 1. Notice the large number of standard single-spindle drill presses. A close-up of one of the single-spindle, drill-press operations is shown in Fig. 2. In this operation the spark-plug holes were drilled, reamed, spotfaced, countersunk, and tapped in the cylinder head. A "magic chuck" was used, and the operator changed tools for each of the operations. One of the first of what are termed "semiproduction" machines is shown in Fig. 3. Cylinder heads were loaded into the front of this machine and moved automatically from station to station, where spark-plug holes were spotfaced, drilled, reamed, countersunk, and tapped. Another single-spindle drill press is shown in operation in Fig. 4. This particular operation involved drilling, reaming, and tapping intake and exhaust flange attaching stud holes. The semiproduction counterpart of the single-spindle drill press shown in Fig. 4, is shown in Fig. 5. This machine was used for drilling, reaming, and tapping the intake and exhaust flange attaching stud holes. A special two-spindle drill set-up to machine the three concentric counterbores for the valve springs in the intake and exhaust rocker boxes of the cylinder head is shown in Fig. 6. In Fig. 7 is shown another semiproduction machine used to drill, ream, and tap four holes for the rocker box cover attaching studs.

A Greenlee automatic-transfer machine, the most recent evolution of the processing methods employed in cylinder-head manufacture, combines all of the foregoing operations. A great amount of thought and effort by the production engineers of the Greenlee Bros. Co. and Wright



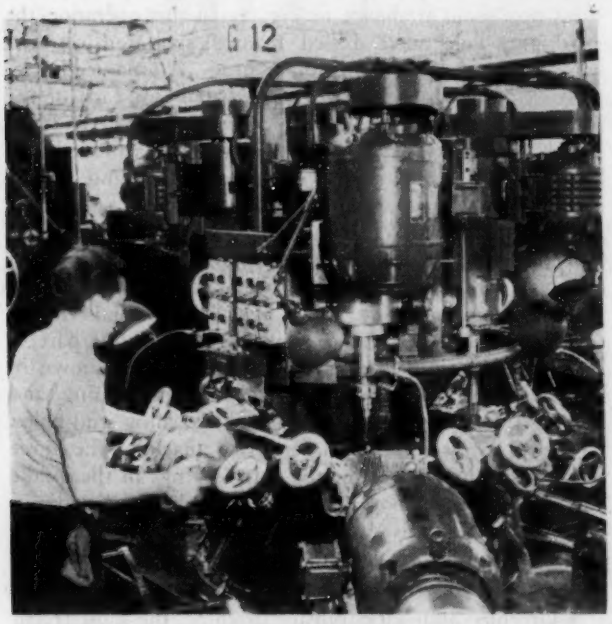
■ Fig. 2 – Close-up of one of the single-spindle, drill-press operations: drilling, reaming, spotfacing, countersinking, and tapping spark-plug holes – a "magic chuck" was used and the operator changed tools for each operation



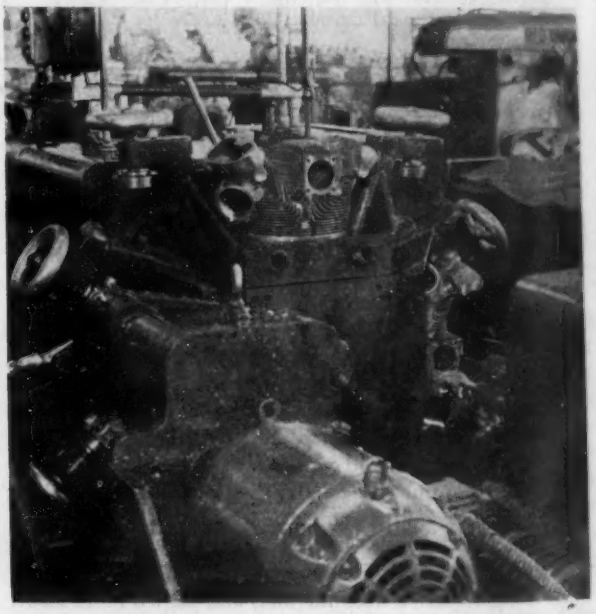
■ Fig. 4 – Another single-spindle machine in operation for drilling, reaming, and tapping holes for intake and exhaust flange attaching studs

Aeronautical Corp. organizations are represented in the development of the "Greenlee." A portion of this machine is shown in Fig. 8. This machine outmoded the semiproduction machines shown in the previous pictures, and has rendered obsolete the machines and fixtures employed on single-spindle drill presses used previous to semiproduction machine tools. In this machine, the cylinder heads are mounted on steel carrier plates and loaded onto one end of a conveyor running its entire length. During a one-way, 200-ft trip through this machine, the cylinder head receives over 70% of all the machining operations required in its manufacture. These operations were previously performed on 46 standard machines. In place of 46 skilled and semiskilled operators on each shift, three girls operate this machine and they can be trained in a few days. A master control panel seen behind the operator takes care of all functions and by means of colored lamps indicates the type and location of any trouble that might occur. Cylinder heads are removed from the discharge end of the conveyor at the rate of one every 45 sec. There are seven automatic-transfer type units in operation producing cylinder heads and assemblies at Wright plants.

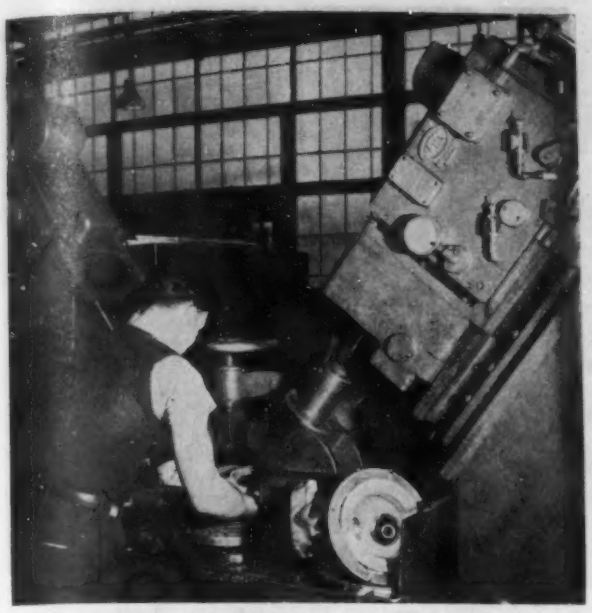
After the cylinder head is assembled to the cylinder barrel, the cylinder flange attaching holes must be drilled.



■ Fig. 3 – One of the first semiproduction machines on which spark-plug holes were spotfaced, drilled, reamed, countersunk, and tapped – cylinder heads are loaded at the front of the machine and index automatically from station to station



■ Fig. 5 – Semiproduction machine used to drill, ream, and tap holes for intake and exhaust flange attaching studs – to accommodate both front- and rear-row cylinder heads, drill units were made removable so that either four or seven drills, and so on, could be used (as shown, only four spindles are being used)

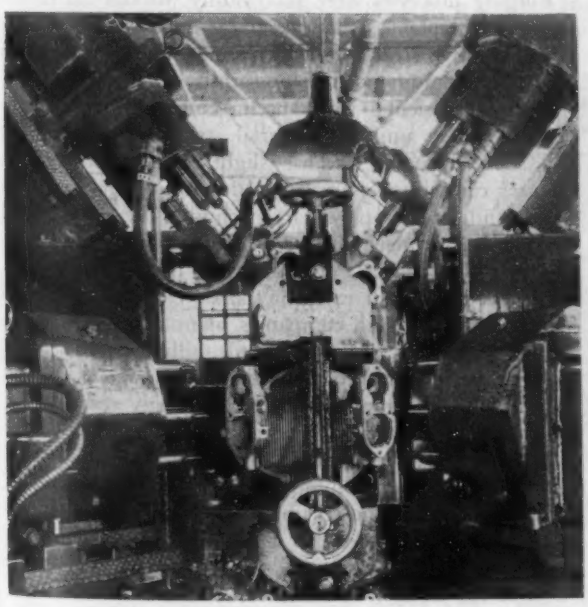


▪ Fig. 6 – Special two-spindle drill used for machining the three concentric valve spring counterbores on both the intake and exhaust sides of the head

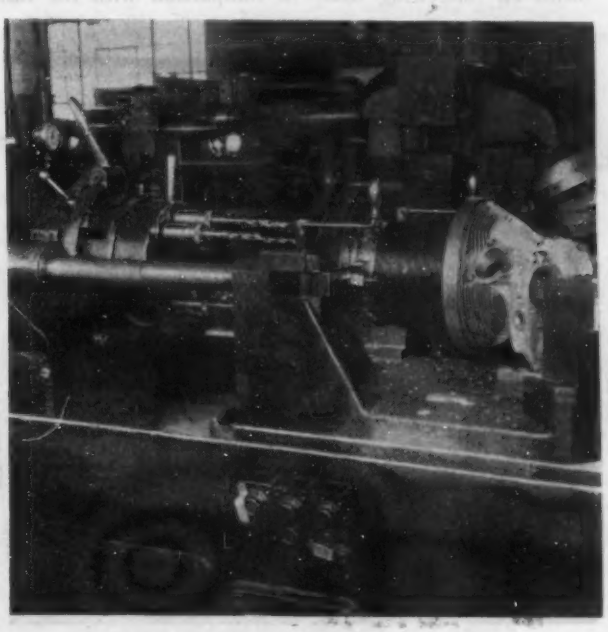
▪ Fig. 8 – Portion of Greenlee automatic-transfer machine – note master control panel behind operator, which takes care of all functions and indicates the type and location of any trouble that may occur

This drilling is done in the assembled condition, as the holes must be accurately located with respect to the intake and exhaust flanges. The original method of drilling the holes in the cylinder attaching flanges is shown in Fig. 9. Cylinder assemblies were drilled one at a time and then moved to a similar machine for reaming. Shown in Fig. 10, is a 40-spindle machine, which now combines the drilling and reaming operations required on the cylinder attaching flange, and does them in one-sixth of the time previously required. This machine features four groups of 10 spindles each, occupied by 20 drills and 20 combination reamers and countersinks, plus automatic indexing to present the cylinder attaching flange to each group in turn.

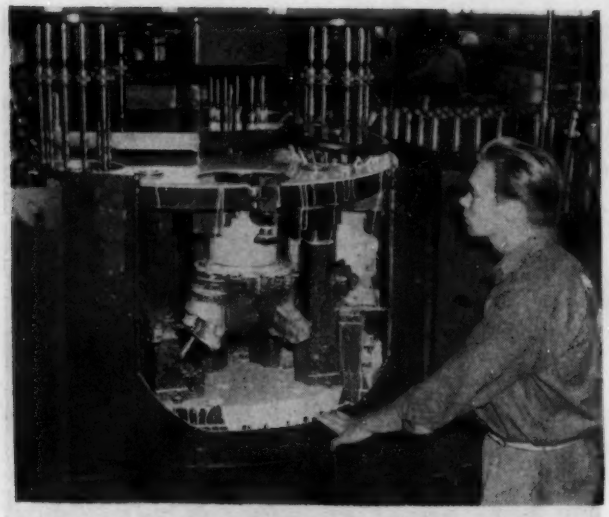
The completely fabricated cylinder assembly may be painted with black enamel or it may have the aluminum surfaces sandblasted and the exposed steel surfaces sprayed with a coating of pure molten aluminum. The enameling operation was formerly done on a one-at-a-time basis and required a prodigious amount of handling to complete. A continuous conveyor now moves the cylinder assembly, suspended by an adapter, through all cleaning baths and baking ovens. Coatings of primer and enamel are applied by dipping and spinning to remove the excess. These coatings were formerly applied by hand spray guns. Fin-



▪ Fig. 7 – Semiproduction machine for drilling, reaming, and tapping four stud holes in each of the rocker boxes



▪ Fig. 9 – Original method of drilling attaching holes in cylinder flange – parts were processed one at a time and were moved to another machine for reaming

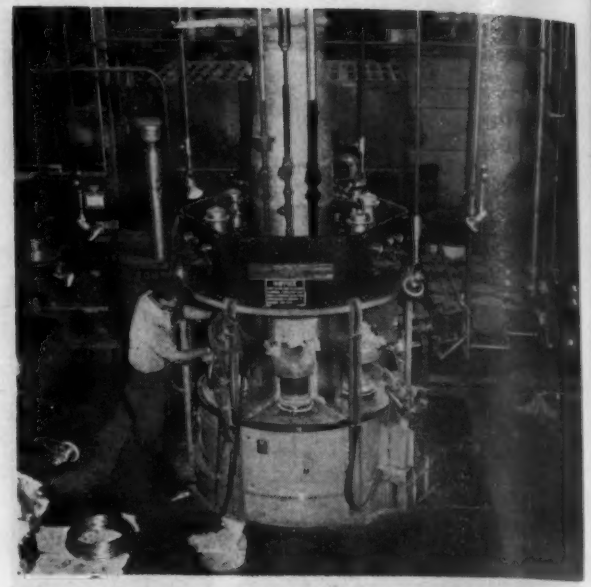


■ Fig. 10 – This 40-spindle machine combines drilling and reaming operations on attaching holes in cylinder flange

ished surfaces and inside surfaces are masked to avoid being covered. The alternate finish of sandblasting and metallizing was originally applied one at a time following degreasing and cleaning. The obsolete method of metallizing is shown in Fig. 11, and required about 15 min per cylinder assembly. Cleaning is now done in an automatically operated unit. Sandblasting is performed in an automatically indexing six-spindle machine. An automatic six-station machine is shown in Fig. 12, which applies a uniform metallized coating to the exposed steel surfaces of the cylinder assembly in 75 sec. One man loads and unloads the machine. The man at the left uses a hand gun to touch up the deep fins across the top of the cylinder head. This operation could have become automatic but the complication of a reciprocating gun station built into the machine was not felt to be justified.

The development of the forged aluminum cylinder head forms an interesting study in comparison with the cast

■ Fig. 11 – Metallizing or spraying cylinder assemblies with a coating of pure, molten aluminum to prevent corrosion is shown as originally performed – this job took about 15 min



■ Fig. 12 – Automatic six-station machine that metallizes a cylinder assembly in 75 sec – man at left uses a hand gun to cover a portion of the deep fins at top of cylinder head, for it was decided that the complication of a reciprocity spray gun station in the machine was not warranted to cover the deep vertical fins

aluminum cylinder head. Forged aluminum cylinder heads had been manufactured in limited quantities prior to 1937 by The Bristol Aeroplane, Ltd., of England. The forged aluminum cylinder heads at that time exhibited some properties which made them desirable from an operational standpoint. However, to produce usable cylinder heads from a forged billet and compete with established manufacturing practices used in the production of cast aluminum cylinder heads was an assignment of no small magnitude. Studies of current practices revealed that forged cylinder heads as produced were essentially toolroom jobs. After study by the production and design engineers, Wright purchased the first cylinder head forging dies in 1938.

Forging processes were successfully worked out to produce an acceptable billet weighing approximately 65 lb. An exhaustive study of heat-treating cycles was made to provide maximum possible uniformity of metal structure. As the experimental work on fin cutting progressed, the necessity for a uniform material structure became more and more apparent because slight metal condition differences caused an appreciable effect on cutter life at the speeds and feeds employed in machining the fins. Special equipment and machines were designed and built to profile the valve pockets. Profiling the intake and exhaust ports was considered early in the development, and provision was made with companies manufacturing machine tools to furnish the necessary profile-milling equipment.

The experimental work on the processes explained in the previous paragraph was extremely costly and of necessity had to be performed on modified standard machine tools. For instance, to machine fins on the forged billet, a standard knee and column type of milling machine was modified to relieve the friction load on the lead screws to less than 5 lb. Two Keller attachments were then connected to the table to control the horizontal and vertical travel of the table in order to cause it to follow a master profile. The first experiments in fin cutting were made, using gang milling cutters without much success, due to severe vibration of the work and the work table. Standard

milling saws were also tried with no better results being obtained. The vibration conditions encountered with the two methods mentioned were eliminated by the use of specially designed cutters 13 in. in diameter, incorporating nine carbide-tipped teeth, carefully ground and honed to a fine finish. When operated at a surface speed of 4500 fpm and a feed of 70 in. per min, these cutters produced a very fine surface finish on the sides of the fins. The selection of a coolant for use in cutting the fins was a difficult problem because the high centrifugal speeds of the cutter tended to throw the coolant off. Compressed air and tallow were the first successfully employed coolants. The tallow was injected into the work by intermittent blasts of compressed air. Sufficiently low temperatures were maintained to permit handling of the work without difficulty. Experiments with liquid coolants have now progressed to the point where they may soon be used in production, as compressed air and tallow are difficult to confine and difficulty is experienced in the cleaning of the cylinder head and the machines used to cut the fins.

As now quantity produced, the forged cylinder head has the intake and exhaust valve pockets and ports machine formed on special Snyder machines built to Wright Aeronautical Corp. specifications and special three-spindle high-speed Cincinnati hydro-tel milling machines. The cylinder heads are handled one at a time in the Snyder machines and three at a time in the Cincinnati hydro-tel milling machines. The fins are milled on Cincinnati milling machines designed especially for this job. These machines were developed from the experimental unit previously described. After forming the valve pockets and the intake and exhaust ports and milling the fins on the forged aluminum cylinder head, the condition of the head approximates that of a cast cylinder-head casting. Therefore, machines very similar to those used to produce cast cylinder heads may be utilized to complete the forged cylinder heads. The operations required to produce either the cast or forged heads are essentially the same from this point in cylinder-head manufacture.

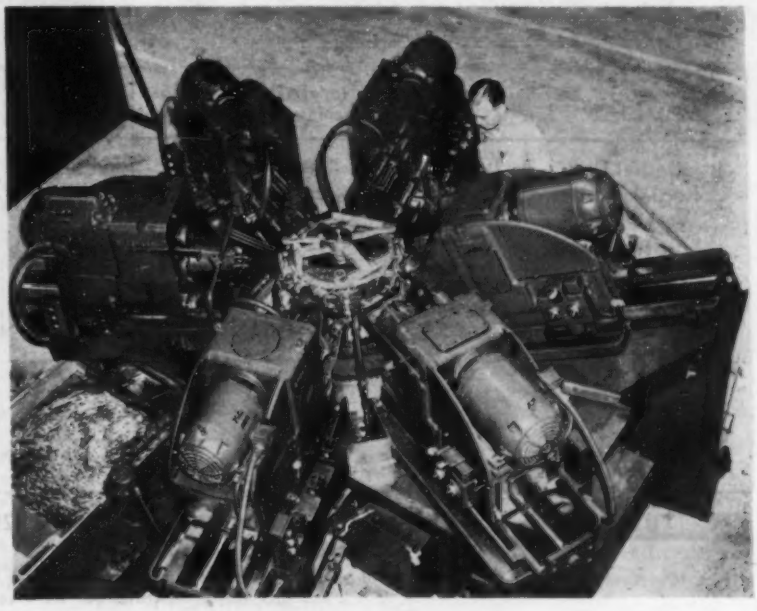
An interesting comparison of costs may be made between the forged and cast cylinder head, as it is noted that the increased machining time is more than offset by elimination of difficult foundry operations and casting scrap.

The supercharger front housing has undergone some interesting developments of tooling and production methods during the years when production has doubled and redoubled. Radial drills, radial tappers, and vertical mills have been replaced by special Greenlee horizontal and angular multistation automatic indexing machines for rough and semifinish boring, facing, and drilling of radial holes. Handling operations have been reduced approximately 65% and the number of men required per three-shift day has been reduced 90%. The total cost of the Greenlee equipment was approximately 50% more than for standard machine tools. However, in view of the saving in production time, this difference in cost was made up in two months of peak production. In Fig. 13, is shown a special six-way, 14-station automatic machine for drilling, reaming, boring, facing, and tapping intake pipe holes in a supercharger front section. This machine replaces seven standard machine tools and operators, and does the same jobs in one-twelfth of the time required on standard machines.

The analyses involved in the selection of quantity production machines take into consideration such items as space requirements, cost, productive man-hours, machine loads, and skilled labor requirements. An hypothetical figure for the machine load is established, based on a monthly production of 1000 engines. A month is arbitrarily defined as 720 hr. An allowance of 20% is figured for items such as scrap, set-up time, and all unforeseen delays. Typical analyses are shown in Figs. 14 and 15. These show respectively, the justification for the purchase of the Greenlee automatic transfer machine for machining cylinder heads, and the Greenlee horizontal and angular 14-station automatic indexing machine for machining supercharger front sections. In these analyses, the loading and handling time consumed by standard machines was noted to be greater than the actual time required for the machining operations, therefore unusual opportunities for savings were presented through the application of continuous automatic equipment and multistation, automatic indexing equipment.

The use of conveyor lines has been extensively employed throughout all plants to simplify the handling of parts. Temporary storage space necessary at a large number of standard machines has been eliminated, and the machining cycle inventory of parts in process has been greatly reduced. The savings in space effected by a reduced machining cycle inventory, and by a smaller number of machine tools, makes possible the most efficient use of floor space. The result becomes evident in a plant of smaller size, with more compact machine lines requiring less supervision.

So far, the mention of production developments and methods has been confined to relatively orthodox machining methods and machine tools encountered in an air-



▪ Fig. 13 – Special six-way, 14-station automatic machine, which drills, reams, bores, faces, and taps holes in a supercharger front section, and replaces seven standard machines and their operators – this operation is performed in one-twelfth the time originally required

Machine	Area Sq. Ft.	No. of Machines	Mach. Load /1000 Eng.	Cost of Machine	Cost of Fixtures	Handling Oper.	Per 3 Shift Day Men Req'd	Per 3 Shift Day Prod. Hrs.
Standard Machines	6,658	42		$286,500	$ 33,000	17	107	797.5
				$319,500				
Greenlee	2,890	1	0.323	$244,266	$ 69,000	2	8	7.8
				$313,266				
SAVINGS EFFECTED	3,768	41		$ 6,234		15	99	789.7

■ Fig. 14 – Process analysis covering Greenlee automatic-transfer machine for manufacturing cylinder heads

craft-engine factory. A development of an unusual nature is noted in the case of stud driving. By way of explanation, it is noted that each stud driven must be tightened to a certain range value measured in inch-pounds, and when driven home must project to a specified height. A photograph of a hand torque wrench is shown in Fig. 16. By previous practice, a stud was driven by hand with a standard stud driver to a height predetermined by the operator, at which time the torque wrench was applied to drive the stud to its required projected height. This stud must then be measured for projection within the torque limits specified for the size of stud employed. This work is now done by use of a torque driving machine manufactured by the Scandia Mfg. Co. The stud is driven to its correct projected height and released by the machine if the assembly torque is within the range value specified. If the torque is over that specified, the machine stops. If the torque at the correct projected height is less than that specified, the stud is not released and the machine automatically backs the stud out of the tapped hole. A picture of this machine is shown in Fig. 17. This machine was originally conceived for use in tightening railroad track interlocking plate bolts to a definite torque value, and was developed for the Pennsylvania Railroad. The adaptation of this machine for various uses in aircraft-engine production, such as driving studs, tightening crankshaft crankcheek clamp screws, and installing threaded exhaust valve seats, is an example of the open-minded manner with which the technical production unit operates in the investigation of new processes. It is interesting to note that tightening the above-mentioned crankshaft crankcheek clamp screws at one time required the use of a 7-ft wrench with three husky men on the end of it. The clamp screw was tightened until it had stretched a specified amount measured over a special ball-ended adapter. Injuries were frequent among men assigned to this job. Use of the Scandia machine has greatly simplified this operation.

The crankcases used on aircraft engines at one time were made from special aluminum-alloy forgings. As the horsepower output per engine cylinder increased, it became necessary to increase the thickness of the crankcase sections until a point was reached where the space required for the counterweights and other parts inside the crankcase was reduced to a minimum. Up to this point there was still no prospect of keeping the specific output per engine cylinder from going on up, and the crankcase performance was marginal. The limits of forged aluminum-alloy material had been reached and it therefore became necessary to turn to a stronger material. The first designs of steel crankcases were worked out on the basis of a steel forging machine profiled to reduce the weight as much as possible when completed. This practice was similar to that used on the superseded aluminum alloy crankcase. The steel crankcase, strangely enough, was somewhat lighter than the aluminum-alloy crankcase and was very satisfactory in operation. Subsequent designs of crankcases fabricated from steel have incorporated modifications which permit them to be processed more easily. This is made possible by surface contours inside and outside of the crankcase which permit the maximum percentage of sur-

Machine	Area Sq. Ft.	No. of Machines	Mach. Load /1000 Eng.	Cost of Machine	Cost of Fixtures	Handling Oper.	Per 3 Shift Day Men Req'd	Per 3 Shift Day Prod. Hrs.
Radial Drill	760	5	4.1	$ 32,500	$ 5,000	4	13	98.5
Radial Tapper	231	1	0.875	3,600	2,000	3	3	21
Vertical Mill & Rotary Table	243	1	0.81	10,500	1,000	1	3	19.5
	1,234	7		$ 46,600	$ 8,000	8	19	139.0
				$54,600				
Greenlee	480	1	0.355	$ 40,000		2	1	8.5
Greenlee	476	1	0.37	40,000		1	1	8.9
	956	2		$ 80,000		3	2	17.4
SAVINGS EFFECTED	278	5		$25,400 more		5	17	121.6

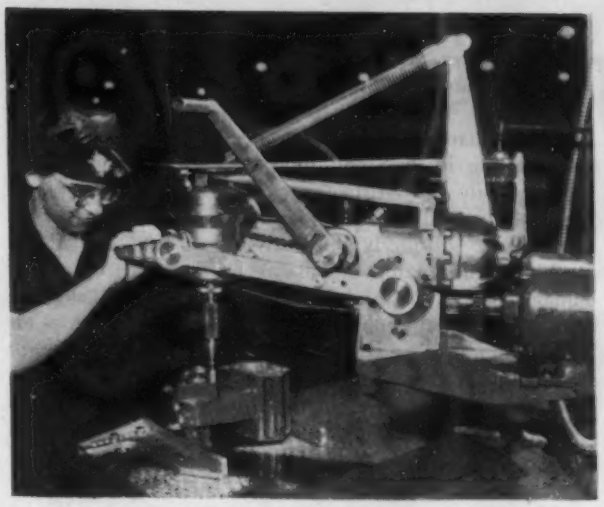
■ Fig. 15 – Analysis showing comparison of automatic multioperational equipment versus standard machine tools and covering machines for processing supercharger front section

face to be processed by turning and grinding, and require a minimum amount of milling.

Some real strides have been made in machining steel crankcases in quantity production. The crankcases are roughed out, semifinished, and finish ground to final size from forgings. The roughing and semifinish operations have, until recently, been performed on turret lathes. At the present time, special automatically indexing, four-spindle Bullard machines with 34-in. chucking fixtures are in operation replacing the turret lathes. These mammoth machines are approximately twice the size of standard Bullards and weigh around 40 tons. Each Bullard takes care of the work previously performed by five to six turret lathes, and in each 24-hr shift produce 2 tons of chips and 2 tons of roughed-out crankcases from 4 tons of forgings. There are now four of these machines in operation and eight more being installed. In addition to the machines and operators which the Bullards save, tool life has been longer than expected, due probably to the increased rigidity of spindles and tool holders.

Some other processes which have undergone revisions to adapt them to aircraft-engine use or have been adopted without change from other industries are worthy of mention. For instance, shaved gears are being employed to a considerable extent throughout the engine. The greater accuracy of tooth profile, spacing, and better surface finish, which are benefits of a shaved gear, have, in a number of instances, improved gear performance in the engine. The adoption of speeders to supplement conventional methods of inspection on production gears has had a beneficial effect on gear quality. This method of checking is to run a gear with its mating master gear under some load. A visual indication of tooth surface contact can be obtained, and an audible indication as to the quality of the gear may be noted by a trained operator. Actual engine operation is not simulated by the speeders, however, and the engine is still the best inspector for gears.

▪ Fig. 16 – Stud being driven by means of a torque wrench – notice the conventional stud driver on which the torque wrench fits (torque in inch-pounds is indicated on a dial on the handle of the wrench)



▪ Fig. 17 – Representative type of torque driving machine is shown here, where studs are being driven into an oil pump body – torque limits are varied by a simple adjustment on the machine

Thread rolling, though not a new process, through accurate control of blanks and machine equipment to produce quality threads is finding a real place in the industry. The testing of plated silver, steel-backed bearings and their use on a production basis is a graphic illustration of process development. When these bearings were demanded in production quantities, a trickle of usable bearings and a torrent of scrap resulted. As now quantity produced at Paterson, N. J., a scrap rate of less than 4% is experienced. All types of plating, the methods, and the equipment used are highly developed. Numerous developments and innovations have been worked into heat-treating processes to gain certain desirable ends. Developments of a nature to prevent distortion are very necessary to produce satisfactory engine parts. A new process to provide forced cooling after carburizing, in order to eliminate decarburized surfaces, has resulted in higher quality standards being maintained on engine parts. Induction hardening and brazing are also advantageously employed in processing some parts.

Production methods for cold working and work hardening the surfaces of some engine parts are being studied and put into operation. The requirements of processing equipment to control the exposure time of the selected surfaces to the effect of the medium used, are presenting new problems. Surface-finish specifications are a requirement on every stressed part in the aircraft engine, and the methods used to put the best possible finish on engine parts are continually improved. Examples of improved finishes in the engine are the thrust faces of gears, shafts, thrust retainers, and so on. These surfaces were once ground, but are now superfinished. This applies to both flat and spherical surfaces and is an instance where an automotive process was adopted by the industry without change. The production developments described in detail are but a few that could be treated here. Cylinder heads and assemblies, crankcase sections and assemblies, supercharger housing sections, crankshafts, master and articulated rods, gears, pistons, and numerous additional stressed engine parts, are now quantity produced by highly developed processing methods.

The scarcity of man-power generally had a marked effect on the industry. In order to keep up with the demand for parts, it became necessary to employ more and more female labor. This was made possible through the use of electric or hand hoists at each machine to handle heavy parts during loading and unloading. Female labor can now be easily and satisfactorily trained and used as operators. Mechanical lifting devices eliminate the necessity for "brute force" formerly required to load and unload the machines, minimizing the chance for personal injury and damage to machines and finished parts. These lifting devices removed some objections to female labor. Ideas for labor-saving devices generally originate within a department created for this purpose. Workers and supervisors now offer many suggestions for work handling which are developed by this special department.

Summing up the processing developments that have taken place and their effect on the industry, the overall picture is quite enlightening and improvements are found in unexpected quarters. The machines, of necessity, become more automatic and multioperational. Not as many machines are needed to meet production demands, and as fewer machines are needed, less space is required in which to install them. Conveyor systems and good work-handling devices to lift and transfer parts from one machine to another result in smaller machine cycle inventories and less floor space for storage. The number of operators and the supervision required have not increased in proportion to the number of parts produced. A smaller factory, more closely supervised personnel, and a higher quality product is the result. The ultimate in aircraft-engine production methods has not been attained, and to approach the ultimate you can be sure it will be due to the further application of human effort and ingenuity, which are the attributes responsible for the excellent showing presently made by the aircraft-engine industry.

AVIATION VAPOR LOCK

concluded from page 471

guide in the application of the fundamental V/L-pressure-temperature relationships previously established. With a knowledge of the pressure and temperature characteristics of any given fuel system, these relationships may be directly utilized to analyze the vapor-liquid conditions which may lead to vapor locking.

V. Conclusions

1. Aviation fuel systems will vapor lock with the evolution of small amounts of vapor, which may occur far below the boiling point of the fuel.

2. The evolution of vapor below the boiling point of the fuel is usually caused by the presence of dissolved air in the fuel.

3. The amount of air which a fuel will dissolve is characteristic of the fuel and can be determined experimentally or may be predicted with reasonable accuracy.

4. While the solubility of air in aviation fuel is not great, the vapor formation accompanying its evolution may be out of all proportion to the amount present.

5. The equilibrium amount of vapor which a gasoline will form for given initial and final conditions of temperature and pressure may be predicted by means of suitable equations presented herein. For the purpose, a nomograph applicable to 7-psi Rvp aviation gasoline is also presented.

6. With a knowledge of the temperature and pressure characteristics of a given fuel system, the vapor-liquid conditions existing throughout the system can be estimated in those cases where the initial condition of the fuel with respect to air saturation is known.

DISCUSSION

Questions Authors' Use of Reid Test

—C. L. Crawford

Gulf Research & Development Co.

It is to be regretted that the relationships between temperature and pressure and either specific volume or V/L ratio are not commonly measured, and hence, for the purpose of analyzing vapor-lock behavior they are calculated from the commonly available petroleum inspection data. The method proposed by the authors for calculating these relationships is the most reliable yet made available, being soundly based on the vapor pressure of the gasoline and the solubility of air in the gasoline. It is questionable, however, to the extent that it uses the Reid "vapor pressure" as an indication of the vapor pressure. Otherwise, the assumptions involved probably do not produce errors of significant magnitude when the calculation is applied to small values of V/L (less than 10) for a conventional aviation gasoline whose lightest component, say isopentane, is present in fairly large quantities (more than 10%).

Dr. O. C. Bridgeman's correlation of air solubility with density of the gasoline has been shown to be reliable. The method of obtaining relationships between vapor pressure and temperature from the Reid test is precarious at best, and the assumption that the vapor pressure remains constant while vaporization proceeds is not safe even within the limiting conditions just mentioned, and is definitely invalid when vaporization proceeds until more than one gasoline constituent is evolved, as in the case of motor fuel, the most volatile part of which contains many compounds. The weak spot in the proposed method of calculation is the use of the Reid "vapor pressure." The relationships between pressure, volume, and temperature for a gasoline are represented by a surface in a three-dimensional graph, and while the P-V-T surface for a gasoline may be used to determine where the Reid point will be (it won't be on the surface), it does not follow that the P-V-T surface or any part of it is fixed by the Reid test.

By definition, the Reid "vapor pressure" is not a property of the gasoline, although it is influenced by some properties of the gasoline. It measures the behavior of the gasoline in the presence of some air and some water, after some of the gasoline is evaporated, the amounts varying from one test to another. In so far as it is influenced by the properties of the gasoline, it can be used as a qualitative indication of that influence and, for what might be termed a homologous series of gasolines, it can be used comparatively, as for controlling the operation of distillation equipment in a refinery. However, it is inherently unsuitable for use in precise calculations and correlations; also, it is unfair as a specification of volatility because it penalizes some gasolines and favors others with regard to actual vapor-lock behavior.

The immediate value of Dr. Robertson's calculation method cannot be underestimated, but whenever it is desirable to know the P-V-T relationships applying to a sample of gasoline, it is most desirable to determine the data by direct measurement. Such results are as reliable as the precision of the method employed will allow, involving no assumptions, and there is little difference in the difficulty of making the measurement directly and the measurement of true vapor pressure from which the calculation can be made. It would appear feasible now to begin the general practice of measurement of P-V-T data using the apparatus described in Dr. Robertson's paper. It is none too soon to advocate the replacement of the Reid test by a measurement of greater significance.

FARM POWER and the POST-WAR TRACTOR

by L. B. SPERRY
Farm Tractor Division, International Harvester Co.

FOR several million or so years, man made his dog, cow, mule, and horse work for him, but this power did not relieve him of sweating over the food problem. He found the horse did the best job on the farm, and by 1918 he was using about 27,000,000 horses and mules in the United States alone.

Quite a while after Watt had made his steam engine someone put the boiler and engine, in combination, onto a wheel truck, and by using horses to pull it from place to place, made the first dent in the mechanical farm power problem.

It was only a step further until some brighter man put a belt around the engine pulley to a pulley on one of the truck wheels and sold his horses. The belt wasn't so good in bad weather, and the next thing was sprocket-and-chain drive to replace the pulley and belt.

These outfits were used mainly for threshing machine operations and for wood sawing, but one day a drawbar was added to the truck framework and a plow was hitched to the drawbar. Power plowing was here – plowing was the hard work of the farm.

The steam tractor was not too successful. It was very heavy with its boiler full of water and its wood or coal fuel. Furthermore, a licensed engineer was needed to run the engine. Water was not always good and frequently

[This paper was presented at a meeting of the Chicago Section of the SAE, Chicago, Feb. 8, 1944.]

ALTHOUGH the displacement of horses by tractors has been taking place on farms for only 20 years in any large amounts, tractor power is finding it increasingly harder to displace animal power. Mr. Sperry believes this condition is due to two factors:

1. Tractors cost too much for the operators of smaller farms.

2. Evidently there is still something lacking in the ability of the tractor to do varied farm work as suitably as horses.

To overcome these difficulties, at least in part, Mr. Sperry suggests the following as the most important requirements for tractors of the future:

1. Provision for quickly attaching or removing mounted farm implements.

2. Greater ease in controls for manipulating these implements.

3. Operator's comfort for long hours on the tractor.

4. Tractor sizes suited to the small acre farm and its limited income.

5. Engine combustion-chamber shape and vaporizing equipment to be studied intensively so that we provide the maximum in power per gallon of fuel, since it is evident that premium fuels are not going to be popular with farmers.

■ ■ ■

THE AUTHOR: L. B. SPERRY (M '20), whose 35 years in the tractor engineering field have earned him the title of Dean of Tractor Engineers, is the assistant to the vice-president of engineering, Farm Tractor Division, International Harvester Co. In 1908, a few years after graduating from the University of Minnesota, Mr. Sperry accepted a position with the Milwaukee Tractor Division of International Harvester, and has been with the company ever since as engine designer, construction engineer, tractor designer, head of tractor design work in the gas power engineering department in Chicago, and general supervisor of tractor engineering. Active in Society work, he is a member of the SAE National Tractor War Emergency Committee and the SAE Tractor & Farm Machinery Activity Committee.

■ Fig. 1—Various methods of plowing used by man

had to be hauled long distances, and a lot of it was required per day. Fires from the sparks were not uncommon, and boiler explosions were serious problems. However, the steam tractor was a real step forward and provided the natural search for an improved powerplant. A unit of less weight and free from the need for so much work to supply fuel and water became the aim of inventors.

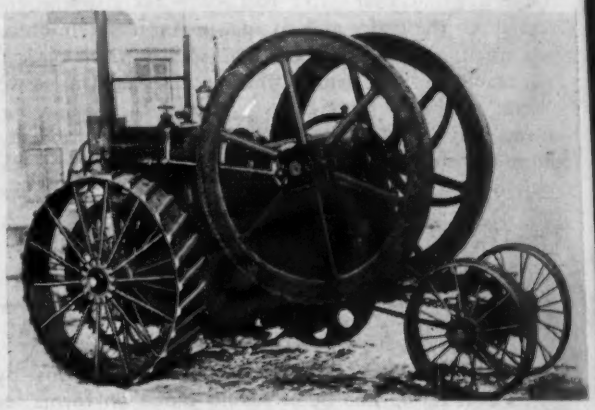
The gas engine had been known for years prior to the development of the steam tractor, but *gas* as a fuel was not the answer in the tractor field as it was a problem how to provide gas for a traveling powerplant.

The discovery in 1859 of oil in Pennsylvania paved the way for the needed fuel. Whale oil as a means of light had been in use for many years, and coal had been distilled as far back as 1681 for the tar and pitch it provided. Probably the need for these was to caulk ships.

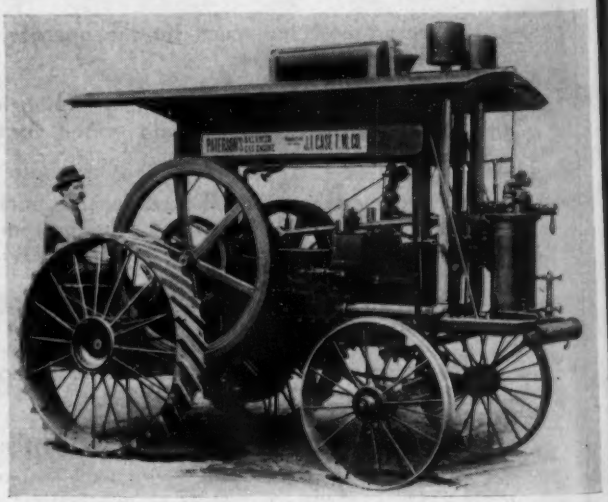
In 1845 Scotch shale rock was distilled, and from these developments came the means for using the petroleum – "oil from rocks" – that was discovered in 1859.

The first use of this oil continued to be for the tarry products. Later it was found that one of its less volatile elements was a good fuel for lighting. Kerosene became a national product. Naturally, there began to develop ideas that other fractions of petroleum must be useful, and lubricating oils were developed, but the lighter fractions were still a waste product. Engine makers soon proved that gasoline could be carbureted and burned in a gas engine, and from this developed the present method of spraying liquid fuel into an air stream for use in an engine. It was not until the Otto cycle patents expired, however, that much real progress in gasoline engine building became possible.

The need for kerosene in home lighting had built up great stocks of gasoline, so it was logical that gasoline would become the fuel that made possible the light power unit to replace the heavy steam tractor. The steam tractor had been an *important* factor in the farm power problem for 20 or more years before it was displaced, but in all this time, and for many years yet to come, the farmer's use for



■ Fig. 2—Burger tractor built in 1889



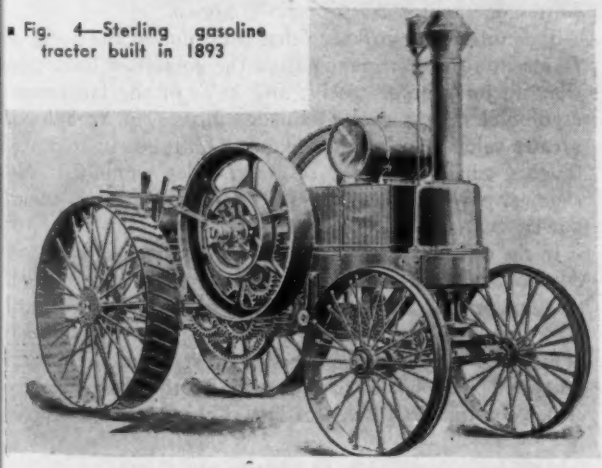
■ Fig. 3—Case tractor built in 1892

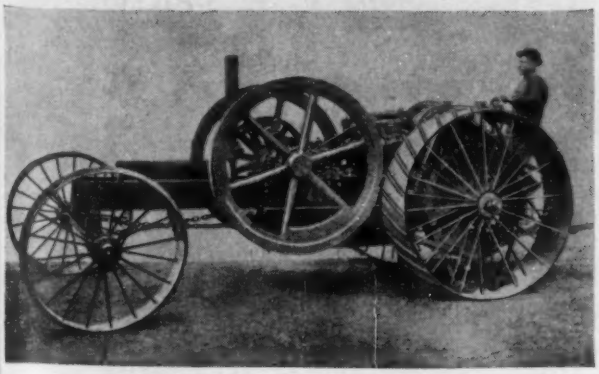
portable power was confined to plowing, as a field operation, and to belt work.

In Fig. 1 we see the various methods of plowing that man has used: the ox team plowing about an acre per day, the horse team plowing two acres per 10-hr day, and the tractor. The tractor speed was only equal to horse travel at first, but pulled more furrows. Today, plowing is approaching 4 mph, and approximately 5 acres per 14-in. bottom per 10-hr day is possible.

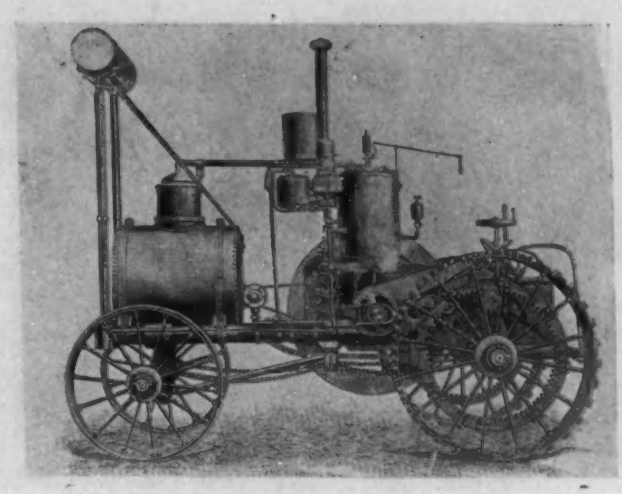
The Burger tractor shown in Fig. 2 was built in 1889 and operated around Madison, S. D. in 1890. Our L. F. Burger spent some time in the field with it. Six more were built about 1895 and placed in the Dakotas. A Charter single-cylinder engine was mounted on the chassis of a steam engine in the belief that a more ready acceptance would be given to gasoline power if the rest of the unit was of familiar construction. Practically all of these early models patterned their truck mechanism along steam tractor lines for this same reason, but, naturally, different engines were used. Very little attention was given to the dirt problem inherent in farm work, and lubrication depended almost entirely on the operator. Grease cups, or even wick-filled oil pockets, provided the main-bearing lubrication. Sight drip oilers were used for piston and connecting-rod bearings. Some of us know well how unsatisfactory these devices were. Gearing was entirely exposed to the elements, and lubrication was obtained by using a paddle and hard grease scraped into the gear teeth. Gears were cast-tooth type and generally made of cast iron. Naturally, these exposed and poorly lubricated parts wore out very fast.

■ Fig. 4—Sterling gasoline tractor built in 1893





■ Fig. 5—Otto tractor built in 1894



■ Fig. 6—VanDuzen tractor built in 1894

It is interesting to know that the request for these early gasoline tractor undertakings came to the various companies from farmers who evidently had used and were not satisfied with steam.

In 1892 the Case Co. built their first tractor, shown in Fig. 3. It was powered by a 2-cyl, opposed engine. Note how very much like a steam tractor this unit appears. The ignition mechanism was piston actuated so that the lateness of the spark caused bad engine economy and power. Case did no further work on gasoline tractors until 1910.

Fig. 4 shows the Sterling tractor, which was brought out in 1893 by a Mr. Hockett and a group of farmers near Sterling, Kan. It was built by Charter Engine Co. Note how closely it resembles the steam construction. No sales of this tractor were made.

Otto built a tractor in 1894, and 14 were made and sold in 1895 to 1896. See Fig. 5.

VanDuzen also built a tractor in 1894, shown in Fig. 6. This is reported to be the forerunner of the Huber line of tractors. This tractor was so much like a steam unit that it had a whistle as part of its equipment.

In 1899 S. S. Morton and the Ohio Mfg. Co. entered the field and built a friction-driven chassis with a hopper-cooled engine on what appears to be a specially designed chassis. See Fig. 7. This organization later built tractors for the International Harvester Co.

In 1901 Hart-Parr, using a heavy-type, aircooled, twin-cylinder engine, built their first tractor (Fig. 8), and report is that it was sold in 1902 to a farmer in Iowa where it worked for at least 17 years. In 1903 Hart-Parr built 15 tractors, and the success with these tractors was very good. Hart-Parr Co. deserve their recognition as really the founders of gasoline tractor activity.

In 1905 International Harvester Co., under the direction of E. A. Johnston, began experimental work on tractors, and in this same year the Ohio Mfg. Co. and the International Harvester Co. began to build tractors. The first one built was a 10-hp, friction-driven, forward and reverse unit. The whole engine was shifted on rollers by an eccentric lever control to accomplish the forward or reverse travel. This tractor is shown in Fig. 9.

In 1908 a really venturesome man built a tractor. Henry Ford, ignoring all designs of steam or gasoline tractors, built his model of a tractor, shown in Fig. 10. Evidently, he did not like the design, for it was 1919 before he tried

■ Fig. 7—Tractor built by S. S. Morton and the Ohio Mfg. Co. in 1899

to make and market his Fordson. Easily recognizable is his famous Ford car engine and radiator.

From 1908 to 1920 many companies entered the tractor field, but nearly all perished before or very soon after going into production. It remained for strong companies, with a line of agricultural machinery to sell with their tractors, to stay in and live down the inadequacies of their early ventures. In fact the darkest days of gasoline tractor life were making their appearance in the years just prior to World War I. It is quite probable that the war demand for tractors saved the situation.

Although pretty dependable tractors were being made by the end of the war, really protective engineering for engines and chassis had not been given necessary attention, and this was definitely needed if internal-combustion-engine power was to live.

In fact this problem of engine cylinder, ring, valve, and bearing wear from dirt entry became so acute a trouble to the tractor industry that about 1920, we developed and built a uniflow-type steam engine and a fully enclosed condensing system for the engine and boiler needs to have ready if the internal-combustion engine could not be made durable.

Fig. 11 shows a small steam tractor with protected mechanism, which never got past the experimental stage.

The McCormick-Deering 15/30 (Fig. 12) and 10/20 tractors were developed during 1918 and 1919, and went into production in the early twenties. They proved that tractors of sterling performance and long life, with low upkeep, could be made.

Almost concurrently, we developed a smaller sized similar engine as the power to use in our row-crop tractor, which became known throughout the world as the Farmall.

Enthusiastic reports from farmers who had operated our first 25 of these units brought about the decision to release it for sale in Texas, where it had been very much liked by the cotton growers, until we could produce the required volume for greater sales. Incidentally, our recently announced cotton picker, which mounts on our present Farmall "H" or "M" will solve the problem of cotton raising on a profitable basis despite the cheaper labor of foreign countries.

■ Post-War Tractor Market

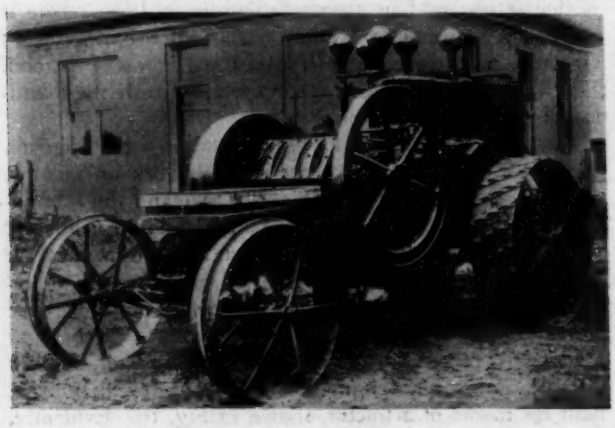
The introduction of the general-purpose type of tractor in 1923 was the major step toward replacement of horses and mules by mechanical power. Prior to this tractor type, which slowly proved its ability to cultivate nearly all row-planted crops, tractors on farms displaced very few horses. They merely relieved the horses of the hardest work, plowing, discing, and so on.

Horses were at their highest peak, in numbers, on farms in 1914-1918 (World War I demand for food) with about 27,000,000 listed in 1918, and only 85,000 tractors on farms at that time. In 1923, due to depressed farm conditions as much as to inroads made by four-wheel tractors, horses were reduced to 24,000,000, while tractors on farms reached 428,000. It took about 350,000 tractors to replace 3,000,000 horses or about 1 tractor to 8½ horses.

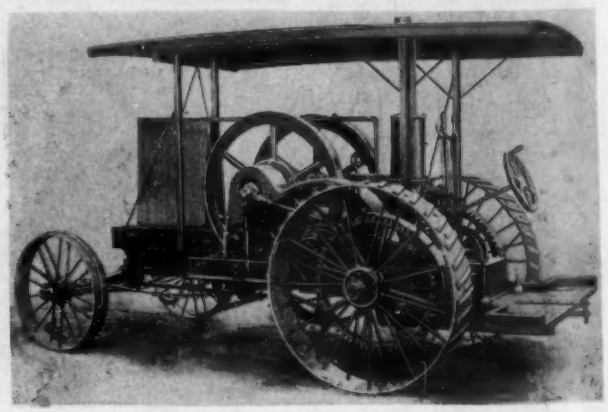
It is interesting to know that the value of the horses on farms in 1918 was greater than the *combined* book value that tractors, motor trucks, and 40% of the farm-owned automobiles have ever attained. Horses in 1918 had a greater value than horses remaining on farms in 1942, plus tractors, trucks and 40% of automobiles, combined. Mechanical power had been much less expensive than animal power.

In the next six years (1923-1929) tractors on farms nearly doubled, while horses were reduced only 20%. In this six-year period, approximately 400,000 tractors displaced 4,250,000 horses – a very high ratio of more than 10 to 1.

Between 1929 and 1942, horses were reduced from approximately 20,000,000 to 14,000,000 by the increase in use of 840,000 tractors. Even as late as this period, when the



■ Fig. 8—Hart-Parr tractor built in 1901



■ Fig. 9—10-hp tractor built in 1905

■ Fig. 10 – Tractor built by Henry Ford in 1908 – an experimental farm engine drawing a disc harrow

■ Fig. 11—Small steam tractor with protected mechanism, which never got past the experimental stage

Farmall type was so very popular, one tractor displaced only seven horses. Evidently the farmer has not yet found the tractor capable of fully displacing animal power, and this despite his use of automobile and truck for all manner of road work and hauling.

What is the reason that tractor power finds it increasingly harder to displace animal power? Two answers immediately appear:

1. Tractors cost too much for the operator of smaller farms.

2. Evidently there is still something lacking in the ability of the tractor to do varied farm work as suitably as horses.

Until we have implements that can be put onto or removed from our tractors without assistance of expert mechanics, and so simple in mounting that hours of work are not required, with a small machine shop needed to make parts fit, the horse and harness will remain on the farm.

It is reasonable to assume that medium and small farms must be providing much of the work that causes the continued use of horses.

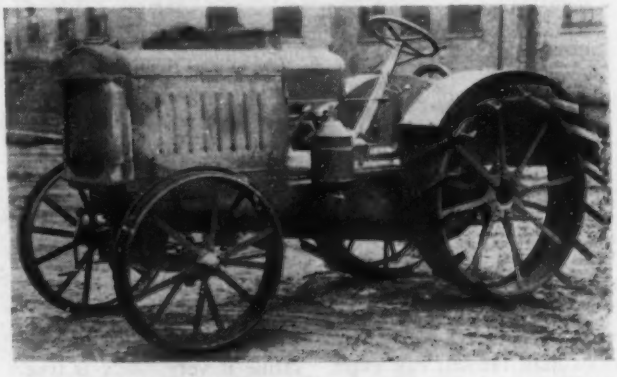
Our work will not be finished until we have tractor power on every farm. The market is at least three times as great as that part of it we have filled to date. We have 10,000,000 horses and 4,000,000 mules to displace, and there are over 4,000,000 farms not yet supplied with tractor power.

There is an immediate market, because the war effort has interfered with the industry's normal production and sales, conservatively estimated at 400,000 tractors, and each year the obsolete models are greatly increased. This demand equals two full years of industry's highest volume of tractor production, and therefore will afford at least two years to develop, perfect, and bring into production, new or post-war models. We will need this much time for our development work and to get into production on the new models.

However, existing models, in the face of such need, will find ready sale as soon as they can be provided.

We have prepared Table 1, which shows the disposition, by geographic sections in the United States, of:

Tractors, horses and mules.
Farms between 3 and 20 acres.
Farms between 20 and 50 acres.
Farms between 50 and 100 acres.
Farms between 100 and 500 acres.
Farms over 500 acres



■ Fig. 12—McCormick-Deering 15/30 tractor of about 1920

Table 1 – Distribution of Farms, Horses, and Tractors

(Data from 1940 census and a few 1943 estimates made by U. S. Department of Agriculture)

New England and Middle Atlantic							
Total Farms	Total Horses	3-20 Acres	20-50 Acres	50-100 Acres	100-500 Acres	Total Tractors	Over 500 Acres
135,000	131,000	30,500	23,600	29,400	47,400	34,000	1,800
348,000	607,000	64,000	55,100	91,700	135,000	156,000	2,200
483,000	738,000	94,500	78,700	121,100	182,400	190,000	4,000
1 farm	2 horses				1 farm	1 tractor	

East North Central and West North Central							
Total Farms	Total Horses	3-20 Acres	20-50 Acres	50-100 Acres	100-500 Acres	Total Tractors	Over 500 Acres
1,006,000	2,327,000	123,600	146,000	268,100	460,400	525,000	7,700
1,090,000	3,736,000	77,600	85,500	156,000	675,800	638,000	95,000
2,096,000	6,063,000	201,200	231,500	424,100	1,136,200	1,163,000	102,700
1 farm	3 horses				1 farm	1 tractor	

South Atlantic, East South Central, and West South Central							
Total Farms	Total Horses	3-20 Acres	20-50 Acres	50-100 Acres	100-500 Acres	Total Tractors	Over 500 Acres
1,019,000	1,470,000	210,000	285,200	253,200	255,400	81,600	15,500
1,023,000	1,707,000	250,000	310,100	232,400	220,000	81,200	10,500
964,000	2,410,000	167,000	222,500	196,300	330,400	233,600	48,000
3,006,000	5,587,000	627,000	817,800	681,900	805,800	376,400	74,000
1 farm	2 horses				2 farms	1 tractor	

Mountain and Pacific Coast							
Total Farms	Total Horses	3-20 Acres	20-50 Acres	50-200 Acres	100-500 Acres	Total Tractors	Over 500 Acres
233,000	1,075,000	37,400	28,500	29,000	77,700	90,600	60,900
276,000	485,000	102,500	64,200	35,300	52,400	109,000	21,300
509,000	1,540,000	139,900	92,700	64,300	130,100	199,600	82,200
1 farm	3 horses				1 farm	1½ tractors	
					2,254,500	1,929,000	

Need 325,000 tractors to have only one per farm for farms 100 to 500 acres in size.

States Comprising Geographical Areas Listed in Chart

New-England	West South Central	West North Central
Maine	Arkansas	Minnesota
New Hampshire	Louisiana	Iowa
Vermont	Oklahoma	Missouri
Massachusetts	Texas	North Dakota
Rhode Island		South Dakota
Connecticut		Nebraska
		Kansas

East North Central	Pacific	East South Central
Ohio	Washington	Kentucky
Indiana	Oregon	Tennessee
Illinois	California	Alabama
Michigan		Mississippi
Wisconsin		

South Atlantic	Middle Atlantic	Mountain
Delaware	New York	Montana
Maryland	New Jersey	Idaho
Virginia	Pennsylvania	Wyoming
North Carolina		Colorado
South Carolina		New Mexico
Georgia		Arizona
Florida		Utah
		Nevada

Examination of Table 1 should be made from two viewpoints:

1. That of the existing power distribution.
2. What must be accomplished to put mechanical power at work on the unsupplied farms.

The farm sizes in these groups are given in acres actually used for cultivation and crop growing; forest, swamp, mountain, and desert areas are not included.

Further, it must be apparent that any farmer who cultivates over three acres of ground has need for power greater than man himself can exert. While it very likely is true that an area of 20 acres, unless quite intensively worked, does not provide enough income to warrant tractor power, it cannot be denied that keeping animal power for such farms requires a very great part of the area merely to feed the animals, whereas our problem is to feed people. One of our post-war problems is to design and provide mechanical equipment and power that this small acre farmer can afford. This is one post-war job that must not be neglected, for there are over 1,000,000 farms of between three and 20 acres. Perhaps only one tractor can be sold to four or five farmers to use on a cooperative basis, but this is 200,000 units.

There are also about 1,250,000 farms of between 20 and 50 acres, and not many of this size have become tractor powered because of small income. These two markets alone represent over one-third of the total farms in the United States.

It also is of interest to note that in the great corn and grain growing area there is found but little more than one tractor for two farms, and there is still more than one team of horses for *every* farm in this area. This certainly supports our theory that, good as tractors are, their convenience still lacks that afforded by animal power. Post-war design of tractor and implements must find some way to work out a means to displace fully the inefficient horse.

In certain areas of the South where farms and income are small, tractor power has not been generally accepted. However, during the latest decade or so, industry is moving into and help is moving out of the area, so a great transition is developing. Diversified farming was increasing rapidly even before Pearl Harbor, and these changes will create a demand for machinery on these farms. Note that only 9% are presently supplied with tractors and that about 40% of animal power is located on these farms. There are over half of all farms in this area, and over one-third of all farm area. More than 2,100,000 farms in this section are small acreage – below 100 acres.

Since this is a virgin tractor market, our best experience toward providing a versatile, trouble-free, and *moderate-priced* tractor is demanded.

Here is field enough for more tractors than have been sold in all-time sales of tractors in the United States. A post-war duty confronts the designers and builders of tractors and of all other farm machinery.

When these post-war models are ready we should find a ready market for sale of 2,000,000 or more tractors, and the United States' total may exceed 5,000,000 tractors. This does not include needs for industrial fields nor export sales possibilities.

Eighteen years ago we predicted an ultimate tractor sales possibility for the industry of 300 tractors per day, but this opinion was somewhat forced at that time because we were trying to justify the creation of our Farmall works. The industry sales in the last 10 years several times have been over 600 units for 300 days per year.

We are now talking of a market in excess of 2000 tractors per day per year of 300 days, if we include industrial requirements.

There will be, in addition, a continued demand for limited sales of garden-type tractors to those urban workers who are living on three-acre homes around our manufacturing centers.

War-Type Materials

The development in the field of metallurgy, due to war need, has been great, and some of this will be useful in future tractor manufacturing. Some of the greatly increased volume production of materials which we have used but little heretofore may afford opportunity to increase the use of these materials and thereby improve our products without undue cost penalty. There is no question that tractor designers are keeping alert to these possibilities and will, as we have in the past, make full use of these things which afford improved performance consistent with their cost.

Cast iron, because of its cost and because tractors need weight to be effective, will always make up the greatest metal tonnage for the farm machinery.

Steel, in all its varieties, will continue to be our predominant tonnage material. It has been generously used in farm machinery and it will not likely find use in many new places nor in greater volume than farm machinery has long used.

Tractors are necessarily heavy because they work pulling not carrying loads, and the factors that enable them to pull loads along or through the ground are determined by the weight imposed on the ground through the tractor wheels.

Our tractors are currently designed with as little weight in them as load stresses on gears, shafting, bearings, and so on, will permit, and, in general, the cheapest metals that can meet the requirements have been used.

We, therefore, cannot expect to displace steel or cast iron with a more expensive and lighter metal, thereby further reducing tractor weight already too light for maximum performance. Practically all of our tractors use and need added weights when doing heavy farm work.

The war demand has increased the volume of such metals as aluminum, magnesium, molybdenum, copper, nickel, and tin, and the producers of these materials will naturally endeavor to continue their activities in peace uses.

We shall use aluminum to a greater degree in our engines if the post-war price will permit. Better engines can be made through the use of aluminum at strategic points where cast iron has largely been used to date. We have known this fact but could not justify the use of aluminum at its pre-war cost. We will, however, need to make up the weight reduction aluminum will give us by using more heavy materials at other points of the tractor.

This same consideration will determine any use we can make of magnesium.

As to molybdenum – this can be used only as an alloying agent where we have previously used such metals as nickel, chromium, and tungsten, and must compete with these alloys. The total weight of all these alloys we have found desirable, regardless of their cost, is only a few pounds per ton of steel, so much increased use of alloys is not likely.

Copper and its alloys make the cheapest satisfactory bearing metal where surfaces move on each other under load, and we have used it rather liberally for this purpose. Our tendency will be to continue its use for these purposes, but the trend will be to reduce the thickness of the metal used in bushings, bearings, and wear plates. Copper and its alloys have also found their place as heat-transfer materials and have been used almost exclusively for radiator materials. This will continue to be the case, but some slight change in quantity may occur as engines are modified through the use of aluminum, and so on, in their construction. Also, should pressure cooling systems prove popular, we will save materially on the quantity used for radiators.

Tin has been used as sparingly as possible because of its cost, but it is a peculiar material and limited to certain fields of usefulness. Even if it should become a drug on the market we will not likely use much more of it in tractors. It will be difficult to displace where we have used it because of its peculiar bonding and protective qualities.

Plastics of many valuable types have developed, but most, if not all of these, have limits in their usability in our particular fields. They fail under temperature and shock. Rigidity changes rapidly at their critical temperature, also strength changes from the same cause. They find use in accessory and trim features, and will be selectively used as their worth is proved. They must be oil resistant, and, for many uses, must not absorb moisture.

Rubber tires made the greatest improvement in tractor popularity since the development of the all-purpose type. A terrible deprivation when taken away from us, but should again become available or perhaps be displaced by a better product, synthetically made, at the end of the war.

We should, ultimately, create a definitely more uniform quality and a better product from synthetic development than the best natural rubber we have known.

Comfort, safety, and styling are all in need of improvement in new designs.

Comfort should be given the fullest consideration because the use of the tractor is based on long hours of sustained operations and no chance to stop for stretching or a snack ever comes into farm worker's life. Better seats, labor-saving devices used in the manipulation of implements, weather protection – these are the main requisites for comfort.

In the field of safety, attention has been directed to the hazards of hand-crank starting of engines. Personal examination of accidents, reported industry-wide, shows a small percentage of all accidents are chargeable to hand-cranking. Electric starting will be practically 100% equipment as soon as war conditions will permit, and this hazard should disappear.

We have for years been studying the dangers that accompany working around machinery, and have provided enclosures, shields, and the like, to cover these hazards.

Shields and safety devices removed and left off, carelessly made drawbar hitches, refusal to operate at slow speeds where fast driving is hazardous, and other similarly careless work cause almost all the accidents reported. Only continued education can correct these conditions.

High road speeds and high center of gravity of all-purpose tractors are not conducive to safe operation. We must reduce top speeds on this type of tractor and make the farmer depend on his motor truck or automobile for speeds over those suited for field operation.

Styling must wait until new developments may be brought into production, because the immediate post-war need is for machines delivered to the farmer at the very earliest possible date.

■ Post-War Fuel

Before taking up the post-war fuel situation we wish to point out that engine development for the farm tractor has pretty much been influenced, to date, by the residual fuels that have developed as the result of distilling crude to meet other more active markets – and we must recognize our engines have entirely different requirements for torque, acceleration, and long hard pulls than engines for automobiles or for trucks. Our dirty-air problems are also very much more severe.

Gasoline was first used because the great market was for kerosene, and gasoline supply became a storage problem. The great volume of automobile production increased the gasoline demand and caused increases in the price of gasoline. In order to improve roads for the automobile, a tax was generally placed on gasoline. This situation resulted in efforts to provide engine equipment for farm tractors that could use tax-free fuels.

As the electric light reduced the need for kerosene, the search for other places to use it resulted in vaporizing devices adaptable to tractor and other heavily loaded engines, and so, engine fuel became kerosene and distillates, though gasoline was preferred except for its cost.

Cracking processes for obtaining needed quantities of gasoline are making kerosene and lighter distillates expensive, so the diesel engine has developed an appeal among farmers because diesel fuel is still relatively cheap.

What will the fuel be that the post-war farm tractor may find attractive?

The oil industry has predicted for post-war:

1. About five times as much demand for 100-octane fuel as pre-war demand. However, they admit two to three times as much may be more nearly right. Maximum of 20% can be made unless octane rating lowers.

2. Premium-grade demand will probably double. 85-octane is probable.

3. Regular grade will reduce about in proportion to increase in demand of premium grade. Octane value may reach 76.

4. Disposition of some refiners is to cease making third-grade fuel. This later item affects the tractor operators very keenly, for there are 1,900,000 farmers with present tractors and engines, who must have this quality or less expensive fuel, and who must not be ignored nor unduly penalized.

Admitting the difference in opinion that exists between tractor engineers regarding the desirable engine fuel, we are certain that the farmer demand will continue to be for cheap fuel when times are hard, and that such fuels must always be available – especially when times are bad. Our tractor engines must, therefore, be of a design that readily adapts its fuel equipment to use of this lower quality fuel.

■ Fig. 13—Four-wheel drive built in 1918

There is no difference in opinion about the fuel we would all prefer if all fuels cost alike – gasoline of high grade suits everybody at all seasons of the year – but the cost of fuel needed for a day's work is generally a matter of serious concern to the farmer. Moreover, we have in use at present, over 1,000,000 tractors that will continue to work for many years, and which engines will not use premium fuels efficiently enough to warrant using this higher cost fuel.

Examination of Nebraska test results achieved on basically identical engine designs fitted by makers with equipment suited for 70-75-octane gasoline, or with other equipment suited for 38-45-octane distillates, shows equal or greater power per gallon of fuel is provided by the distillate-burning engines.

This supports our contention over many years that the cheapest farm power is obtained with the lower grade fuels.

We do not want to be misunderstood – our position is more horsepower-hours per gallon of fuel and at materially less cost per hour or per acre of work done; and our efforts to obtain more power from a gallon of fuel by study of better combustion-chamber shapes and more efficient vaporizing will be steadily increased.

Greater maximum power *is* obtained by using gasoline, and higher octane gasoline will still further increase the engine power – but even the highest octane gasoline cannot equal the cost per horsepower-hour that distillate provides.

Our information is that, to raise the octane rating of fuels will reduce the amount of such fuel per barrel of crude oil, and will also add to the cost per gallon of the higher octane material. Both of these results become a serious consideration to the tractor industry. If there is an increase in residual materials when making higher octane gasoline, will this residual have to find a market? Could this byproduct become an interesting fuel for the tractor?

If there is no increase in the residual products due to the production of the higher octane gasoline grades, what becomes of the rest of our barrel of crude? In our threatened shortage of crude oil we should not develop refining practices that consume increased proportions of the basic

■ Fig. 14—Six-wheel drive

material merely to provide cake when bread is what we work on. These questions add confusion to the decision on the farm tractor engine type.

Will there be a diesel-engine fuel, and at what cost, when the post-war refining is fully determined?

There is no question about the diesel's efficient use of a low-quality fuel, but the opinion that the diesel is a success on *any* grade of fuel, and *likes* vegetable oils, beeswax, or butter on Sunday has been one great cause of its limited use to date. It is true that the diesel engine can run on almost any injectible fuel, but it has its problems of maintenance increased when compelled to use unsuitable types.

Its other problems interfering with its more general use are, high first cost and starting trouble in cold weather, and there is already a rising cost of the grade fuel it really demands for proper performance. It has an appeal when times are as flush as at present, but it has not a bright future until its first cost and starting problem are solved.

■ War Produces Competition

Because of the immense field for tractor power still unfilled on our farms, when war production stops we may well expect some attempts from management of those large and well equipped war-built factories to enter this field, especially those who have been making tanks. However, such management will do well to fortify itself with plenty of cash reserves and iron nerves; for the road to the farmer's power problem is a specialized one with plenty of "dirt" to cause trouble, and it will find that in addition to the tractor, a whole line of implements is needed to assist in selling the tractor.

Tractor and other farm machine development is peculiar work. To be successful a great deal must be known about farm work itself, and this work varies with natural conditions in different sections. There are at least three types of engineering involved to produce successful farm machinery:

1. Technical engineering – which there is plenty of and capable.

2. "Imagineering" – in this work, engineers must be directed by an understanding of farm requirements. Help of this type is not too plentiful.

3. Protective engineering – this training is secured only in the schools of farm machine operation, and requires long and widely scattered work in actual farm operations.

No doubt attempts will be made to hire away from established tractor and farm machinery builders, men who can direct these new firms in their development programs. Granting they do these things, what constructions can they offer that we have not explored? In the *farm implement* field they bring nothing of value from their war work to its production.

Constructions in their efforts at tractor development that are most likely to be tried will be multiple-wheel drives and half or full tracks, as they are set up to produce these and know the records made by these machines.

These have done a satisfactory job for many of the Army requirements, but satisfactory life and performance for the Army are measured in hours and miles, whereas tractors must be good for years of service and for many acres plowed and cultivated per year. Five thousand miles is considered a hard test for the Army, but it represents only 250 hr of operation at most, and it is permissible to have some repair work done during such a test. The average tractor works 750 or more hr per year, and if it needs serious attention in this time it is not considered satisfactory. We can multiply this annual work by 10, and with upkeep of nominal amount per year for the life expectancy and performance that present tractors provide. Many tractors work well over 1000 hr per year, and many of our tractors are 10 years old and still in use.

We have predicted that the most likely attempts by tank and transport vehicle producers into the tractor field will be with chassis following war vehicle lines, but these farm machines must be powered with smaller-than-war-type engines to protect the operating economy required in farm operations, and transmissions must be wholly redesigned to provide suitable working speeds. Almost no farm equipment requires over 40 hp – most of the hours spent in farm operations require under 20 hp. These two facts mean that they must develop fully the products they will offer to the farm trade. This development will require years of work in view of their present knowledge of its requirements.

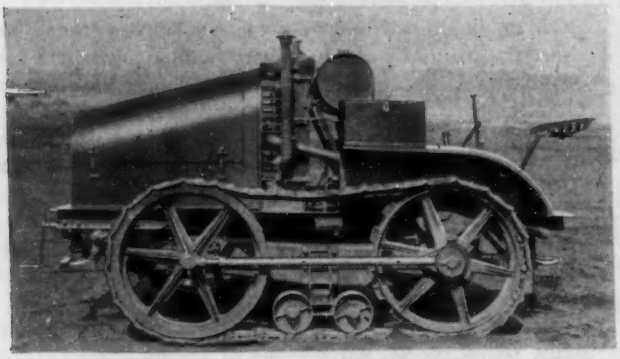
Our company has done quite a good deal of experimental work with these types of chassis.

A few pictures of tractors made in the period immediately following World War I will, no doubt, be interesting.

Fig. 13 shows a four-wheel drive, which is one of the 10 units we built in 1918. It was a well enclosed gear-drive tractor and was provided with a differential steering unit that was quite successful. One unit was in use near Aurora, Ill., for so many years that repairs needed became a problem for us to supply – yet we could not persuade this owner to part with it. Expense of manufacture and patents on the differential steering were deciding factors in the conclusion not to manufacture it.

The six-wheel drive shown in Fig. 14 was an outgrowth of the four-wheel unit, and five of these were made for the California citrus fields. The competition was crawler-type tractors. Our four-wheel drive unit just described had not done too well in crossing irrigation dykes and ditches. In this work the six-wheel drive was exceedingly successful, but its differential steering did not turn short enough for the headland requirements. The patent situation applied to these tractors as much as to the four-wheel unit.

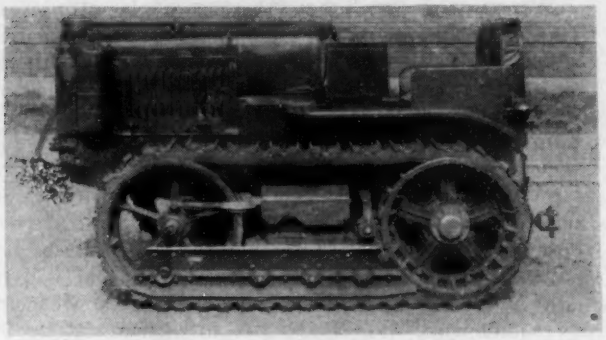
Fig. 15 shows an 8/16 track model following several years of work on this type of tractor. Expense of the track and short life of track parts, as we knew how to make



■ Fig. 15—International 8/16 track-laying tractor

them with our limited knowledge of the track-layer problem at this early date, discouraged our effort on this design. Bates steel mule was one of the outstanding half-track models, and it was a prominent unit for several years. However, all who believed in track-laying vehicles ultimately concluded that half-track measures were only "half-baked" ideas, and the full track-layer was left to fill the needs that the steel-wheel tractor was not meeting, that is, work in sandy and loose soils.

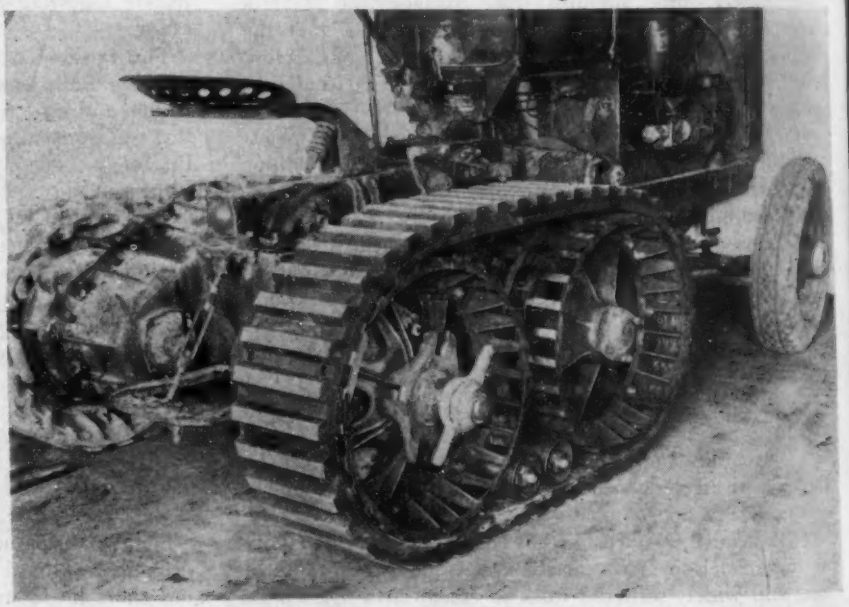
International Harvester Co., in the early thirties also



■ Fig. 16—Experimental crawler tread made of rubber with lug plates riveted thereto

spent much time and effort to develop a rubber-band type of track-laying tread, similar to the tank treads shown in Figs. 16 and 17. We later built several models with link and various types of rubber-belt tracks, but the net result was disappointment.

Admitting that great rubber track improvement has been made from that time, our experience with these very latest band tracks on our M-5 prime mover and other units in war use has shown that very short track life on hard traction will be experienced.



■ Fig. 17—Another experimental crawler tread made of rubber

We have pointed out many ways in which our best tractors of today must be further improved, and have shown that there still are ample markets for the right types of tractors.

We are repeating what we consider some of the most important requirements for such tractors.

1. Provision for quickly attaching or removing mounted farm implements.
2. Greater ease in controls for manipulating these implements.
3. Operator's comfort for long hours on the tractor.
4. Tractor sizes suited to the small acre farm and its limited income.
5. Engine combustion-chamber shape and vaporizing equipment development to be studied intensively so that we provide the maximum in power per gallon of fuel, since it is evident that premium fuels are not going to be popular with farmers.

Up to this time we have not been free from War Production Board restraint on materials and manpower needed for new development. Until opportunity is afforded to build our ideas into working models and prove their value, it is difficult to predict what we will build for our post-war model, however, the following specifications should provide a type attractive to sales department and purchaser.

■ The Engine

1. Superelastic powered, fueled by oxygen from the surrounding air and burned through contact with an indestructible catalyst. This eliminates carburetor and fuel tank, and the oxygen separating device eliminates the need for an air cleaner.
2. Engine starting accomplished by radio-activation, thus solving cranking problems, and by the use of radio-activated headlights we have solved the problems of storage battery and electrical equipment.

■ The Chassis

1. Torque converter transmission with control of vanes and stator blades automatic so that 100% efficiency is

concluded on page 518

An Investigation of CHAFING on AIRCRAFT-ENGINE PARTS

IN RECENT years, chafing, galling, or fretting corrosion of surfaces in contact has become a serious problem in aircraft engines concurrent with increased demands for more power and minimum weight.

Deflection of parts designed for minimum weight increases the tendency of clamped or pressed surfaces to move slightly relative to each other, especially under increasing loads. This motion causes chafing, which introduces stress concentration in the affected areas.

The authors describe a group of laboratory tests evaluating a limited number of methods for eliminating chafing, primarily for application to aircraft-engine development.

Results indicate that chafing can be prevented by:

1. Elimination of relative motion.

2. Provision of an adequate lubricant or coating to reduce friction.

THE AUTHORS: H. C. GRAY has been with Wright Aeronautical Corp. since 1936, the year he graduated from Worcester Polytechnic Institute, starting as experimental test observer and reaching his present post of project engineer in charge of applied mechanics and stress measurement group in the engineering department. R. W. JENNY received his industrial experience at Wright Aeronautical Corp., and is now employed as design engineer for Curtiss-Wright Development Division. He is a graduate of Rensselaer Polytechnic Institute.

by H. C. Gray* and R. W. Jenny**

CHAFING, galling, or fretting corrosion, as the phenomenon to be discussed is variously called, is a problem which has become more important in aircraft-engine parts as the power output has increased and caused a corresponding increase in vibration problems. The phenomenon is characterized by apparent picking out and flow of metal on surfaces which are supposedly rigidly clamped one to the other. Picking out and flow of metal increases the stress concentration in the area where it is present, which, of course, leads to reduced strength and, in many cases, failure. Since the phenomenon had an injurious effect on aircraft-engine parts in service operation, an investigation was initiated to determine conditions which would cause chafing and to discover, if possible, various means of eliminating it or neutralizing its effects. Figs. 1 to 5 are typical examples of chafed parts as encountered in radial aircraft engines.

It has been discovered that, in order to have chafing between two mating surfaces, both pressure and motion between the surfaces are necessary. The motion necessary to cause chafing may vary from an infinitesimal amount to motion of such magnitude that incipient oil film wedging action may be maintained. The load at which chafing occurs may vary widely, but chafing may occur as long as motion is present and oil film wedging action is not initiated. It has been discovered that chafing can also be caused by a variation in clamping loads, and we explain this by pointing out that, under extreme pressures, the clamped surfaces will expand and contract in a direction at right angles to the direction of load application with changes in pressure. This phenomenon is proof of our statement that the motion causing chafing may be infinitesimal.

It was the purpose of this investigation to evaluate various chafing prevention methods, as well as to gain some experience in the mechanics of chafing itself.

After some discussion, it was decided to attempt to carry on the investigation on a laboratory machine in order to evaluate chafing prevention methods under controlled conditions and without costly engine testing. It was our purpose in this, not to eliminate engine testing, but to carry on a rather extensive laboratory test program under accelerated chafing conditions. In this way, we proposed to evaluate chafing prevention methods rapidly and to elim-

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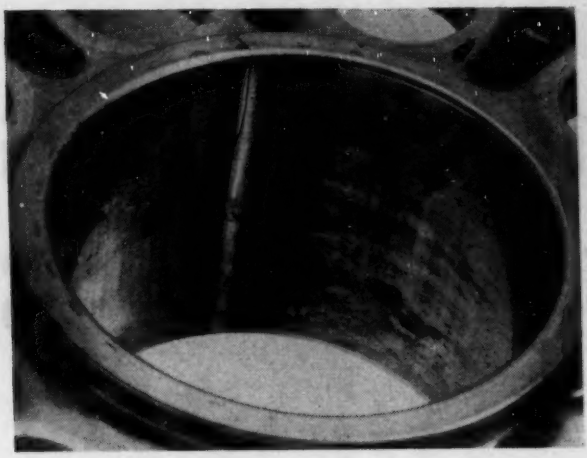
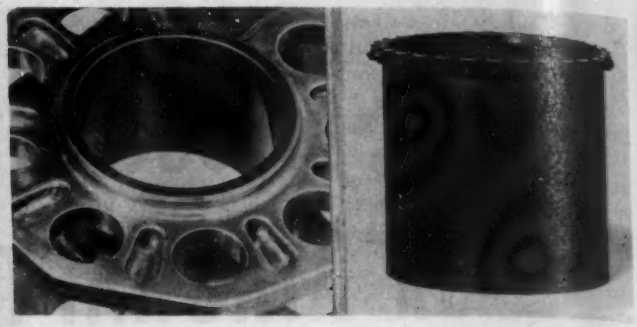
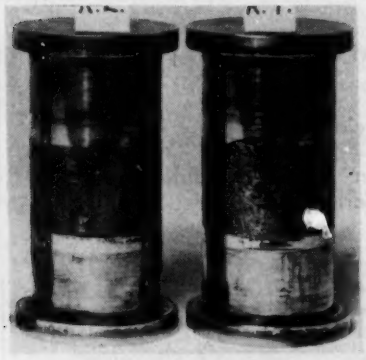


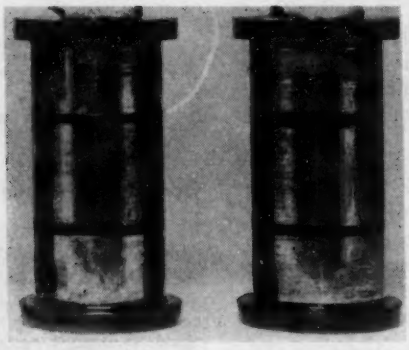
■ Fig. 1 (left) – Master-rod bore with chafing as originally encountered – bearing tight fit in bore



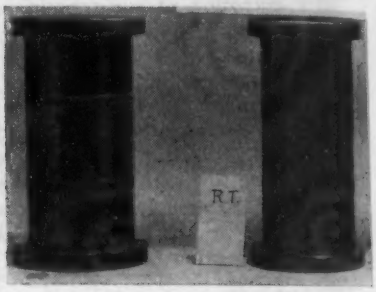
■ Fig. 2 – Master-rod bore (above left) and master-rod bearing (above right) after chafing prevention methods were applied – grit blast and lead plate on bearing outside diameter – bearing loose fit in bore



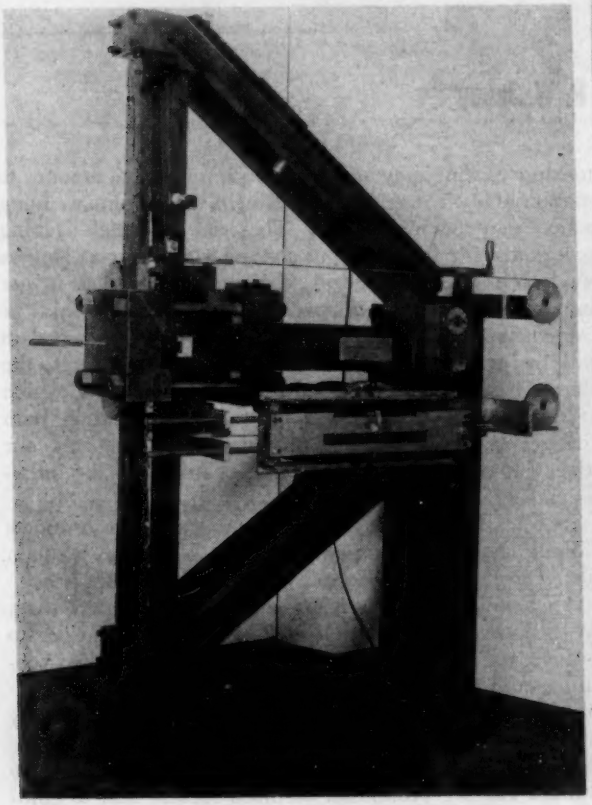
■ Fig. 3 – Standard untreated damper pins with chafing as originally encountered



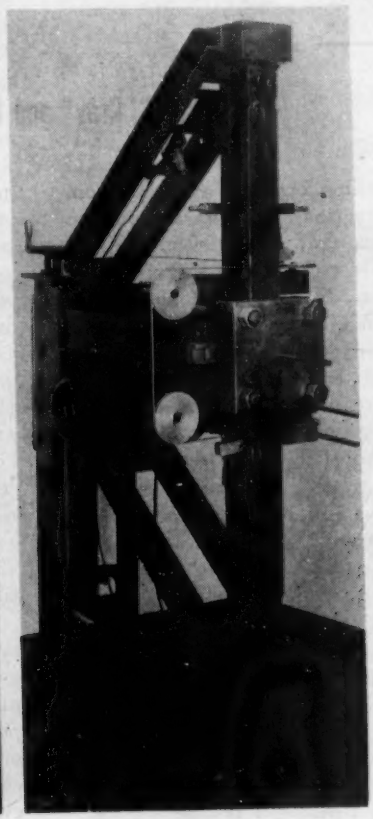
■ Fig. 4 – Damper pins with unsatisfactory antichafe coating – black iron oxide coating



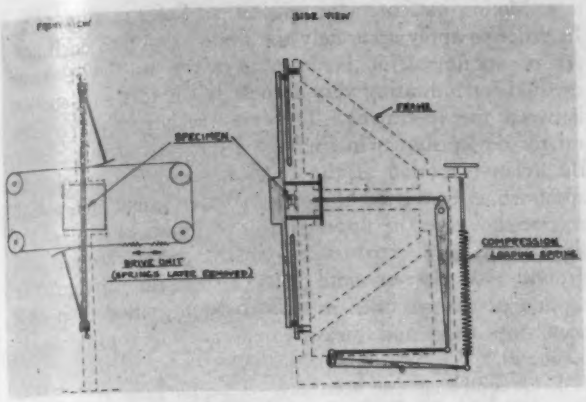
■ Fig. 5 – Damper pins with successful antichafe methods applied – grit blast and lead plate



■ Fig. 6 (left) – Front view of machine for evaluation of antichafe materials



■ Fig. 7 (right) – Rear view of chafe test machine



■ Fig. 8 – Schematic diagram of chafe test machine

inate unsatisfactory methods in the laboratory. Next, only those chafing preventives which gave satisfactory results in the laboratory were to be subjected to engine proof tests, which could be concentrated on those chafing prevention methods which indicated promising results, and a great deal of costly and sometimes inconclusive engine testing could be avoided.

Our first, and by far most important, immediate problem in starting this investigation was the development of a machine for the evaluation of antichafe materials. This development was carried on by Calibron Products, Inc., in cooperation with Wright Aeronautical Corp. engineers. Figs. 6 and 7 show the general appearance of this machine.

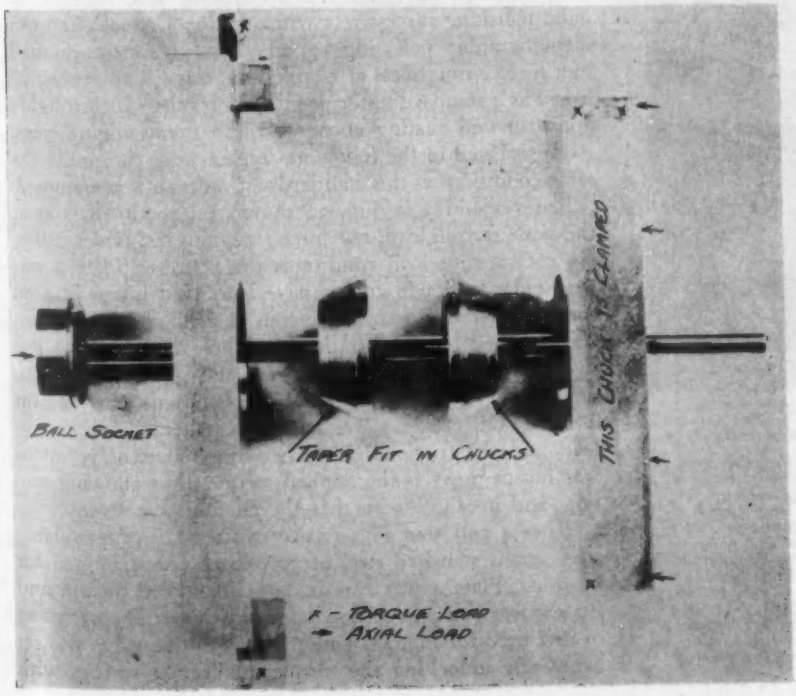
In the discussion of this machine, we shall divide the mechanism and indicating instruments into separate elements, and discuss each in turn.

1. Pressure application linkage (Refer to Fig. 8.) – The pressure application linkage consists primarily of a compression loading spring which is calibrated for extension versus load, and a lever system which transmits and magnifies this load and applies it to the chafing specimens. The mechanical advantage of the lever system is 198 to 1. Since the specimen must also be operated upon by a slight torsional motion, it was necessary that the upper horizontal member of this lever system have some torsional flexibility. This torsional flexibility is indicated in the diagram by the fact that the upper horizontal member is comparatively slender. Note that a hand wheel is provided for extension of the loading spring and application of compressive loads.

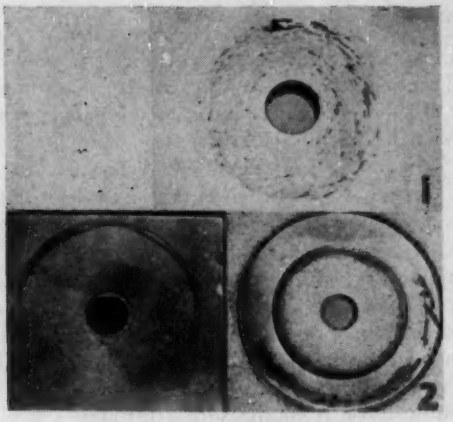
2. Torque application linkage (Refer to Fig. 8.) – The torque application linkage is shown in the front view of Fig. 8. It consists of a drive unit by which a simple harmonic motion of variable amplitude can be applied to the springs indicated in the diagram. This spring deflection can apply a torque load to the specimen. The torque load is transmitted from the spring through a steel cable to two lever arms, one at the upper end of the machine and one at the lower, which are mounted on fixed pivots. The motion or torque transmitted to the specimen is carried by a long beam member with a yoke at each end acted upon by a pin located on the previously mentioned levers. It should be noted that the springs indicated in the cable system were removed for the greater portion of the tests conducted, which resulted in actual motion being applied to these specimens rather than simply torque.

3. Specimen design (Refer to Fig. 9.) – The specimen as designed for this chafing test machine is shown in Fig. 9. The specimens are in the shape of truncated cones with the bases arranged to make contact. The contact surfaces are flat rings with an outside diameter of 1.52 in. and an inside diameter of 1.00 in. This results in a total contact area of 1 sq. in. These specimens seat on their conical surfaces in the two large blocks shown in Fig. 9. The block at the right is fixed in the frame of the machine, while the block at the left is clamped to the arm which is acted upon by the torsional load linkage.

4. Spherical seating arrangements (Refer to Fig. 9.) – In order to ensure even contact pressure, a spherical seating arrangement was included in the specimen clamping device. This spherical seat is shown to the left in Fig. 9. The center of the spherical surfaces is located in the plane of the contact of the two specimens. In this way, the specimens, when clamped, are allowed to adjust themselves so that the contact surface is evenly loaded about its circumference and load cannot be concentrated on one side alone. In some of our early tests, this spherical seating arrangement was not used, and the need for it was immediately apparent. (Note Plate 2 in Fig. 10.)

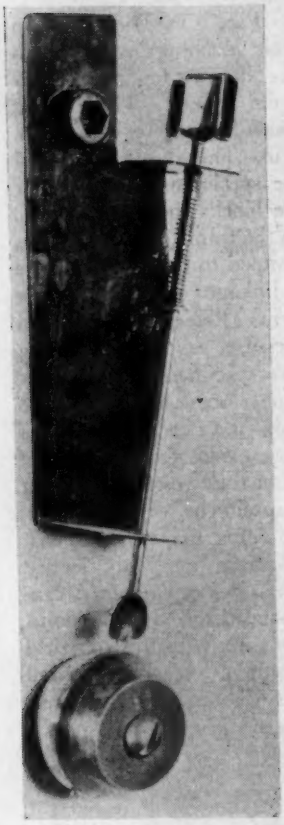


■ Fig. 9 – Specimen clamping apparatus – chafe test machine



■ **Fig. 10 – Gaskets**

1. Nylon – standard test. On this specimen, absolutely no marks were present after test with nylon gasket. Marks shown were caused by later test with steel gasket. 2. Beryllium copper – standard test. Chafing is present. Notice the uneven loading – spherical seating clamp not used.



■ **Fig. 11 (left) – Motion measurement apparatus – chafe test machine**

■ **Fig. 12 (above right) – Miscellaneous pilot tests**

1. Steel on steel; load: 3000 lb; motion: 0.004 in. (forced); time: 15 min. This is considered average chafing and is our standard reference test. 2. Steel on steel; load: 23,000 lb; motion: not forced – eccentric at 0.004 in.; time: 15 min. Friction force exceeded torque applied, hence no motion or chafing. 3. Steel on steel; load: 3000 lb; motion: not forced – eccentric at 0.004 in.; time: 15 min. This test was identical with No. 2, except for low load. Torque caused motion and chafing resulted. 4. Copper-plated steel on steel; load: 3000 lb; motion: 0.004 in. (forced); time: 15 min. Forced motion caused chafing comparable to standard test (No. 1). 5. Copper-plated steel on steel; load: 3000 lb; motion not forced – eccentric at 0.004 in.; time: 15 min. Test identical to No. 3. Copper caused increase in friction, stopped motion, and prevented chafing.

5. Motion measurement apparatus (Refer to Fig. 11.) – In order to apply accurately the same motion to each new set of specimens, it became necessary to devise some method for indicating very accurately the torsional motion between the specimens. This was accomplished by the mirror device shown in Fig. 11. By means of this device, the relative motion at the circumference of the chafing specimen is converted to rotational movement of a small mirror shown at the upper end of the shaft in Fig. 11. A light beam is directed at this mirror and reflected to a ground glass plate at some distance from the mirror. The motion of the specimen is sufficiently amplified to permit accurate setting and measurement within approximately 0.0002 in.

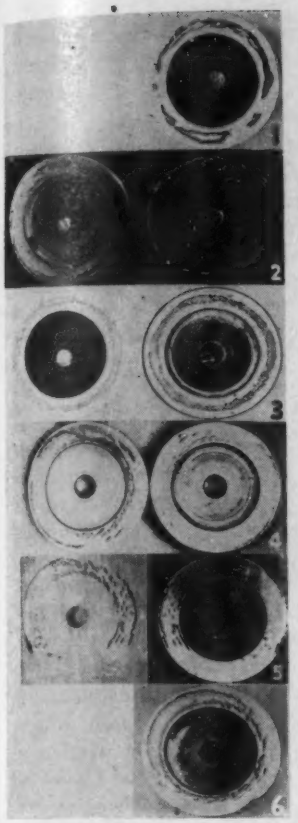
6. General machine accuracy – The machine as designed has proved to be very accurate. Tests on the compressive loading mechanism have indicated that its accuracy is within 2% of the total load indicated. The accuracy of the torsional load application linkage is somewhat less than this, but with the torsional motion measurement device, the torsional motion could be measured very accurately to within limits of a few tenths of a thousandth of an inch. We have found that conditions of surface roughness and trueness of surface can have more effect on difference in successive tests than the accuracy of machine linkage.

Results of tests made to date are shown in Figs. 10, 12-21. The results shown are intended merely to compare the relative effectiveness of a variety of materials and treatments as chafing preventives. Many of the materials tested are not intended for use in this type of application, yet a few indicated good results. The list is naturally far from complete, but sufficient testing was done to make recommendations concerning several satisfactory methods of chafing prevention.

Fig. 12, Plate 1, is a picture of two standard steel specimens and shows average chafing conditions. These specimens are of AMS-6250 nickel-chromium steel, Rockwell C-30 hardness, and were run under 3000-lb compressive load with 0.004-in. motion for 15 min. This standard specimen and all plain steel specimens were finished by hand polishing successively with Nos. 240C, 400C, and 000 metallographic polishing paper. This treatment resulted in a surface roughness of 10 micro-in. rms. This was established as a standard test, since it gave results which roughly approximated chafing encountered in many engine parts. All tests listed in the following figures were run under the same conditions as this standard test unless otherwise noted.

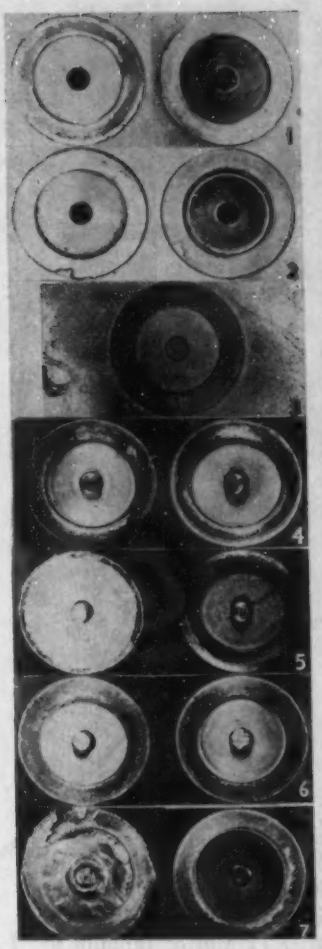
Plates 2 and 3 in Fig. 12 show steel specimens run at the same torque load, not forced motion, but under different compressive load conditions for 15 min. Plate 2 was run under 23,000-lb compressive load, and it can be seen that the chafing in this specimen is negligible. This can be explained by the fact that the applied torque was less than the frictional torque between the specimens. Thus, no relative motion took place and hence no chafing was encountered. Plate 3 was run under 3000-lb compressive load and the frictional force between the two specimens was not as great as the applied torque, thus allowing motion, and it can be seen that the chafing was severe.

Plates 4 and 5 in Fig. 12 are two tests of copper-plated steel versus standard steel at 3000-lb compressive load for 15 min. Plate 4 was run at 0.004 in. forced motion and Plate 5 was run at a torque value which would correspond to that used for Plates 2 and 3. This torque value was set by merely unlocking the torque application springs with



■ **Fig. 13 – Electroplated metals on steel**

1. 13% copper – 87% tin on steel – standard test conditions: 3000 lb; 0.004-in. motion; 15 min. Copper picks up on steel. 2. Co-deposited copper-tin alloy – standard test. 3. Porous chromo plate – standard test. Addition of collodial graphite had no effect on chafing. 4. Standard chrome plate – standard test. 5. Nickel plate – standard test. 6. Copper plate – standard test.



■ **Fig. 14 – Electroplated metals on grit-blasted steel**

1. Lead plate and grit blast – standard test.

An excellent antichafe method. Dark areas in lead-plated specimen did not make heavy contact – rest is highly polished. 2. Lead plate and grit blast; tested at 23,000 lb; 0.004-in. motion; 15 min.

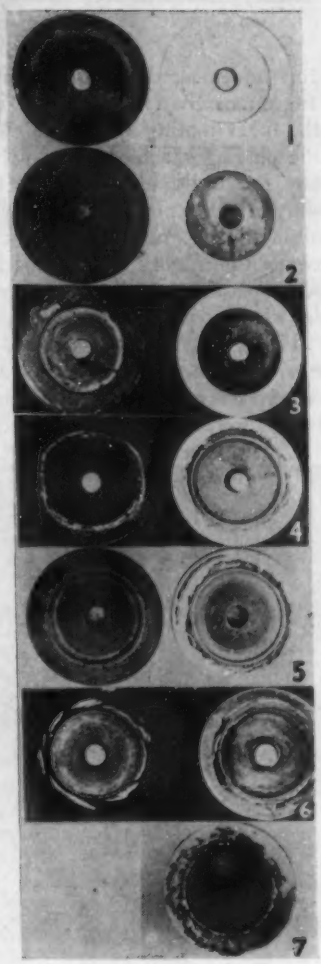
Even at extremely high load of 23,000 lb with forced motion, no chafing is evident. 3. Grit-blasted steel – standard test. Specimen shown has slight polish – some have chafed under higher loads. 4. Copper plate and grit blast – standard test. This test and the following ones show little beneficial effect on chafing. 5. Silver plate and grit blast – standard test. 6. Tin plate and grit blast – standard test. 7. Aluminum foil and grit blast – standard test.

the eccentric set for 0.004-in. motion of specimen. Note that the chafing under the forced motion was comparatively severe, while the chafing under the applied torque was negligible. This indicates that copper plate tends to increase the frictional force between the chafing specimens and may be expected to cure chafing in applications where a moderate increase in frictional force can prevent motion.

Fig. 13 shows a series of electroplated materials on steel which were run in the chafing machine against standard steel. From top to bottom, they are listed in decreasing order of effectiveness. It is evident that, in cases where motion does exist, as was the case in all of these tests, electroplated materials are generally ineffective in preventing chafing.

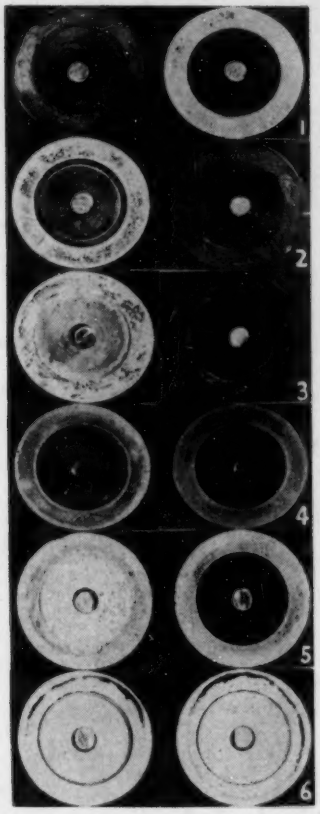
Fig. 14 lists a series of tests of electroplated materials on grit-blasted steel run against standard steel. From top to bottom, they are listed in order of decreasing effectiveness as chafing preventives. Plates 1 and 2 are lead-plated grit-blasted steel run at 0.004-in. motion for 15 min. Plate 1 was run at 3000-lb compressive load and Plate 2 at 23,000-lb compressive load. Since very little, if any, chafing is present in either one of these two plates, it is indicated that this process is extremely effective in preventing chafing, even where sizable motion and heavy pressure exist. Plate 3 shows one specimen of plain grit-blasted steel. In this plate, it can be seen that the grit-blasted steel was polished lightly but very little chafing occurred. However, in other tests on similar steels, some chafing has been found on plain grit-blasted steel, though it ordinarily chafes considerably less than standard steel.

The remaining plates in Fig. 14 show other various plated materials on grit-blasted steel and, in general, they are much less effective in preventing chafing than the lead plate and grit blasting.



■ **Fig. 15 – Special coatings on steel**

1. Zinc and iron phosphate – bonderizing – standard test. This specimen indicates good antichafe properties for a thick bonderized coating. (See Fig. 20.) 2. Baylee process – standard test. 3. De Khotinsky cement – resinous – standard test. Good antichafe – cement will melt at about 250 F. 4. Low-temperature black iron oxide – Jetal – standard test. 5. High-temperature black iron oxide – Ferrox – standard test. 6. Low-temperature black iron oxide – Dulite – standard test. 7. Cast hard metallic coating – Colmonoy – standard test.



■ **Fig. 16 – Nonferrous metal on steel**

1. Sulfide-coated bronze on steel – standard test. Relatively minor chafing. 2. Cast bronze on Parco-lubrized steel – standard test. Minor chafing – for a comparison, note that Parco-lubrizing is similar to bonderizing (See Fig. 15.) 3. Magnesium on Parco-lubrized steel – standard test. Chafing only slightly worse than No. 2. 4. Magnesium on steel with silica oil (200-sec viscosity) – standard test. Silica oil of this type has beneficial effect as antichafe. Compare with No. 6. 5. Magnesium on steel with Lubriplate – standard test. Effectiveness of Lubriplate less than silica oil. 6. Magnesium on steel – standard test. Magnesium more severely damaged than steel.

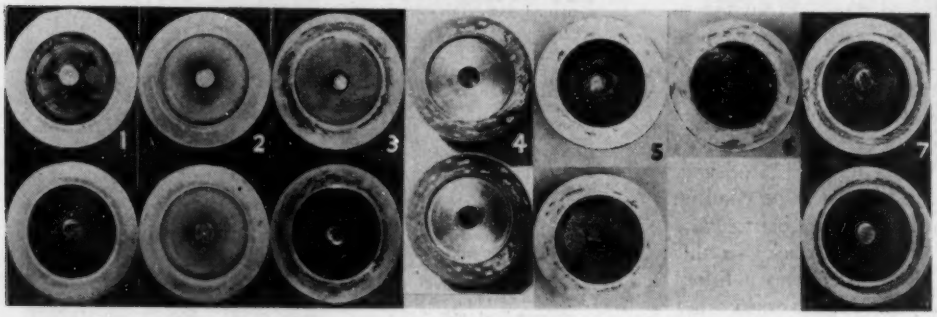
Fig. 10, Plate 1 shows a steel specimen which was first run under standard test conditions with a nylon washer approximately 1/16 in. thick between this specimen and the other mating specimen. After 15 min of running in the chafing machine, absolutely no chafing or marks of any sort could be noticed on the specimen. The marks shown in the plate are those which were made during an additional 15-min run with a standard steel washer between the mating chafing specimens. This indicates that some plastics may be very effective when used as gaskets for chafing prevention.

Plate 2 in Fig. 10 shows a pair of chafing specimens run with a beryllium-copper washer between them. Moderately severe chafing is present, but the interesting thing to be shown by this plate is that uneven loading can cause severe chafing where the load is high and where the load is low, chafing will not necessarily exist. The specimens shown in this plate are of an early type which did not include the spherical seating arrangement.

Fig. 15 shows a series of special coatings of various types on steel, run at the standard conditions of 3000-lb compressive load, 0.004-in motion, for 15 min. It can be seen readily that the first three specimens were very effective in chafing prevention, while the latter four wre relatively ineffective. Plate 1 shows a zinc and iron phosphate coating, Plate 2 a coating by a similar process, and Plate 3 is a specimen which was coated with a resinous cement which is heat softening and can be used only at temperatures up to 250 F.

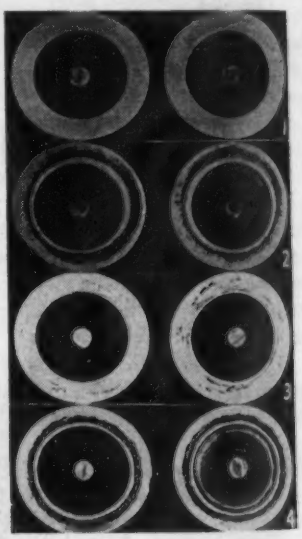
Fig. 16 shows the chafing characteristics of several nonferrous metals on steel under standard chafing test conditions. They are listed, as usual, from top to bottom in order of decreasing effectiveness as chafing preventives. In most cases, it will be noted that uncoated, soft materials chafe somewhat more than the steel against which they were run. Note in Plate 6, which is a specimen of cast magnesium on standard steel, that the chafing of magnesium on steel is somewhat less severe than that of steel on steel. This, and other similar tests, led us to the belief that unlike materials, in general, chafe less than like materials.

Fig. 17 shows specimens which were run with various lubricants. Of very special interest is Plate 1, which is a specimen of ordinary steel run with a material known as silica oil. This material apparently has a peculiar property of increasing the friction between specimens. In this test, several attempts were made to evaluate chafing on the chafing surfaces with this silica oil smeared on the contact surfaces before starting the test. In each case, the specimens did not chafe on the usual chafing surface but were forced to move and thus chafe in the conical clamping surfaces. This indicates that the friction on the flat surfaces which nor-



■ **Fig. 17 – Lubricants on steel**

1. Silica oil – 5000-sec viscosity – standard test. This type of silica oil has the property of increasing friction. In this test, motion and chafing occurred on conical clamp surface. 2. Silica oil – 200-sec viscosity – standard test. Different type of silica oil. Apparently maintains film and greatly reduces chafing. 3. Petrolatum and graphite – standard test. Only fair as antichafe. 4. Engine oil – 120-sec viscosity – standard test. Chafing was severe – deep gouged marks. 5. Engine oil – compounded – standard test. Results similar to No. 4. 6. Lubriplate – standard test. Results similar to No. 4. 7. Silica oil – 185-sec viscosity – standard test. This oil is of same type as that in No. 1, but lighter grade. Increased friction caused specimens to weld.



■ **Fig. 18 – Miscellaneous steels**

1. Graph-Mo graphitic steel with silica oil – 200-sec viscosity – standard test. Same oil as in Fig. 17, No. 2 and it gave similar results. 2. Graph-Mo graphitic steel – standard test. Results only slightly better than plain steel. 3. Nitrided steel on chrome-plated steel (chrome plate held at 1000 F for 1 hr – standard test. Slightly better than plain steel. 4. Nitrided steel on chrome-plated steel – standard test. Chafing was worse than plain steel, although chrome plate was harder than in No. 3.

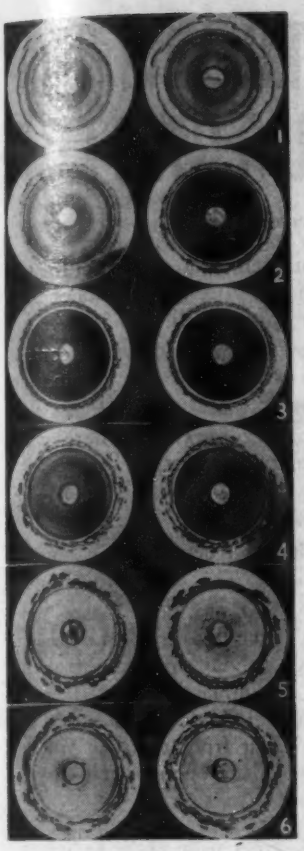


Fig. 19 – Effect of steel hardness

1. Rockwell C-60 steel – standard test. Chafing was severe – hardness alone is not a chafing cure. Note the increase in chafing as hardness decreases. 2. Rockwell C-60 to C-40 – standard test. 3. Rockwell C-35 to Rockwell C-40 – standard test. 4. Rockwell C-30 to Rockwell C-35 – standard test. 5. Rockwell C-25 to Rockwell C-30 – standard test. 6. Rockwell C-20 to Rockwell C-25 – standard test.

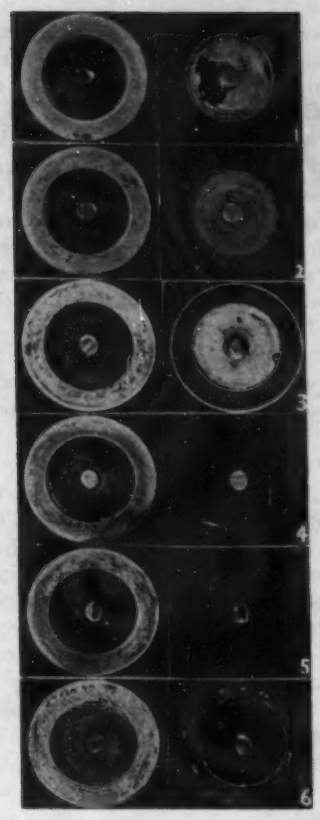


Fig. 20 – Bonderizing – effect of time in bath

1. 30 min. (stirred) – standard test. Good chafing preventive. Note increased chafing with decreased time (also decreased thickness). 2. 30 min – standard test. 3. 20 min – standard test. 4. 15 min – standard test. 5. 10 min – standard test. 6. 5 min – standard test.

mally chafe must have been several hundred per cent greater than normal for steel on steel. This was the only case where this phenomenon was noted.

The remaining plates in Fig. 17 show standard steels with lubricants as noted in the figure. In general, it should be noted that chafing with ordinary petroleum lubricants has deeper gouges and picking up of metal than those encountered on standard steel specimens. It should also be noted on Plate 7, which used a silica oil similar to that in Plate 1 but of lighter viscosity, that the friction had been increased actually to cause a welding, in appreciable measure, of material of one specimen to the other. The nicks in the upper portion of the righthand specimen were caused by hammer blows which were necessary to separate the specimens which had become welded together. It is believed that this welding action was caused by higher frictional heat and a welding of the steel. Note that other than this high friction oil phenomenon, lubricants are, in general, not effective in preventing chafing, with the exception of the No. 200 special silica oil noted in Plate 2, which did actually reduce chafing. This must be due to some peculiar characteristic of this special oil in maintaining a film under extreme pressure. This No. 200 silica oil is of a different type than those mentioned above.

Fig. 18 shows specimens of a few special steels under varying conditions. Plate 1 is a graphitic steel used with No. 200 silica oil as noted in Fig. 17, Plate 2, and it is noted that chafing is relatively less than standard steel. Plate 2 is Graph-Mo steel on standard steel without any lubricants and, in this case, the chafing is nearly as bad as plain steel on steel. Plate 3 shows nitrided steel versus chrome-plated steel where the chrome plate has been heated to 1000 F for 1 hr. In this instance, the chafing is very slightly less than for standard steel on standard steel. Plate 4 shows nitrided steel on ordinary chrome-plated steel and, in this case, the chafing is extremely severe. In general, none of these materials are satisfactory as chafing preventives.

Fig. 19 shows the effect of varying hardnesses in standard steels. Starting with Rockwell C-60 steel at the top and decreasing in hardness to Rockwell C-20 steel at the

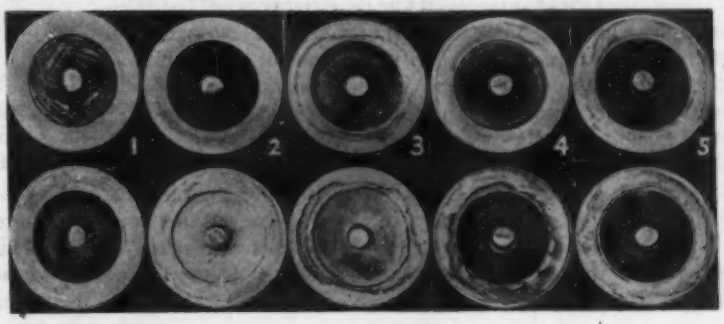


Fig. 21 – Effect of thickness of silver plate against soft iron plate

1. 0.005-in. silver – 0.001-in. iron – standard test. Chafing is slight – marks are mostly discoloration. Note increase in chafing as plate thickness decreases. 2. 0.003-in. silver – 0.001-in. iron – standard test. 3. 0.003-in. silver – 0.005-in. iron – standard test. 4. 0.001-in. silver – 0.0005-in. iron – standard test. 5. 0.0005-in. silver – 0.0005-in. iron – standard test.

bottom, it is seen that chafing increases with decreasing hardness. However, increasing hardness alone is not a good method for preventing chafing, since the chafing indicated on the Rockwell C-60 specimen is severe.

Fig. 20 shows a series of specimens which are intended to evaluate the effect on chafing of thickness of special coatings. The coating used for this test was zinc and iron phosphate. Plate 1 in Fig. 20 shows a phosphate coating as applied with 30 min in a bath which was agitated. Plate 6 in Fig. 20 shows specimens which were phosphate coated only 5 min in a still bath. It is noted that, as the length of time in the bath decreased and the thickness of the coating decreased accordingly, the severity of chafing increased. This coating, when moderately thick, is a good chafing preventive, but is effective only as long as it remains thick (about 0.001 in.). Wear, which causes coatings to become thin, will decrease antichafe effectiveness.

Fig. 21 is intended to show the effect of thickness of plated silver when run against a plated iron surface. The plates are listed, as usual, in the order of decreasing effectiveness. The chafing decreased directly with increasing thickness or, to state it otherwise, as the plating was decreased in thickness, the chafing became increasingly severe. In general, silver appears to be a satisfactory method for preventing chafing only under light loads.

From the tests mentioned above, and others, the following general conclusions can be drawn:

1. Chafing can be prevented by:

a. Increasing the compressive load to a value where all sliding motion is eliminated.

b. Providing a gasket which can absorb the motion or allow motion against steel without pickup.

c. Providing a coating which can serve as a shear member or which can provide an antifriction surface. It is believed that those coatings which are effective in preventing chafing act as an antifriction bearing with many infinitely small balls or rollers. Lead plate is probably effective because of inherent lubricating properties in lead.

d. Providing a plated or otherwise treated surface to increase friction greatly and to stop all sliding motion.

2. The severity of chafing:

a. Increases with increase of motion at a fixed load up to the point where the motion applied will cause an oil film wedge action to be maintained.

b. Increases with increase of compressive load if a given sliding motion is maintained.

3. Surfaces of unlike metals in general chafe less than like metals.

4. Steel surfaces of the same finish chafe more than surfaces of different finish roughness.

From the data gleaned to date, the correlation between laboratory and engine tests on chafing preventive measures has been good. However, all engine applications of the various cures for chafing should be thoroughly proof tested, since the characteristics of the chafing trouble in each instance may vary widely. The possibilities of eliminating chafing by eliminating load and motion factors should also be thoroughly investigated as being preferable to the other cures mentioned. If coatings prove to be necessary, only those mentioned above as having good results should be applied for engine test. Careful evaluation of the effect of any chafing preventive measure on fatigue strengths should be made before final application to engine parts. Sometimes, treatment of the surface causes reduction in fatigue strength which may be more critical than chafing.

FARM POWER and the POST-WAR TRACTOR

continued from page 510

always secured. An automatic temperature control for the fluid used in the converter so that viscosity does not change regardless of atmospheric or other conditions.

This torque converter provided with a manual control that enables full reversibility of drive, thereby eliminating reverse gearing as well as brakes.

Selective speed ranges from zero to maximum are available by throttle control of converter pump capacity.

2. Gyroscopic control of center of gravity so that overturning hazards are eliminated.

3. Electric eye steering so that obstructions in field operations are automatically bypassed. This also eliminates all hazards while on highways.

4. Radio-activated, quickly attachable, farm implement control. This has a two-way push-button switch that enables the operator to remove the implement from the tractor, return it to its shed and automatically pick out the farm machine wanted, bringing it from the shed and attaching it to the tractor without the operator leaving the tractor seat.

5. An automatic coupling device for drawbar-operated machinery. No longer do men stand between the tractor and its load when drawbar connections are needed.

6. Since all farm implements are radio-controlled, and positioned properly on the tractor, there will be no need for power take-off, tumbler shaft, and shields.

7. Deluxe upholstered seat with transparent, flexible, plastic cab stored within the seat back.

This cab inflatable to full shape at operator's wish by admitting exhaust gas into the cab structure (as is done on aircraft life boats). The use of the engine exhaust provides heat for the operator's comfort in cold weather by permitting controlled, continuous flow of the gas through the cab structure.

8. A further, most important feature in my post-war tractor is the electronic, high-frequency control for specific gravity of tractor materials which enables the operator to change the tractor weight at will. He can secure specific gravity up to ten times the weight of water for hard ground, heavy traction needs, also for very heavy industrial hauling needs, or can have a specific gravity of somewhat less than water to use when in muddy ground or when fording streams, climbing steep grades, and so on.

9. The whole tractor styled smooth as Lady Esther face powders and streamlined like a "Varga" girl.

10. Two-way radio equipment regular.

All patent protection has been dedicated to the people of the U. S. A. so that the industry may feel free to begin development work at any time on tractors having these features.

I am certain that when our young designers have worked out all details, and these tractors are universally supplied, the farmers will no longer have to sweat in order to earn their bread.

The SHAPE of TRUCKS TO COME

by MERRILL C. HORINE
Mack Mfg. Co.

OF course the trucks that first emerge from the factories after the war will have long, narrow engines standing upright between the narrow frame rails at the front, just like a horse between the shafts of a wagon. That's where the idea originated, of course, and it's been mighty convenient for the manufacturer, too.

THE post-war truck will have many improvements over earlier models, but it will also have many disappointments for those expecting too many revolutionary changes.

Frames, for instance, Mr. Horine believes, will be continued in motor trucks for a long time to come—however, they will be stronger and more efficient than those of the past.

On the score of powerplants, particularly, there will be many disappointments, for high-octane gas is too expensive to justify its use in truck engines, although there will be many refinements made in the engines.

Both brakes and cooling systems must be much improved. They will be made to operate more efficiently and with less troubles developing in service.

Clutches and transmissions will not be very different for a while, although in time, Mr. Horine believes that synchronized shifts, fluid drives, and other changes will be introduced for trucks.

THE AUTHOR: MERRILL C. HORINE (M '17), sales production manager of Mack Mfg. Corp., recently returned from Washington where he had been research consultant to the director of the Automotive Division, War Production Board, for more than a year. His career in the automotive field has been varied. He was assistant technical editor of *Motor Age* and editor of *Commercial Vehicle;* he served in the Air Service in World War I as the first man to be assigned full time motor transport work; and in 1918 he joined the engineering department of the Mack organization. Mr. Horine was 1938-1939 chairman of the SAE Metropolitan Section, and is now a member of the SAE Truck & Bus Activity Committee.

The engine has to be narrow because the front wheels have got to have room to turn. That space isn't any good for load, and besides, the driver has to see over it – if he can. It makes the controls mighty convenient and leaves a nice, long driveline with plenty of room for the regular clutch and transmission.

Then, if the designer forgot to put enough gears in the job, there's plenty of room for an auxiliary transmission or a transfer case.

The trouble is, of course, that this horse-and-buggy arrangement takes up a lot of wheelbase – particularly if the cab is placed buggy fashion, behind the engine. On most trucks about half the wheelbase is taken up with the engine and cab.

Long ago this drawback was realized and we had the cab moved up over the engine. That put more weight up front, gave the driver a better view of where he was going, and saved wheelbase.

The trouble with the high-cab construction is that it makes it difficult for the driver to get in and out – almost impossible for him to use the right side door – and generally sacrifices a lot of accessibility, complicates the controls and doesn't always provide room enough for the escape of air from the fan.

No use talking about a transverse engine across the rear, as in buses. We've still got to load and unload over the tailgate, and we can hardly afford that much concentration of weight at the rear.

There's been talk of flat engines under the body, like some of the buses, but anyone who has ever nailed lath on a ceiling knows why mechanics don't regard the accessibility of such engines too highly.

That's why a new arrangement is being considered, in which the engine consumes no extra wheelbase, its weight is nicely distributed, the cab is ideally located and easier to get into and out of than any you've ever seen, and the engine and associated parts can be standard construction. I'd like to tell you about it, but . . .

■ Styling

Did you ever notice that most of the really lasting and satisfying style forms are those which just developed naturally along truly functional lines, because those who originated them were completely style-unconscious and made them the way they did because it was the most efficient form they could conceive to do the job?

[This paper was presented at meeting of the Metropolitan Section of the SAE, New York City, April 20, 1944.]

If you think the beauty of a sailboat is the result of some brilliant stylist's æsthetic expression, then you don't know anything about a sailboat. It just has to look the way it does or it wouldn't sail as well as it does.

Nobody believes that a big Pacific-type locomotive is so truly an expression of the very spirit of power because locomotive designers gave its appearance the slightest thought. As a matter of fact the early woodburners were embellished a bit according to the Civil War concepts of art, but the less latter-day designers thought about appearance and the less conscious effort was expended on embellishments, the better-looking they became.

It was only when the streamlining craze caught the public fancy that "art" came to the roundhouse. And what desecration has been wrought on the noble steam locomotives as a result. You can see them on the New York Central, the Pennsylvania, the Rock Island, and the Southern Pacific, and they're awful.

The diesel-electric streamliners, however, have given birth to a distinctive art form – wholly new and completely functional. They look beautiful because they are appropriate to their purpose. The same thing applies to the lines and proportions of a huge airliner, a great bridge, like the Golden Gate Bridge – even the modern telephone instrument.

So I hope that post-war designers will forswear their vulgar efforts to make a motor truck look like a passenger car. The bus designer struck out along functional lines some years ago and while the gaudy circus-wagon paint effects which so many operators have demanded have somewhat marred the effect, modern buses in their intrinsic proportions embody a serene dignity of line and mass approached by no other motor vehicles unless we include some of the best examples of large tank trucks.

The chief objection to gaudy tinware on a motor truck is not its æsthetic offensiveness, however, nor the relative fragility of the tacked-on moldings, brittle diecastings, and billowing sheet metal; but it is the serious violence which these things do to accessibility either for inspection or maintenance of vital parts, their interference with proper ventilation of the hood, the brakes, the tires, and the cab, and their inhibition of frank and honest ruggedness so needed in such parts as radiators, fenders, lamps, hoods, and steps.

There's a new cab construction coming up which removes the temptation to the Christmas-tree trimming school of design by incorporating within itself . . . Sorry, I promised I wouldn't disclose that one.

■ Frames

Did someone suggest that the post-war truck would be frameless, as pre-war buses were and as post-war passenger cars will doubtless be in time? I cannot agree. Aside from frameless tank trailers, I think we are going to have to resign ourselves to the necessity for frames on motor trucks for a long time to come.

By that I do not mean that we will always hang the components on glorified stepladders. There is just no sense whatever to simple beam construction for a chassis frame.

Just 30 years ago the Austin company in England built a truck with a girder-type frame. The truck didn't last, but that was because of a lot of other innovations included in its design which did not contain as much common sense as the deep girder frame.

Every engineer knows the importance of sufficient section modulus in a frame and he also should know that you can get a lot more stiffness out of an increase in depth than you can from either width or thickness of material.

The trouble with frame design is that we lay our frames in on the assembly drawing with reference to the axles only. Then, when it comes to mounting a flat-floored body we build a lot of staging from the frame up to the floor.

Not right away, of course, but in due time, you are going to see a frame which starts at the underfloor level and then drops down – like an inverted-arch bridge. It's going to have cross-members which really do a job, too, bracing the frame crosswise and supporting the full width of the body as well.

No more sills and bolsters except just enough to hold the body together while it is being lowered onto the frame. Maybe these will be detachable assembling jigs only, removed after the body is mounted.

I'd really like to tell you more about this frame, but there's a shop out in Pennsylvania where they're still trying to find something wrong with it. I can give you this hint, though: Everybody knows that a tube is the most efficient shape for torsional stress – but how about using it for the longitudinals instead of for the cross-members?

■ Suspension

To most people, the term "suspension" means springs.

You're right, that's wrong! There's a lot more to suspension than just springs.

As a matter of fact, there isn't a standard motor vehicle in the world that really has such a thing as suspension. A thing is suspended when it hangs; but our motor vehicles don't hang from the axles by their springs – they are propped up by them.

In Napoleonic days, though, coaches were actually suspended. High arms – extensions of the reaches between axles extended up forward and back of the body, long leather straps cradling the body between them. That's the way the first railway coaches were made. Sometimes the high arms were cantilever-type leaf springs. On the roads of that day, such suspension was a necessity.

No, I don't propose going back to the De Witt Clinton for a model to follow in designing the suspension of post-war trucks, but I do say that the higher off the ground we get our centers of suspension the safer and easier riding our vehicles will be, other things being equal.

Maybe the leaf spring doesn't lend itself to that as well as other types. Certainly the coil spring, if used directly between the axle and frame doesn't. It could be that the torsion spring is the answer. There are numerous ways in which it can be adapted – only a few of which have been tried.

When we get our rubber back, we can make splendid torsion springs out of it, for rubber works beautifully in torsion and shear.

And how about knee-action for trucks? Search me. I know of one manufacturer who tried it experimentally and found that he gained absolutely nothing for the extra weight and cost. Maybe he didn't do it right.

¿Quién sabe?

■ Powerplants

I'm afraid there will be more disappointments on the score of powerplants in post-war trucks than from any other source. The public has been fed a terrific amount of pap about the revolutionary things that high-octane gas

will do to the gasoline engine. We have been encouraged to dream of jeep-sized engines which will pack the wallop of a locomotive at a motorcycle's gas consumption.

Unfortunately, engineers have a handicap in realizing the dreams of the press-agents that the latter gentry are free of. That handicap is fact. There are two groups of facts that stubbornly obtrude:

FACT NO. 1 – We are not going to get 100-octane gas for our trucks. Even 86-octane fuel will be too costly to pay its way. Even if we got it, we couldn't come anywhere near boosting the power and economy of engines to the point predicted.

The trouble is that so long as we stick to a carbureted, throttle-controlled engine we must continue to do our work at a compression pressure well below that at which even pre-war third-grade gas pings.

High-octane gas has only one merit over that of lower rating – it can be used without ping at higher compression pressure. We must continue setting our compression ratio at the figure which will give us the maximum compression pressure which the fuel will tolerate under full-load, open-throttle conditions.

A little gain in octane in the regular grade will admit of a little gain in compression ratio. This will give us a little gain in maximum power and torque and a little better economy as long as we operate at wide-open throttle. But, unfortunately, this advantage disappears when we lower the compression by throttling the engine.

FACT NO. 2 – Higher bmep will not save much in engine size or weight. For the same horsepower, an engine with higher bmep may have somewhat smaller piston displacement, but it must have just as big a crankshaft, bearings just as large, just as heavy a flywheel, the same valve gear, and so on, as the lower compression job, for these things are determined by the output of the engine – in terms of torque and speed.

Except for somewhat shorter stroke and piston length, little is to be saved in the weight of the block. Nothing can be lopped from its length, for the length of a conventional engine is set by the space required by its twelve valves. Maybe the new engine will have to be even heavier because of the higher pressures and temperatures of combustion.

Diesels, you know, operate at very high compression and at constant compression, too, but it is necessary to cut back their bmep and still build them heavier than gasoline engines of the same power.

Of course there will be refinements. If it were not for the extravagant buildup of the octane occultists, the improvements in post-war engines would be quite impressive. This includes the small but worth-while gains that will accrue to higher octane fuel. Here are some of the things to look for after the war:

There will be generally cleaner and more accessible designs. Some of the accessories which we have been tacking onto our engines as though they were afterthoughts which had not occurred to the designer when he laid out the engine will finally be accepted as essential parts and will be provided for from the start.

Such things as oil filters, thermostats, governors, starters, and generators will be recognized at last as here to stay and will be built in, as well as the carburetor and distributor.

There will be more overhead-valve engines, principally because their valves are more easily serviced.

Bigger crankshafts, better balanced, and carried on larger, more durable bearings will be provided.

Longer-lived, more rugged valves and exhaust valve seats will be used.

There will be improved cylinder-head gaskets and hold-down studs.

Generally improved induction systems will be used – better manifold distribution, freer porting, more effective and foolproof air cleaners – more accurate and dependable governors.

A great deal more attention will be given to crankcase temperatures. Renewed attempts will be made to adapt oil coolers of various sorts, which will undoubtedly result in reduced lubricant temperatures, under heavy duty, and quicker warmup.

The real solution will be more fundamental than this and the change will involve some important byproducts in the way of better bearing alignment and engine stiffness. Maybe you will hear about it before long. It's interesting.

Every once in a while somebody goes crazy trying to dope out a perfect transmission which will make a gasoline powerplant truly adapted to highway vehicles and murmurs plaintively about steam. "If we had steam power," he muses, "there wouldn't be any transmission or clutch."

Of course he next reflects upon all the well-known reasons why steam failed to survive as a motor vehicle power source and why it can never come back.

Some day, though, some one of these may be so far gone that he forgets it can't be done and the crazy loon will do it! Don't say I predicted steam trucks right after the war, though – or even later on. You just cannot depend upon crazy people. But it could happen!

Gas turbines are undoubtedly on their way. The obstacle has always been the lack of a metal possessing sufficient strength and dimensional stability at elevated temperatures. We are a lot closer to a solution of that problem than ever before. Don't look for them immediately after the shooting stops; but they're coming.

I do not seriously anticipate rocket propulsion of motor trucks within our time, so axle makers may rest easy.

■ Cooling

I doubt if we'll ever have a perfect cooling system; not until we stop calling it a cooling system, anyway.

It's mighty important what we call things, you know. For example, we couldn't have electric refrigerators so long as we continued to think in terms of ice boxes.

What we really want in place of what we still insist on terming a cooling system is a temperature modulating system – something to maintain as nearly as possible a constant operating temperature.

We can never hope to get the maximum thermal efficiency from our engines until we learn how to maintain them within a narrow range about the ideal temperature regardless of speed, load, or ambient temperature.

We can never hope to get freedom from cylinder-head-gasket trouble, valve warpage, stud-stretching, blowby, oil pumping, oil sludging, ring coking, bearing corrosion and a host of other ills, as long as we have overheating and overcooling.

Minimum valve slash; minimum valve stem, wristpin, piston, and bearing clearances; lighter crankcase oils with safety; and minimum overheat of intake gas – these and many other aids to efficiency await narrower extremes of operating temperature.

So far, water cooling has done a better job of maintaining temperature than air; but it would be convenient if

water could get a little hotter before it boiled – particularly at high altitude.

Maybe someone knows how to control aircooling to closer limits and above 212 F. That would be fine if it could be made to work without so much horsepower going into the fan or blower – especially when we need less, not more cooling.

Until then, and for the sake of the much stiffer construction we can get with a block-cast, water-cooled engine, we will continue to interpose a liquid conducting medium between the cylinder walls and the ultimate air convection.

So far, we have relied upon two principal means of controlling the cooling of water – thermostats and radiator shutters. Both are usually mere blockers-off and as such rather callous wasters of power. One increases the resistance to the flow of water from the pump and the other that of air to the fan.

Of recent years there has been much interest in variable-pitch fans, which move the air in approximately the volume required for the degree of cooling desired. For many years some resolute manufacturers have used bypass thermostats in preference to the blocking type. The latter merely divert the flow of water; they do not reduce the load on the pump. Maybe we shall have variable-pitch or variable-speed water pumps, too.

Years ago, many manufacturers solved the problem of variable water flow by resorting to thermosyphon circulation and it did a pretty good job of temperature control, so far as the cylinder heads were concerned. Its faults, however, were many:

It widened the disparity in temperature between the head and the barrels. It required appreciably more water to provide the proper reserve for long, tough pulls, and out-size plumbing throughout. It also served best with rather high radiators to provide the gravity head desirable.

Perhaps its worst fault was that it depended for its operation upon a temperature differential between the intake and outlet. Those who have studied cylinder distortion know that the minimum difference between inlet and outlet is desirable and that the nearest approach to this condition is obtained with a thermostatically controlled hot-water bypass, which, of course, operates only with pump circulation.

It is because of a growing realization of the importance of as nearly uniform distribution of temperature throughout the engine as possible that for the past decade designers have extended water jackets down to the top of the crankcase and instead of introducing the incoming water at the lowest point have brought it in closer to the cylinder head and more and more engines are appearing with directed water flow, whereby the incoming water is evenly distributed between the cylinders and in the best examples, as close as possible to the critical region about the exhaust-valve seats.

This trend will certainly increase, as will also the trend to deeper jackets. But when the water enters the head itself and the jackets terminate in a broad exposure over the top of the integral crankcase itself, it would appear that the limit had been reached. Still we shall be far from our goal.

For a truly efficient temperature modulating system, we cannot stop just with the cylinders. Any lubricating engineer will tell you that the ideal engine would also have constant crankcase temperatures.

Most of the additives, detergents, and what not which have been placed in our motor oils are to fight the destructive effects of high crankcase temperatures. Most of the superrefinements of stock selection, blending, and processing are for the purpose of improving the viscosity index of the oil – to maintain its consistency through the widest possible range of temperature.

One way of maintaining crankcase temperatures, at least within hailing distance of water jacket temperatures is by the use of oil heat-exchangers. That means lots of plumbing, however, and the results in road vehicles are hardly epochal. The idea is right, but the approach is wrong. Such temperature maintenance should be inherent in the design of the engine and not dependent upon afterthought gadgets.

All we have to do, then, is to keep on lowering the jackets until they surround the upper half – or more correctly, upper two-thirds of the crankcase and maybe extend across through the main bearing webs.

Then let us have a thermostat located in the cylinder-head jacket where it is exposed to flowing water and let this control a hot-water bypass and a variable-pitch or variable-speed fan and we'll be a lot closer to our goal.

Will these things come sooner or later? Later, I am afraid. Industry likes to creep ahead cautiously; but it does go forward. Still later, there are even better ways just waiting to be tried. If I told you about them, you'd probably say, "Why certainly, of course," but I'm not going to!

■ Clutch and Transmission

As long as we stick to the internal-combustion engine we will continue to have a clutch and transmission problem. Some real progress is being made toward the betterment of these essential but troublesome units.

Within the limitations of the conventional foot-operated friction clutch and selective transmission, a great deal has been accomplished. With the overcenter spring and shifting synchronizers, just about the ultimate refinement will have been reached in some post-war designs.

But that will not be enough. Some tentative progress has been made toward automatic torque conversion in passenger cars – where the task is easiest – which will pilot the way for some practical adaptations to heavy vehicles.

With somewhat higher power-to-weight ratios to be expected confidently in future trucks, six or eight speeds will suffice with hand shifting, while five to seven will serve for automatic or semiautomatic transmissions.

The quicker, more positively, and more timely the shifts are made, the fewer there need be. However, the range of ratios from the highest to the lowest must be in inverse ratio to the power of the engine.

Right after the war we will find that clutches and transmissions will be largely of the familiar pattern, but shortly after will come synchronized shifts, fluid drives, and dual-ranged integral gearboxes in which the two ranges are in series and not overlapping, as heretofore.

One of these arrangements will combine manual and automatic shifting to make the proper use of the gears in the largest trucks no more difficult than in a passenger car. I don't know enough about the details just yet to explain it, but it's well on the way.

Any time we care to we can have clutches that always operate smoothly, yet positively, never grabbing or slipping, and requiring either no pedal at all or very light pedal or treadle pressure. These clutches also will have extremely long-lived linings and will never require adjustment.

The elements are all known and for all I know someone may put them together soon. If not it will probably be because the clutchless transmission appears before the improved clutch is ready. The principle is to combine with power operation high spring pressure and low-coefficient lining with an inertia and vacuum governor. If that is clear, you know what I mean.

■ Drive

There appears to be no ground for expecting a revolution in final drive. It is a far cry from the day when the merits and demerits of worm versus chain versus internal gear versus external gear versus double reduction versus hypoid drives formed the chief topics of discussion wherever trucks were discussed, to the comparatively mild interest lately shown in the rivalries between double- and single-reduction and between auxiliary transmissions and two-speed axles.

The simple fact is that given sufficient power-to-weight ratio and a reasonably moderate engine speed, a transmission with a wide enough range of ratios and wheels of moderate diameter, the simple, single-reduction bevel gear offers the highest efficiency with lowest weight, least maintenance, and lowest cost of any drive, and it is only when vehicles are of such great weight that the maintenance of sufficient power-to-weight ratio becomes impractical that resort to multiple-reduction drives is warranted.

Unquestionably, until most buyers of trucks learn to judge trucks by objective characteristics instead of subjective specifications, manufacturers will continue to supply a demand for complicated drives from users who have not yet learned that gears absorb, but never produce horsepower.

There have been indirect-drive transmissions in trucks before and there may be again, just as there are today in most transit-type buses. These may, in time, be used to produce a primary reduction which will permit single-reduction bevel drives of moderate ratio with a much-to-be-wished-for slowing down of driveshaft speed.

The present ascendancy of Hotchkiss drive is probably permanent, now that the shock-reducing elasticity which it imparts to the drive is more generally understood.

Despite the universal use of all-wheel drive in all tactical Army vehicles, their use in commercial service will decrease rather than otherwise. Not only will front-wheel drive, with its costly complication, its greater weight, impediment to both ease and radius of steering, and the ungainly height of frame which it necessitates make no headway in highway vehicles, but even the four-wheel rear drive in six-wheelers will continue in the minority for reasons of cost and weight.

On the other hand, off-highway use of all manner of trucks from the jeep to the gigantic earth-moving and mining trucks will greatly increase because of the impetus of cross-country military experience and in this field, all-wheel drive will attain increased popularity in both four-wheeled and six-wheeled vehicles.

Antistall differentials, although by no means a new departure, will certainly come in for increased popularity. It awaits only a bold start on some maker's part to precipitate a transition from the bevel-gear type to ascendancy for the torque-dividing type.

■ Steering

One of the most surprising developments of late years has been the rise of all manner of unorthodox steering gears. Everyone seems to have been intent upon producing a gear which was at once efficient and irreversible. Unfortunately, the two are not compatible. So-called irreversibility in a gear is nothing other than inefficiency and while all manner of peculiar arrangements have succeeded in reducing a great deal of the friction which did not contribute materially to the irreversible effect, while retaining that which did so contribute, the gains along this line have too often been achieved at the expense of ruggedness, durability, and freedom from rapidly increasing backlash.

Real gains have been made, of course, and there are better steering gears to choose from today than ever before. In heavy trucks and buses, particularly, however, satisfactory steering control is to be obtained, in manually actuated steering systems, only at the expense of excessive reduction and consequent slow response.

There appear to be two ways out: first, the development of a really cheap and reliable power steering system, on which some real work has been done; and, second, a purely mechanical and manually operated scheme which permits the use of gearing of the highest efficiency and yet positively prevents reactions from the road reaching the steering wheel. Both are bound to be more expensive than present steering gears.

There is also much to be done in reducing the friction elsewhere in the steering system than in the steering gear itself. Steering knuckles, tie-rod and drag-link ball joints, and even the innocent appearing steering column are due for critical analysis and improvement.

It will not surprise me much if before long, on long-wheelbased vehicles – particularly buses – we see the development of four-wheel steer in modified form. We may also witness before long a revolutionary change in the steering system of ponderous off-highway vehicles, which will greatly simplify the problem. It's interesting, but, to paraphrase my military friends – it's a civilian secret!

■ Brakes

In every development, over a period of time, a point is reached when the traditional must be examined in the light of its limitations in the face of ever more exacting requirements.

Without doubt much remains that can be done to improve the behavior of the basic brake system that we have today.

Linings can be still further improved to provide needed durability without too much sacrifice of friction qualities, greater resistance to heat and moisture, and less tendency to produce sound.

Drums can be improved in resistance to heat and abrasion, rigidity, and antiaccoustic qualities.

The convecting qualities of our conventional brakes can be increased somewhat without sacrificing the mudproofness which has been found so desirable.

Certainly, still more effective means can be devised for keeping oil off the braking surfaces. The question is, can these things be carried far enough to meet the braking requirements of the future?

Much remains to be done in developing inherent balance of brakes, irrespective of variations in the traction on the different wheels. Some gain can be made in the accessibility of brakes for adjustment and renewals, for the lubrication of their mechanisms, and for freer operation.

Controllability of brakes is doubtless susceptible of considerable improvement without departing from customary

basic arrangement, but can all of these characteristics be enhanced sufficiently to meet the exactions of post-war necessity?

Already simplification of power braking has started. From a mere appliance to amplify the effectiveness of a system originally intended for manual operation, power braking has gone a good way toward integration into a genuine power braking system. But it must go a long way further.

It is high time that compressors became a part of engines, instead of being adapted accessories; that vacuum cylinders and valves, now that they have been married to hydraulic master cylinders, should become compact chassis components with standardized mounting pads provided for on standard transmissions.

With the inescapable trend toward wider rims and greater tire cross-sections necessitated by the unfortunate limitations of synthetic rubber and certain to be perpetuated by the need for lower tire pressures and fewer plies for the sake of lower operating temperatures at the higher speeds that are coming, it is a matter of serious question whether we can long continue to carry our service brakes on our wheels.

No doubt considerations of maximum safety will induce us to continue the hand brake in that location; but it is highly probable that before many peacetime years have passed we will have moved our brakes inboard.

Maybe some way will be found to extend the hubs inboard of the frame. Perhaps we shall adopt multiple-disc brakes, inclosed in oil, within the rear-axle housing, and overhanging the steering knuckles inboard.

Again, we may go back to the driveshaft brake with either the wet-multiple disc, dry internal shoe, or the external shoe type.

My prediction is that we will certainly take the service brakes away from the rear wheels, but that the front brakes, because of their relatively reduced duty and their superior cooling position will, for some time at least, stay where they are.

Of course, with a metal-to-metal, disc-in-oil type brake, operating at very high, balanced pressure, we may be able to reduce the bulk of such a brake so that room may be found for it wholly within the space encompassed by the tire, with piped connections to a convector remote from the wheel.

Whatever the shape of brakes to come, their characteristics must and will exceed the reasonable possibilities of present basic design.

We have set a decelerating rate equivalent to a 30-ft stop from 20 mph as our current standard. Yet we know that this will not only be inadequate for post-war conditions, but is inadequate today.

In the first place, a decelerating rate of 15 ft per sec squared – equivalent to the above – does not mean a stop in 30 ft from the volition of the driver until coming to rest. It means anything from 45 ft upward, according to the constancy of braking, the shortness of mechanical lag, and of driver reaction.

Furthermore, we know that a braking system which yields such results when correctly adjusted and in prime condition, loses this effectiveness rapidly in service unless accorded a degree of maintenance diligence quite beyond the skill, instrumentation, and means of the average owner.

Our objective need not be a higher instantaneous decelerating rate, but it must be greater persistence of this rate throughout the braking cycle and under prolonged practical operation.

Brake fade must be licked. Brake balance must be maintained. It will not be enough that the braking force must be equalized between the two sides. It must be self-adjusting to differences in traction on the two sides.

Fore-and-aft proportioning, similarly, must not be a fixed quantity. It must be self-adjusting in the same manner, particularly as regards the rather large differences in proportionate loading as between the light and loaded condition and on varying gradients.

Desirable, too, if not strictly a must, is some inertia control to avoid that toe-stubbing action at the end of braking when the driver lacks finesse. Subway-train stops are particularly to be avoided in buses and in trucks with fragile loads.

Just as in the clutch, we must boldly strike away from the limitations inherited from manual operation. With practically unlimited available actuation pressure, we can replace high-coefficient friction surfaces for sterner stuff and reduce the frequency of both adjustment and refacing, so that the brakes can retain a high potency for long periods, unaffected by climatic changes and free from fade.

Of course there is another path to take, if we chose; that is, to devise automatic adjusting means and such ease of relining that short-lived, high-coefficient linings may be adapted to our needs for a long time to come. The issue will be decided, of course, on the cost sheets and by the preferences of buyers.

We must, we will have better brakes – and soon.

■ Cab and Controls

Driver comfort and convenience are now established objectives, not solely on humanitarian grounds, nor yet mainly as an aid in employee relations. On grounds of safety, efficiency and economy, the easing of strain, both physical and nervous, is good business.

No such thoughtful consideration was given the 3-mph teamster, the 15-mph chauffeur of the early power wagons, or even the moustachioed grip-man in the cable car. In the age of speed, however, we have learned that protection from wind and weather are essential, for the faster we travel the more wind and weather there is.

The teamster in his overcoats and sweaters, flannel underwear, felt-lined boots, earmuffs, and laprobe had no need of a heater; but fast trucks require unfettered feet and arms, keen ears and eyes.

There was no ventilation problem in an open seat, but in a closed cab, draftless fresh air is life itself. So the closed cabs, ventilators, heaters, and cushioned seats of motor trucks are here to stay.

Luxurious as those of yesterday seemed, however, the post-war cabs will far excel them. Fully adjustable driver's seats are as necessary as adjustable rear-view mirrors and for the same reason: because leg-lengths, trunk-lengths, neck-lengths and arm-lengths are not and never will be standardized.

Even if eugenics might bring forth uniform human dimensions in the future, adjustable seats would still be desirable for the sake of the easing of strain obtainable by slight variations in posture. Future driver's seats will not only adjust forward and back, but vertically as well, with the pitch of the back-rest variable as well.

Ventilation must be improved – particularly for summer comfort. Heaters will be arranged so that in summer they

will induce a cooling circulation of fresh air where it is needed most – about the driver's feet.

The opening windshield will surely disappear. It is the worst ventilating device in the world. It is a perfect trap for stinging insects and almost impossible to make rain-tight. The bottom-mounted wiper is by all odds superior to the top-mounted one and requires a fixed windshield.

Much improvement is indicated in cab doors. Truck driver's feet are proverbially big. There must be room enough between the front of the door opening and the seat riser, even if it does disturb the external symmetry to slant the bottom of the door forward or make a forward offset in it.

Sturdier sun-visors are required, so formed that they cover the space above the center. With these, the windshield can be raised high enough to afford long-waisted drivers good vision without stooping.

It is doubtful if metal toe- and floorboards will persist for long. They are too infernally hot in summer and frigidly cold in winter. Modern resin-bonded plywood gets my vote.

Maybe tomorrow's cabs will not have coach doors. I don't like sliding doors, but there are several other ways of hanging a door so that it gives an unobstructed opening of sufficient width without swinging so far out. There is a particularly clever arrangement being worked out – but, oh, I'm sorry.

Very little of radical nature can be anticipated in controls. We shall have a gearshift lever as long as there are gears to shift – even if we shift them by power. It will probably sprout from the steering column, where Pierce-Arrow put it 30 years ago, now that all passenger cars have it there.

The three pedals – clutch, brake, and throttle – may give place to two treadles. The brake treadle will be on the left and the throttle on the right.

The hand brake alongside the seat – to its right, of course – will gain popularity because it is more out of the way and in that position gives the driver much more effective use of his muscles.

The foot button beam switch on the left floorboard is undoubtedly a fixture and save for the need for greater accessibility for the light switch, the control pulls on the instrument board seem to be satisfactory.

The control for the turn indicator, which will probably develop into blinking tail-lights and parking lights, as in automobiles, will preferably follow passenger-car practice and take the form of a small lever above the gearshift lever. The useless and ineffective telltale light on the instrument board will give place to a loudly clicking flasher relay to remind the driver when he leaves it on after rounding a turn, or else a steering-wheel-operated shut-off will also be borrowed from our pleasure car friends.

The direction in which improvements in controls are to be looked for is less in type than in placement. There is a great need for careful study of such things as steering column angle, stroke and direction of pedals and levers, and dimensional relationships of the controls.

If drivers are to be discouraged from riding the clutch pedal, then designers must provide a comfortable place, other than on the pedal for the driver's foot. There is need for an adjustable heel rest for the right foot, so placed that the foot can be swung from accelerator to brake pedal easily without lifting the foot. There must be kneeroom under the wheel and the accelerator treadle should be wide enough to permit alternative positions of the foot to avoid cramping on long drives.

■ Tires

Never again is the truck industry likely to undergo so thorough a revolution as that precipitated in the twenties by the pneumatic tire. That revolution, however, was not a matter of an overnight change. It is still in progress and the end is not in sight.

Even had the epochal impact of a throttled rubber supply not struck us, the rapid changes in carcass structure, cord material, rim widths and contours, and tread design, nevertheless, would have sharply influenced post-war truck and bus design.

Just at present we are face to face with a crisis in tires for heavy trucks. By this I do not mean the present scarcity of tires. That is just one of the emergencies of war. The surprising thing about it is that it is not a great deal worse than it is. The crisis of which I speak is the inferiority of synthetics as a material for large tires.

Unless and until the chemurgic geniuses who have brought us thus far are able to build yet mightier molecules than Buna S, we must either lighten our loads considerably or sharply increase our tire volumes. Economically, the former is unthinkable. Legally, we are stopped from the latter.

As a war measure we may be granted an overall width of 102 in. at the tires and thus be able, not without some difficulty, to fit tires of wider cross-section. Perhaps such a liberalization might become permanent. If not, then we are in for some rather fundamental changes in basic chassis design.

There is a way, of course, of narrowing our frames and still maintaining stability. We can also place our tires one behind the other, as in six-wheelers, instead of abreast. But make no mistake about it – once larger cross-sections and wider rims arrive, we will never return whence we came.

In return for this newest of innovations forced upon us by the pneumatic tire, it is to be hoped that those who build them will grant the boon so long and earnestly sought by the SAE, namely, a simplified line of tires whose capacities are more happily graduated than in the past.

■ Accessories

What this country needs is a good reliable gadget that will sound an alarm when tire pressure drops 5 psi – and a truck quiet enough so you can hear it!

That's not a prediction, it's just a wish. I do predict certain accessories, however, because they are needed and the means of attainment well known:

One of these is a set of instruments which, instead of moving tiny hands over dainty dials will sound a buzzer whenever anything is wrong. This buzzer can be so wired to a selector dial that it will stop buzzing when the knob is turned so the indicator points to the trouble.

Certainly a simple enough circuit is involved, and by making it a closed-circuit operation, any derangement in the wiring will start the buzzer sounding.

Such an alarm system will make no demands upon the driver's attention – (And how many drivers bother watching the dials anyway?) until something goes wrong.

When it starts the buzzer, the driver will naturally dial it out to stop the racket and when he does, he cannot escape the warning. The dial might be spring-returned so that the silencing of the buzzer will last only so long as

concluded on p. 556

The DESIGN fo of SHEET M

THE problem of sheet metal production design is separated into two parts, the determining of working limits of elongation and compression in different types of sheet metal forming, and the application of these limits to the design for formability. This paper is concerned with the second part of the problem.

An analytical method of approach is suggested and is illustrated by five examples.

In the case of curved flanges, flanged holes, and beads, formulas are given by which the amount of elongation or compression called for by the design may be computed directly from the dimensions given by the design.

A possibility for a refinement in the use of the formulas is indicated, and an experimental method for achieving this refinement is pointed out.

ONE branch of production engineering is known as production design engineering. The purpose of production design engineering can be stated briefly as follows:

To control the design of the product, both in preliminary and detail stages, so that the manufacture of the product may be facilitated.

One of the many phases of the work of the production design section in the engineering department of an airplane manufacturing company is the control of the design of sheet metal parts. This is the general topic dealt with in this paper.

Of the various aspects of sheet metal production design, one of the most important, and one presenting many interesting problems, is the formability of the sheet metal parts for the airplane. In discussing formability we are interested in two questions, "Can the part be formed?" "If so, can the design of the part be altered in such a way that the forming of the part will be accomplished more quickly and/or with less expensive tools?"

The practice of a few years ago in regards to determining the formability of sheet metal parts for airplanes was to ask a man experienced in sheet metal forming whether or not he thought the part could be made. Drawing upon his experience, he gave his answer, basing it upon his memory of what had happened in the forming of similar parts in the past. But there are at least two weaknesses in this practice. In the first place, a part that appears to be similar to a remembered part may actually have some differences in detail which affect the forming. In the second place, one man's experience of this sort is very difficult to impart to any one else, either for the purpose of interchange of information between two experienced men or for the purpose of training additional personnel to supply the needs of an expanded industry. Because of these weaknesses and others, there is a growing tendency in the aircraft industry to seek scientific methods of determining formability which will give more precise information than is possible when only personal, individual experience and judgment are available, and which are capable of being used by any reasonably well-trained personnel that may be assigned to the problem.

The various types of forming consist of one or more of these three classes of strain: elongation, compression, or a combination of the two. There are two varieties of the third class, bending and draw. In bending, the fibers on the outside of the bend are stretched, while the fibers on the inside of the bend are compressed. In drawing, both elongation and compression are parallel to the surface of the sheet, normal to each other. Actually, this description should be applied to what might be called simplified draw. In practice, this simplified or ideal draw is usually accompanied by some extra elongation in a direction at 45 deg to that of the drawing elongation or compression.

In considering the general problem of formability of sheet metal parts, there are two separate phases to the problem, which may be stated in the form of questions:

1. How much elongation or compression can the metal be subjected to under various conditions?

2. How much elongation or compression does a given design call for? Or, given upper limits on elongation and compression for various conditions, how are we to design within these limits?

Investigation is already progressing towards an answer to the first question. The 16 references given at the end of this paper all concern this problem to a greater or lesser extent. In particular, see References 1, 2, 5, and 11. The question is not easy to answer, however. The elongation commonly referred to in describing the mechanical properties of the sheet metal in question is determined on the basis of tensile coupon tests over what might be described as medium lengths. For example, the elongation given for aluminum alloys is usually specified as referring to a length of 2 in. Tests have shown that much higher elongation (measured in percentage, as is customary) can be achieved

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for PRODUCTION

METAL AIRCRAFT PARTS

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over shorter lengths, such as ¼ in., which are met with in bending over small radii. (See References 1 and 2.) It has also been found that much higher elongations are possible when accompanied by compression in a direction normal to the direction of the elongation, as in drawing. (See References 4, 5, and 6.) In general, it may be said that the possible percentages of elongation and compression depend upon the conditions in which they are found in the formed part. Thus it is necessary, in investigating the first question, to consider the various types of formed shapes, so as to include the various conditions in which the elongation and compression occur.

The answer to the first question is also affected by the types of forming tools that are used. Some tools are not capable of stretching or shrinking the metal as much as are others. If the answers found to the first question are to be used in design, this may be important, because the tool needed for a certain amount of shrink, for example, may be too expensive to be justified by the size of the production run that is being planned.

Production design engineers should have available the results of up-to-date experiments which will show the upper limits on elongation and compression for all the various forming conditions under which they occur, for various materials, and for all types of tools and equipment that may be used. These results will serve as working limits to be used in controlling the design of formed sheet metal parts.

One way in which these experiments could be handled would be through a group set up particularly for the investigation of this sort of problem and other problems related to production. It might be called the "production research group."

The problems brought out by our second basic question fall within the province of the sheet metal group in the production design section of the engineering department. Taking the working limits mentioned above, in whatever state of completeness or incompleteness they may be, this group must be prepared to attack the problem of applying these limits to design, in other words, to be able to answer the second question. This problem is now being studied to some extent. See References 12, 13, and 14.

It is the primary purpose of this paper to present an analytical approach to the problem of designing sheet metal parts within the working limits that have been set. This analytical approach will be presented by means of examples of its application to certain common types of forming. A limitation of the analytical method will be pointed out, and a type of experiment described which will be useful in overcoming the limitation.

■ Curved Flanges

In the forming of concave curved flanges we have an example of one of the forms in which elongation occurs, in convex curved flanges one of the forms in which compression or shrinking occurs. Before deriving formulas for the nominal unit elongation and the nominal unit shrink it is necessary to define the terminology and notation used in discussing flanges. Fig. 1 illustrates the notation.

W = Flange width
r = Bend radius
θ = Bevel angle
ϕ = Bend angle

Note that in 90-deg flanges the bevel angle is 0, in open flanges ($\phi < 90$ deg) it is taken as negative, while in closed flanges ($\phi > 90$ deg) it is taken as positive.

We also define the "flat pattern flange width," which is

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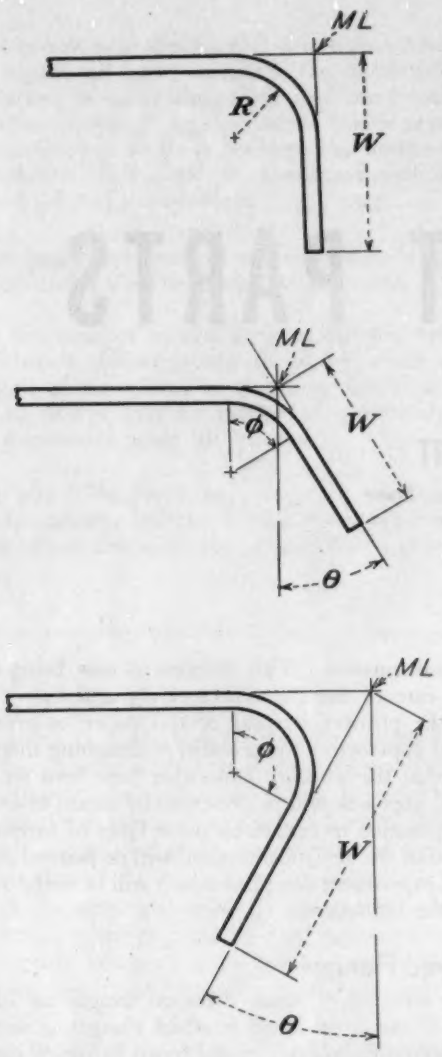


■ Fig. 1 – Upper: 90-deg flange; middle: open flange (θ negative); lower: closed flange (θ positive)

denoted by Y, in Fig. 2. Suppose we have a simple angle of leg width A and W. The width of the blank from which the angle is formed will be less than $A + W$ by an amount we denote by S, and call "setback." This setback is given by the formula:

$$S = 2(r + T) \tan \frac{\phi}{2} - \frac{\pi}{180} (r + KT)\phi \qquad (1)$$

where T = metal thickness.

The factor K is usually taken as being 0.446, sometimes as 0.5. Tables and a nomogram exist from which values of S may be found.

We now define

$$Y = W - S \qquad (2)$$

The term "mold line" (abbreviation, *ML*) will be used to refer to the intersection of outside surfaces of a flanged part.

We will now measure the nominal amount of elongation in concave flanges. Fig. 3 shows a 90-deg concave flange. The mold line is assumed circular, of radius R_m. For parts where the concave mold line is not a circle arc, the unit elongation at various points along it can be found by approximating the radius of curvature at these points and using it in place of R_m in the formula. From Fig. 3 it is obvious that the length of arc of the flat pattern edge (radius R_E) must stretch until equal to the length of arc of the mold line. Thus, if E denotes the unit elongation:

$$E = \frac{\omega R_M - \omega R_E}{\omega R_E}$$

Since $Y = R_M - R_E$, we have

$$E = \frac{Y}{R_M - Y} \qquad (3)$$

also:

$$Y = \frac{E}{1 + E} R_M \qquad (3')$$

$$R_M = \frac{1 + E}{E} Y \qquad (3'')$$

Fig. 4 shows an open concave flange. Here the flat pattern edge is stretched only until it equals the arc length of the circle arc having R_V as radius. Therefore

$$E = \frac{\omega R_v - \omega R_E}{\omega R_E}$$

From Fig. 4 we see that $R_v = R_M - V$, also $R_M = R_E + Y$, and $V = W \sin \theta$

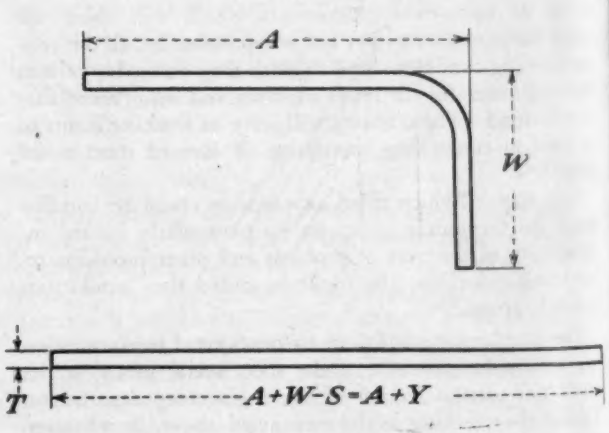
Therefore

$$E = \frac{Y - W \sin \theta}{R_M - Y} \qquad (4)$$

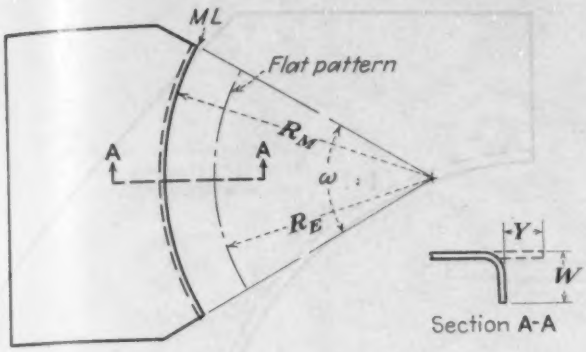
Fig. 5 shows a closed concave flange. Here we have $R_v = R_M + V$, so equation (4) becomes

$$E = \frac{Y + W \sin \theta}{R_M - Y} \qquad (5)$$

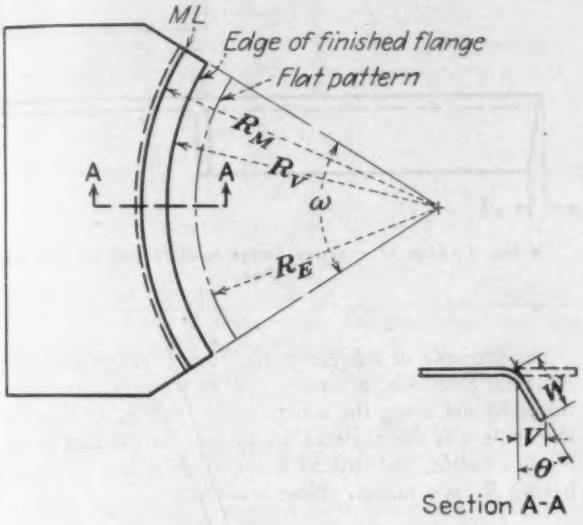
However, note that if we use the convention on the sign of θ noted above, equation (4) becomes identical with equation (5). Also, equation (3) is only a special case of equation (5) because the sine of 0 is 0. Therefore (5)



■ Fig. 2 – Simple angle



■ Fig. 3 – 90-deg concave flange



■ Fig. 4 – Open concave flange

is our basic equation for unit elongation in concave flanges.

This same generalization is possible for the case of shrink in convex flanges, so it is necessary here to derive only the case of the closed flange. Fig. 6 illustrates this case. Note that the length of arc having radius R_E must be shrunk to the length of arc having radius R_v, so, letting $C =$ unit shrink,

$$C = \frac{\omega R_E - \omega R_v}{\omega R_E}$$

We still have $V = W \sin \theta$, we also have $R_v = R_M - V$ and $R_E = R_M + Y$,

thus

$$C = \frac{Y + W \sin \theta}{R_M + Y} \qquad (6)$$

and this is our basic equation for unit shrink in convex flanges.

Note that for 90 deg flanges we have

$$C = \frac{Y}{R_M + Y} \qquad (6')$$

$$Y = \frac{C}{1 - C} R_M \qquad (6'')$$

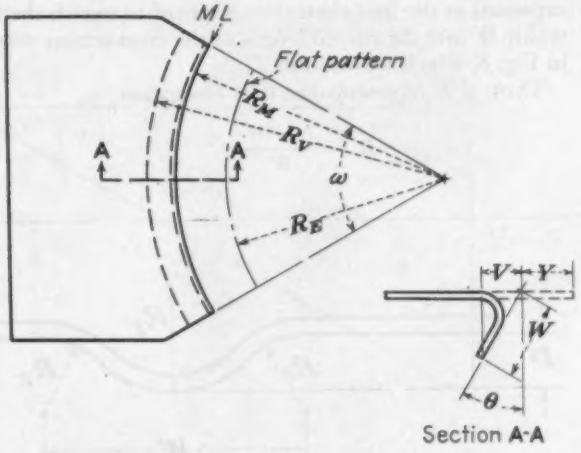
The applications to design for equations (5) and (6) are obvious. For a given maximum unit shrink or elongation there is a direct relationship given by these formulas between mold-line radius and flange width. In the design of the part these two quantities must be adjusted so as not to violate this relationship. For example, in a 90-deg concave flanged part of 0.064 in. 24 S-O, with 0.09-in. bend radius, we may be allowed a maximum elongation of 14%. Then, if a ⅜-in. flange is desired, the mold-line radius must be 4 in. or more. If a mold-line radius of 3 in. is desired, then the flange width must be ½ in. or less. Nomograms for these equations have been prepared by the author and have proved very useful in the drafting room. See Reference 14.

There is one limitation upon the use of these formulas for nominal shrink and elongation. When the ends of the curved flanges are free they tend to pull in, in the case of a concave flange, as shown in Fig. 7. When the concave flange is contiguous with a straight flange, a small adjacent portion of the straight flange is also stretched. The effect of this is to make the actual stretch a little less than that computed, noticeably so for short lengths of flange. Corresponding effects are observed for convex flanges. The formulas are thus conservative. This is part of the general problem of local distribution of strain, which will be discussed later.

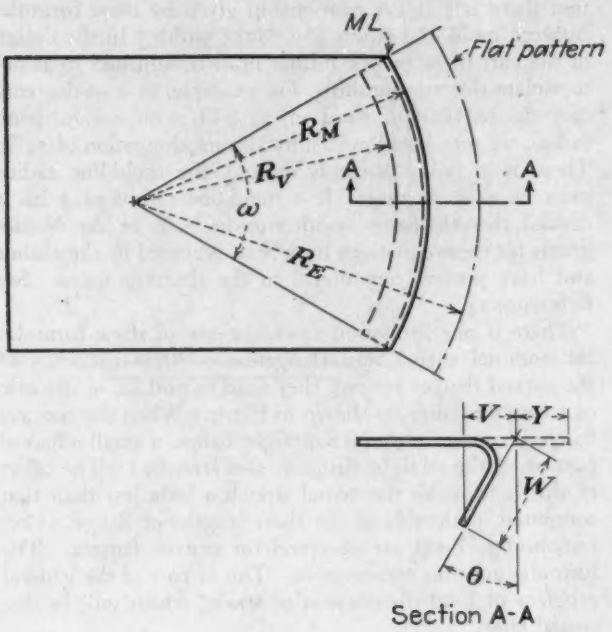
■ Beads

The formed depression used for stiffening a sheet metal part, commonly known as a "bead," is shown in Fig. 8 in cross-section normal to the lengthwise axis and one end is shown in Fig. 9 in cross-section through the lengthwise axis. The terminology is:

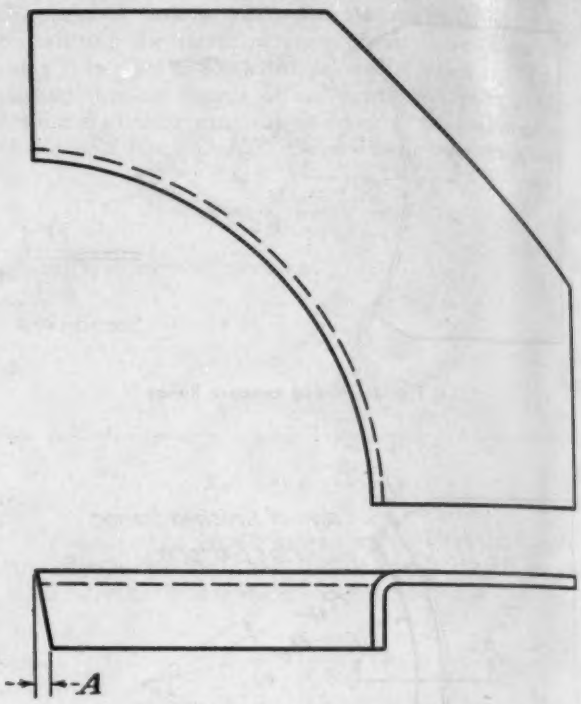
R_1 = Internal radius
R_2 = Edge radius
R_3 = End radius
$R_4 = R_1 + R_2$
D = Depth
W = Width



■ Fig. 5 – Closed concave flange



■ Fig. 6 – Closed convex flange



■ Fig. 7 – Ends of concave flange tend to pull in when they are free

The width may be computed from D and R_3, using Fig. 10, which is a right triangle whose vertices are the centers of R_1 and R_2. Using the Pythagorean theorem:

$$R_3^2 = (R_3 - D)^2 + (1/2W)^2 \text{ or } W = 2\sqrt{2R_3D - D^2} \quad (7)$$

Beads have various proportions, but a good standard shape that is easily formed is given by the following proportions:

$$R_1 = 2D,\ R_2 = 3/2D,\ R_3 = 12D \quad (8)$$

Using formula (7), we would then have $W = 2\sqrt{6D}$

The nominal elongation required to form a bead can be expressed as the unit elongation required to stretch the flat width W into the curved length of the cross-section shown in Fig. 8, which we will call L.

Thus, if E represents the unit elongation

$$E = \frac{L - W}{W} \quad (9)$$

For the sake of simplicity, measure L along the center of metal thickness, accepting the very slight error introduced by not using the neutral axis. In Fig. 10 note that the angle α is the angle of arc of each of the arcs having R_2 as a radius, and that 2α is the angle of arc for the arc having R_1 as a radius. Note also that

$$\alpha = \text{arc cos} \frac{R_3 - D}{R_3} \quad (10)$$

and is to be expressed in radians.

Letting k denote half the metal thickness, we have

$$L = 2(R_2 + k)\alpha + (R_1 - k)\,2\alpha$$

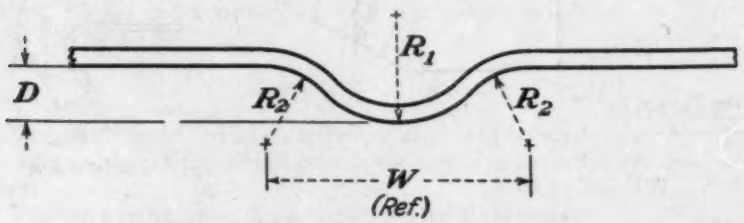
or

$$L = 2\alpha R_3 \quad (11)$$

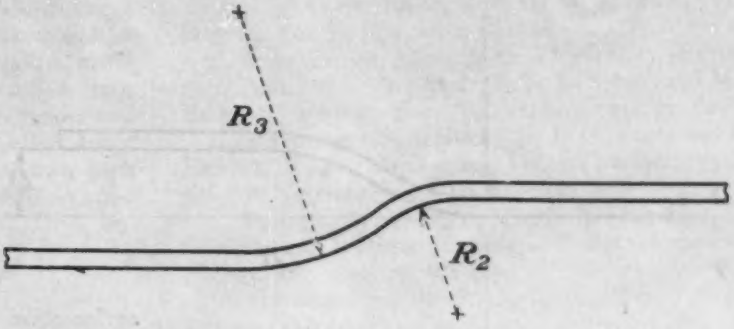
Substituting (10), (11), and (7) in (9) we have

$$E = \frac{R_3 \text{ arc cos} \dfrac{R_3 - D}{R_3}}{\sqrt{2R_3D - D^2}} - 1 \quad (12)$$

This Formula (12) gives the nominal unit elongation in



■ Fig. 8 – Cross-section of bead normal to lengthwise axis



■ Fig. 9 – Cross-section of one end of bead through lengthwise axis

terms of the essential variables, the depth and the radii sum.

Taking the bead proportion suggested in (8), we see that the elongation required for this bead is a little less than 11%.

As stated before, Formula (12) gives elongation in terms of two variables. Let us introduce a parameter x as follows:

$$R_s = xD \tag{13}$$

Then we will have

$$E(x) = \frac{x}{\sqrt{2x-1}} \text{ arc cos } \left(1 - \frac{1}{x}\right) - 1 \tag{14}$$

Form (14) is useful in graphing, as it gives elongation as a function of one variable only. The graph is shown in Reference 13. The procedure in the use of the graph is as follows: Given bead proportions, add the radii to find R_s and divide by the depth to find x. Find the elongation from the graph. Given a maximum elongation, find the corresponding x. Multiply this by the desired depth to get R_s.

Certain interesting facts can be found from Equation (14) by the use of a little differential calculus. Let $F(x) = E(x)+1$. Also let

$$F(x) = \frac{H(x)}{G(x)}$$

where $H(x) = \text{arc cos } \left(1 - \frac{1}{x}\right)$ and $G(x) = \frac{\sqrt{2x-1}}{x}$

As x becomes larger and larger, the limit of each of these is 0, using L'Hopital's theorem on $G(x)$. $F(x)$ is thus an indeterminate form. Using L'Hopital's theorem twice we have

$$\lim_{x \to \infty} F(x) = 1$$

Therefore:

$$\lim_{x \to \infty} E(x) = 0$$

Notice also that $E(x)$ is impossible for $x<1$, because no real angle can have a cosine greater than 1. However, reference to Fig. 8 shows that $x = 1$ is the least value of x that has any meaning for our problem.

$$\text{At } x = 1, E = \frac{\pi}{2} - 1,$$

or about 57%. When $x = 2$, E is about 21%. The value $x = 3\frac{1}{4}$ gives the favorable elongation of 11¼%. In Formulas (8) we have $x = 3\frac{1}{2}$.

The Formula (12) or (14) is very useful in the design of beads. However, it is conservative because in practice, the stretching occurs not only within the bead, but to a slight extent in the flat metal adjacent to the bead. The effect is particularly noticeable in rubber forming of beads that protrude away from the form block, and is also noticeable in punch and die forming. See Reference 13. Once again we are faced with a matter of specific distribution of strain. This will be discussed later.

■ Flanged Holes

The ordinary flanged hole is merely a concave curved flange. However, the so-called "beaded" or "return flange" flanged holes present a different problem. The easiest way to solve it is to find from actual forming experience the diameter of the hole in the blank. Call this D_c. Let D_F be the diameter of the formed hole. Then we have

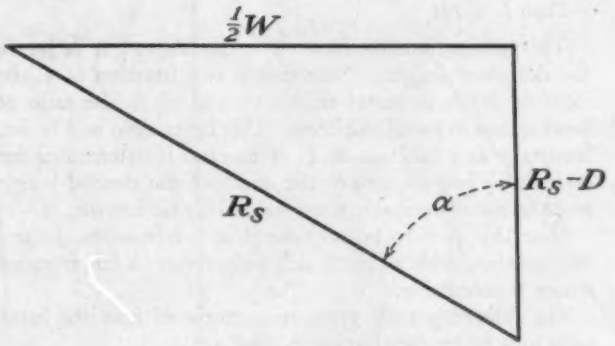
$$E = \frac{2\pi D_F - 2\pi D_c}{2\pi D_c}$$

or

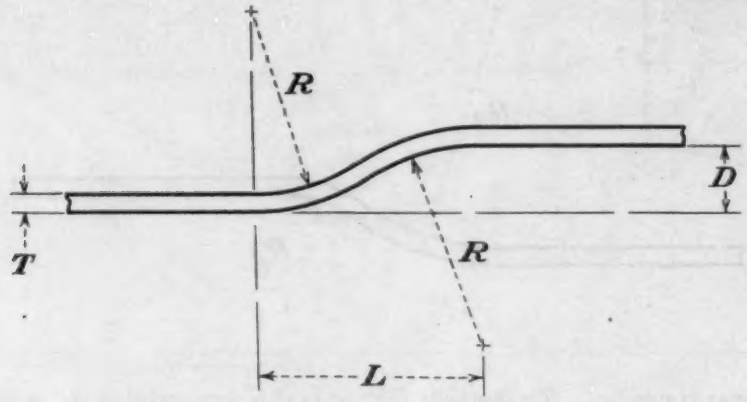
$$E = \frac{D_F}{D_c} - 1 \tag{15}$$

■ Bending

An analytical approach to the problem of minimum bend radius is presented in Reference 1. However, so many extraneous conditions affect this problem that it is better



■ Fig. 10 – Right triangle whose vertices are the centers of R_1 and R_2



■ Fig. 11 – Basic terminology of joggles

to have the production research group determine bend radii directly, presenting the figures to production design engineering to use.

However, there is an application of bending which presents an object for study, the design of joggles.

Fig. 11 illustrates the basic terminology for joggles.

D = Depth of joggle
L = Length of joggle (or "joggle allowance")
R = Bend radius
T = Metal thickness

What is usually desired in the design of joggles is as small a length as possible. This can be accomplished by making the two bends tangent to each other. In this case, R, D, and L are not independent. Fig. 12 helps to derive the relationship between them. The angle β is the bend angle, or angle subtended by the arcs.

$$(2R + T)^2 = L^2 + (2R + T)^2 - 2D(2R + T) + D^2$$

or
$$L^2 = 2D(2R + T) - D^2 \qquad (16)$$

Let $D = k \cdot T$ and $R = b \cdot T$

Then
$$L = \sqrt{(4b + 2)\,k - k^2} \cdot T \qquad (17)$$

also
$$L = \sqrt{\frac{4b + 2}{k} - 1} \cdot D$$

Let
$$p = \sqrt{\frac{4b + 2}{k} - 1} \qquad (19)$$

Then $L = pD$

This proportionality factor p is the important factor in the design of joggles. Note that it is a function of k, the ratio of depth to metal thickness, and of b, the ratio of bend radius to metal thickness. This latter ratio will be set, leaving p as a function of k. This ratio is determined for each individual case, since the depth of the desired joggle and the metal thickness must necessarily be known.

Note that p is to be considered as a minimum, as any larger value, within limits, will only result in larger radii, which is acceptable.

The following table gives an example of how the bend radii may be set for aluminum alloy 24S.

		Values of $b = R/T$	
		$0 < T < 0.102$	$0.102 \leq T < 0.250$
24S-O	$\beta < 60$ deg	$b = 1$	$b = 2$
	$\beta \geq 60$ deg	$b = 1\frac{1}{2}$	$b = 2\frac{1}{2}$
24S-T	$\beta < 60$ deg	$b = 3$	$b = 4$
	$\beta \geq 60$ deg	$b = 3\frac{1}{2}$	$b = 5$

Note that β is given by

$$\beta = \arccos\left(\frac{2b + 1 - k}{2b + 1}\right) \qquad (21)$$

Using the above values of b, p is represented as a function of k by means of eight formulas.

b	p
1	$\sqrt{6/k - 1}$
$1\frac{1}{2}$	$\sqrt{8/k - 1}$
2	$\sqrt{10/k - 1}$
$2\frac{1}{2}$	$\sqrt{12/k - 1}$
3	$\sqrt{14/k - 1}$
$3\frac{1}{2}$	$\sqrt{16/k - 1}$
4	$\sqrt{18/k - 1}$
5	$\sqrt{22/k - 1}$

(22)

Using the function given by (22), Table 1 can be set up that can be used by designers for proportioning joggle. However, in using the table, an exception must be made for joggles that involve other deformation beside bending, such as a joggle in a flange. In this case, p should be taken as 3, or from the table, whichever is greater.

Obviously, other values of b might be chosen, resulting in different tables. Additional tables could be made for other materials.

■ Drawing

There has been some investigation of this problem for aluminum alloys, as found in References 4, 5, 6, and 7.

Table 1 – Values for Proportioning Joggle

$D = k \cdot T$	24S-O		24S-T	
	$T < 0.102$	$0.102 \leq T < 0.250$	$T < 0.102$	$0.102 \leq T < 0.250$
$D \leq 0.2\,T$	5½	7	8½	10
$D = 0.3\,T$	4½	6	7	8
$D = 0.5\,T$	3½	4½	5¼	6
$D = 0.7\,T$	2¾	3¾	4½	5
$D = T$	2¼	3	3¾	4¼
$D = 1.5\,T$	2	2½	3	3½
$D = 2\,T$	1¾	2	2½	3
$D = 3\,T$	1½	1¾	2	2½
$D = 4\,T$	1	1½	1¾	2¼
$D = 5\,T$	1	1¼	1½	2
$D = 6\,T$	1	1	1¼	1¾
$D = 8\,T$	1	1	1	1½
$D = 9\,T$	1	1	1	1¼
$D \geq 11\,T$	1	1	1	1

It is probable that the information supplied to production design by the production research group should take a direct form. The common deep-drawn shapes should be classified cylindrical, rectangular with round corners, hemispherical, and so on. For each, a maximum ratio of depth of draw to radius should be determined. Other ratios could be specified if redraw were permitted. Production design could then judge the formability of various drawn parts suggested by designers and help proportion them to avoid redraw where possible.

■ Local Distribution of Strain

As was stated above, refinements may be made on the use of formulas for elongation and shrink, although they are already useful, by studying the specific local distribution of strain in the various types of formed parts. The technique for doing this has already been developed. It is presented in Reference 3 and is made use of in References 1, 2, 4, and 5, and possibly in other places. It consists of applying by photography a network of lines, rather closely spaced, either rectangular or polar (and others could be used), to the blank which is to be formed. The strain resulting from forming can then be approximately measured by comparing the spaces between lines on the formed part with the original spacing on the blank.

If this method were applied to the forming of a bead in a flat sheet it could then be determined how much elongation is found in the metal adjacent to the bead. This would not only be useful as a refinement to the formula for elongation but even more so in determining clearance distances that should be maintained between beads and other formed features of the part.

This type of investigation should be carried out for every type of forming. In some cases the refinement might not be needed, in other cases it would, but in all cases it would be better to know.

■ Conclusion

There are some types of formed parts not mentioned in this paper. (See Reference 16, Section I.) However, as stated in the beginning of this paper, its purpose was to describe by examples a method of approach to the problem of designing sheet metal parts for formability. The problem is a large one and much work remains to be done. The author hopes that this contribution will be useful in the task ahead.

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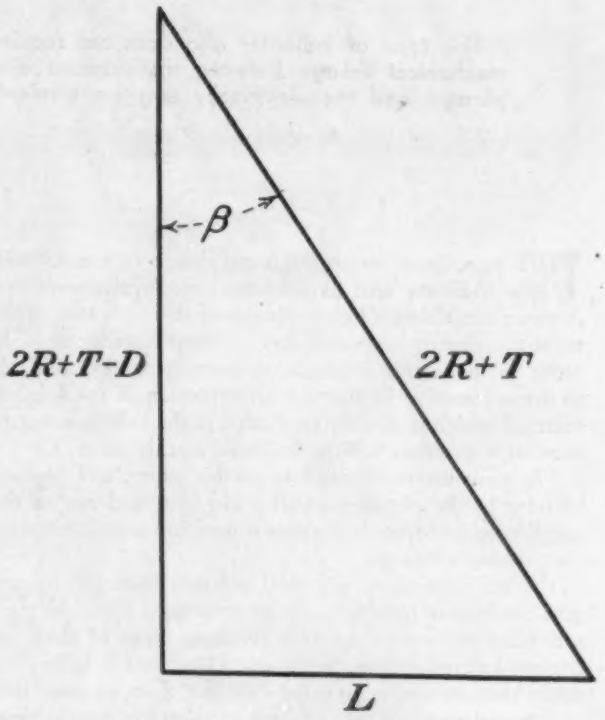

■ Fig. 12 – Right triangle used to derive the relationship between *R*, *D*, and *L*

AN IMPROVED INDICATOR for M

by C. E. GRINSTEAD, R. N. FRAWLEY, F. W. CHAPMAN and H. F. SCHULTZ

Research Laboratories Division
General Motors Corp.

THE principle of operation and the mechanical design of an improved indicator for measuring static and dynamic pressures are discussed in this paper.

The condenser type of indicator was selected by the authors for engine work because it lends itself to an exceptionally compact and sturdy construction, it is easily serviced, it has a high natural frequency, and it is relatively insensitive to shock and vibration.

This type of indicator also does not require mechanical linkage between the pressure diaphragm and the electrically sensitive element.

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THIS paper presents an improved design of a condenser-type indicator and its associated electrical equipment. A paper describing a former design of this indicator equipment has already been published. (See Reference 25.) In order to make this information more generally available to the automotive industry, a large portion of the original material which is directly applicable to the indicator instrumentation problem will be included in this paper.

The term pressure indicator, in this paper, will be used to refer to the pressure-sensitive element, and not to the amplifying and recording systems used in conjunction with the pressure elements.

The selection of an electrical pressure indicator for engine work may involve a choice of several kinds of pressure-sensitive devices, as well as many types of electrical circuits and recording methods. However, it is beyond doubt that no single pressure indicator with its associated equipment meets all the demands of complete engine pressure analysis. Nevertheless, with a fairly complete knowledge of the defects and limitations of the various kinds of indicators available, the engineer may successfully cope with most of the problems of pressure measurements.

Before discussing the characteristics of the various types of pressure elements, it might be well to consider the important requirements of an ideal engine indicator and its associated amplifying and recording equipment.

1. The ideal indicator must give a linear and undistorted response to any instantaneous pressure encountered in the engine to be studied.

2. The natural frequency of the indicator must be well above any frequencies to be measured.

3. It must have an accurate, predetermined calibration that will not be affected by temperature, time, fatigue, or external forces such as vibration or shock.

4. It must be insensitive to all forces except the pressures applied to the diaphragm, that is, it must not pick up engine vibrations. This means that the moving element must have low mass.

5. It must be capable of indicating an atmospheric or

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MEASURING STATIC and DYNAMIC PRESSURES

any other desired pressure datum line.

6. It must be small in physical structure in order that it may be easily installed in modern engines.

7. It must be reasonably simple to operate.

8. It must be capable of recording pressures during a single engine explosion.

At the present time, no known pressure-indicating equipment meets all of these requirements. Fortunately, however, the engine development engineer can compromise on some of these requirements and still obtain instrumentation which is dependable and useful.

Indicators capable of measuring pressures for a single engine cycle are listed in the following classes:

1. Resistance-type indicators:
a. Carbon stack.
b. Carbon granule.
c. Solid carbon resistor.
d. Wire resistor.
2. Piezoelectric crystal-type indicators.
3. Magnetic-type indicators:
a. Moving diaphragm.
b. Moving coil.
c. Magnetostriction.
4. Condenser-type indicators.

All types of pressure-sensitive units have characteristics which make them useful for certain applications. Likewise, most of these indicators possess characteristics which make their use unsatisfactory for other applications. The principal advantages and disadvantages of the various types of indicators and pressure-sensitive devices may be briefly outlined to present their possibilities and limitations for any given application.

1. Resistance-type indicator.

Advantages:

No electrical amplifiers are needed to operate a magnetic-type galvanometer-oscillograph. This is especially true of the carbon-stack indicator, which may be constructed to have a high current output.

Disadvantages:

a. Instability of calibration (particularly under severe conditions).
b. Responds to severe vibration.

2. Piezoelectric crystal-type indicator.

Advantages:

a. Highly linear response.
b. Constancy of calibration.

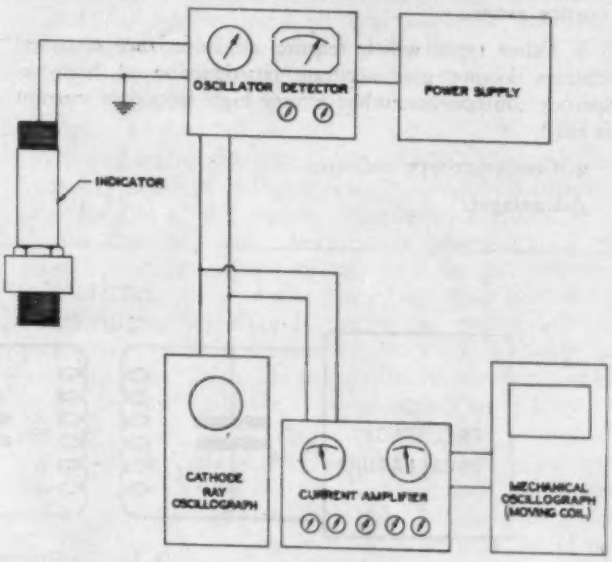
Disadvantages:

a. Requires extremely high-gain electrical amplifiers.
b. Responds to severe vibration.
c. Extreme care must be taken to minimize leakage in the input circuit of the high-gain amplifier.
d. In the present state of development, it is unsuitable for low-speed work, static pressure measurements, and for the measurement of extremely low pressures.

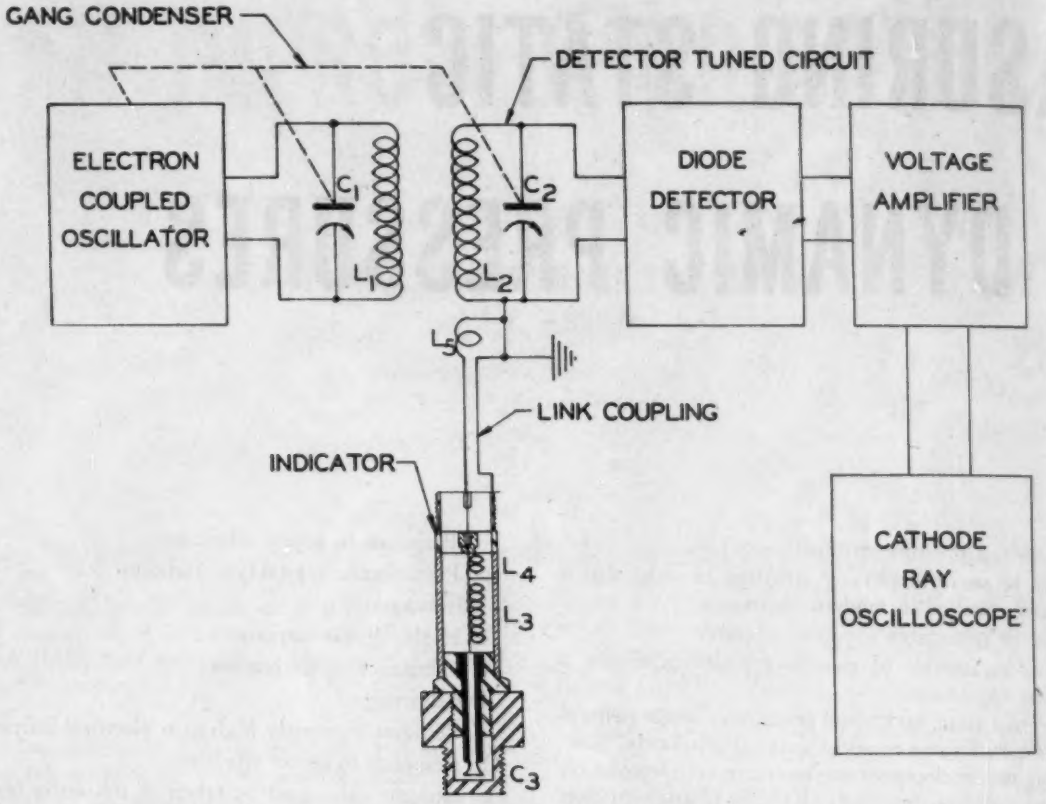
3. Magnetic-type indicator.

Advantages:

a. One type gives rate of pressure change directly.
b. No mechanical linkages are required between the



■ Fig. 1 – General arrangement of equipment



■ Fig 2 – Electrical circuit for equipment

diaphragm and the electrically sensitive element.

Disadvantages:

a. For some types a complicated high-gain amplifier is required to integrate accurately the rate of pressure-change response into pressure-time response over the required frequency range.

b. Other types which require a-c impedance electrical circuits do not give accurate reproduction of high-frequency components unless a very high frequency current is used.

4. Condenser-type indicator.

Advantages:

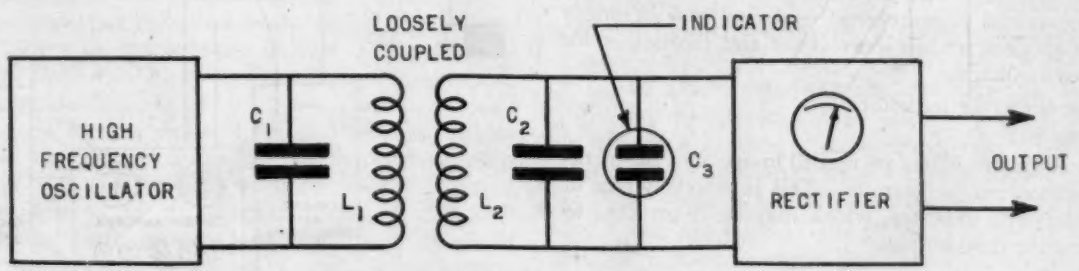
a. As will be shown in this paper, this indicator lends itself to an exceptionally compact and sturdy construction, which is easily serviced, has a high natural frequency, and is relatively insensitive to shock and vibration.

b. No mechanical linkage is required between the pressure diaphragm and the electrically sensitive element.

Mechanically, this indicator offers no disadvantages that are not present in the other types of indicators.

The associated electrical equipment for measuring the change in capacity of the indicator may offer certain limitations, depending upon the type of circuit used. For example, the electrostatic type of circuit for measuring small changes in capacity has the following limitations:

1. High-gain electrical amplifiers are required.



■ Fig. 3 – Original electrical circuit

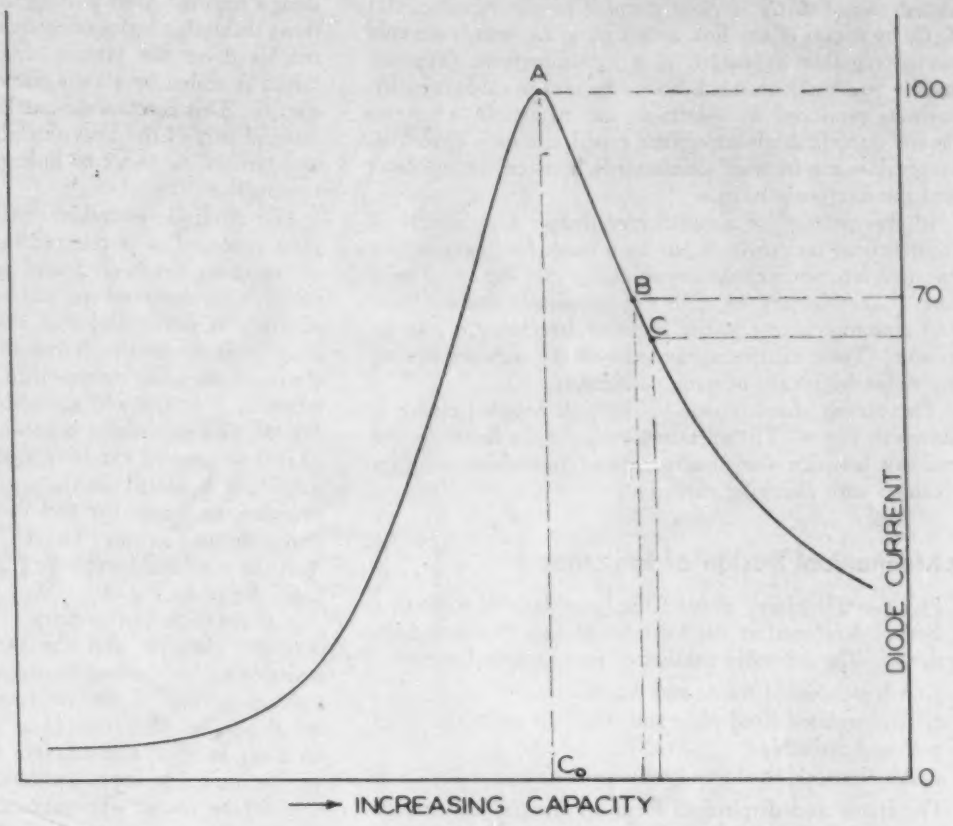


Fig. 4—Response of resonant circuit to a constant frequency shown as a function of the capacity of the circuit

2. Limited low-frequency response.

3. High-input impedance which makes the electrical amplifiers very sensitive to ignition interference.

Some of the above limitations were overcome by using a resonant impedance circuit in conjunction with a condenser-type indicator developed in the General Motors Research Laboratories. One limitation of the original equipment was the method of connecting the indicator to its associated electrical equipment. The capacity of the connecting cable was in parallel with the capacity of the pressure indicator itself and, therefore, any change in cable capacity produced by vibration was superimposed on the capacity variations of the indicator. To minimize this difficulty, it was necessary to use short lengths of low-capacity cable. This cable limitation required that part of the electrical equipment be mounted near the indicator, which in some applications was very inconvenient, if not impossible. In order to eliminate the effect of cable vibrations and to provide for remote location of the electrical equipment, an improved electrical circuit has been developed which utilizes a low-impedance link coupling between the indicator and the electrical equipment. Cable lengths up to 100 ft have been used satisfactorily. The importance of this remote operation is well illustrated in applications on propeller torque stands and in dynamometer rooms of high-output engines. This feature eliminates the necessity of mounting the oscillator detector unit near the engine where it is subjected to air blasts, noise, and excessive temperatures. This feature also makes it unnecessary for operating personnel to enter the test cell.

Principle of Operation

Figs. 1 and 2 show the general arrangement of the equipment. The indicator illustrated has been drawn in its elementary form for simplicity. Its capacity is formed by the diaphragm and the insulated electrode. In the original electrical circuit, as shown in Fig. 3, the capacity C_3 is a part of a tuned resonant circuit, $L_2C_2C_3$. L_1C_1 form part of a stable high-frequency oscillator circuit. The tuned circuit $L_2C_2C_3$ is loosely coupled to the circuit L_1C_1. The high-frequency potential induced in circuit $L_2C_2C_3$ is rectified, and the d-c output current of the rectifier is indicated on a microammeter.

The response of the tuned circuit $L_2C_2C_3$ to a constant frequency is shown in Fig. 4 as a function of the capacity (C_2 plus C_3) of this circuit. This curve is typical of a parallel resonant circuit. Peak output occurs at point A when the circuit is tuned to resonate at the oscillator frequency. This circuit is tuned to its operating point B by increasing capacitor C_2 until the rectifier meter reads approximately 70% of maximum. Any small change in indicator capacity C_3, due to pressure on the diaphragm, affects the response of the circuit according to the curve of Fig. 4.

In order to eliminate the effect of cable capacity between the indicator C_3 and the tuned circuit L_2C_2 the circuit shown in Fig. 2 has been developed. In this circuit a coil L_3 (See Fig. 2) has been built as an integral part of the indicator assembly. The coil L_3 is connected in parallel with the indicator capacity C_3 to form a tuned circuit with

a resonance frequency of approximately 600 kcps. The tuned circuit L_3C_3 is close coupled to the tuned circuit L_2C_2 by means of the link coils L_4 and L_5, which are connected together by means of a low-impedance (approximately 70 ohms) untuned cable. Changes in cable capacity, such as produced by vibration, are negligible when reflected into the high-impedance tuned circuit. Therefore, long cables can be used satisfactorily between the indicator and the electronic units.

In the interests of simplicity of design and rigidity of construction, no provision has been made for changing the resonant frequency of the circuit L_3C_3. See Fig. 2. Therefore, it is necessary to tune simultaneously the oscillator and detector circuits to the resonant frequency of the indicator. These circuits are tuned over the required operating range by means of gang condensers.

The tuning characteristic of the link coupled circuit is shown in Fig. 5. The operating range is the linear section midway between the maximum and minimum readings obtained with changing capacity.

■ Mechanical Design of Indicator

Fig. 6 is a cutaway view of the latest type of condenser indicator developed at the General Motors Research Laboratories. The assembly consists of four principal parts:

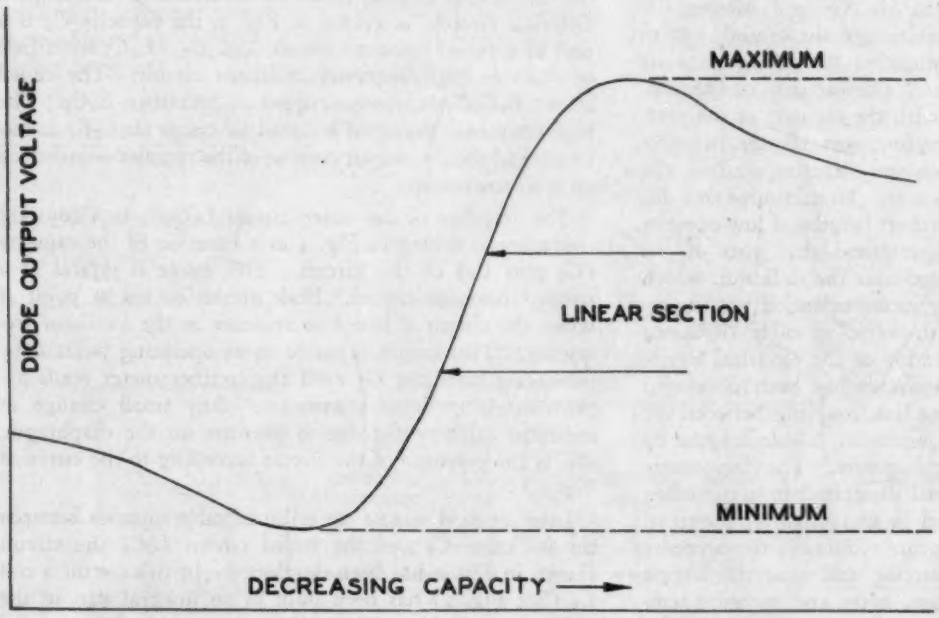
1. A water-cooled frame and diaphragm.
2. An insulated fixed plate and electrode assembly.
3. A coil assembly.
4. An electrode shield and cable connector.

The frame and diaphragm form an integral unit which is machined from a solid piece of invar steel. Invar is used to minimize the effect of temperature changes in the diaphragm. Water cooling keeps the temperature of the indicator within the range where the low thermal coefficient of invar is realized.

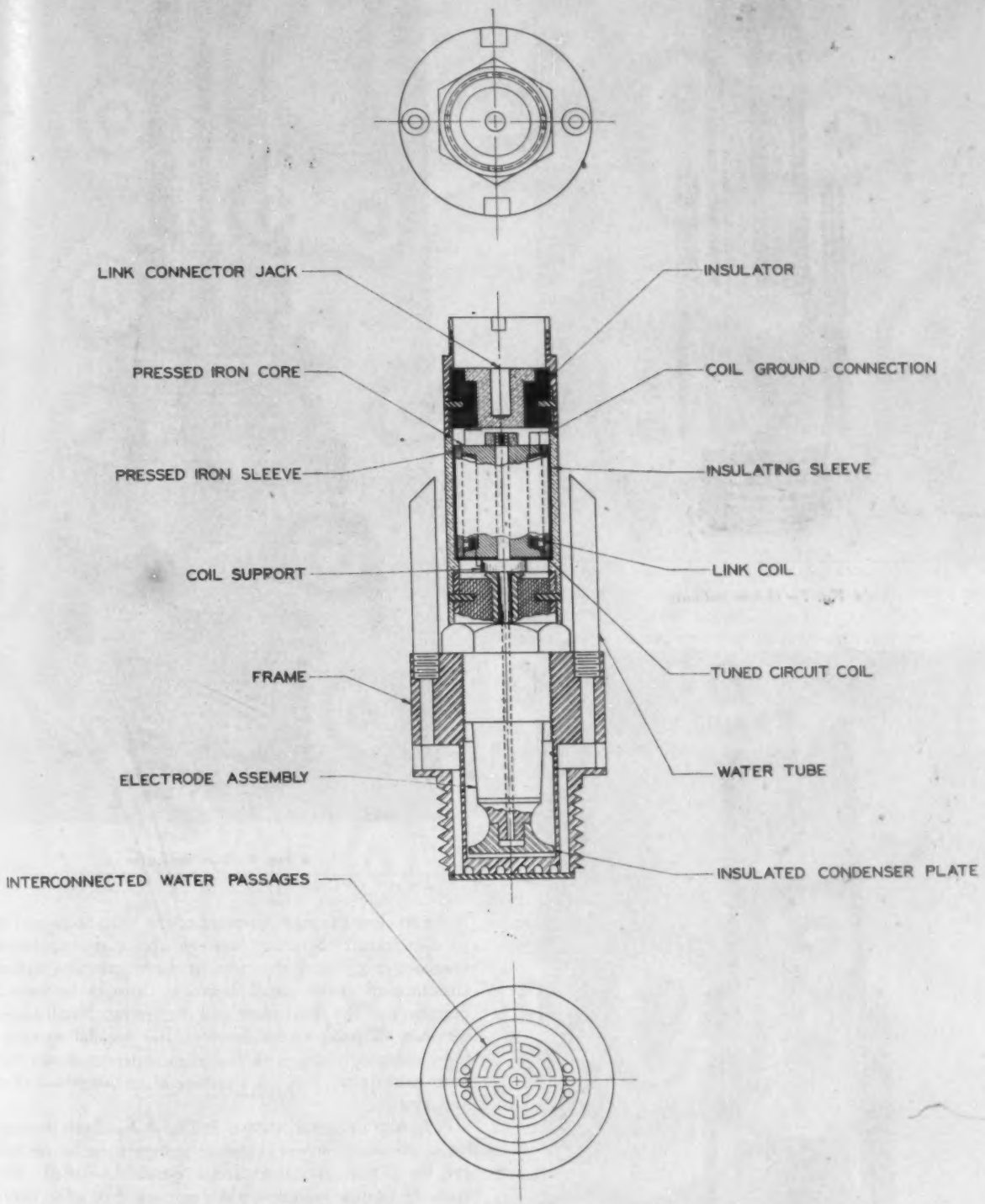
The water cooling of the diaphragm is an important design feature. After passing through the frame, the water flows through a series of interconnected concentric grooves machined on the bottom side of the heavy diaphragm which is sealed by a thin disc copper-brazed in place. See Fig. 6. This construction makes the thin sealing disc an integral part of the heavy diaphragm, and thus eliminates any mechanical joints or linkages. The diaphragm is also extremely sturdy.

The electrode assembly consists of an invar condenser plate mounted in a ceramic insulator. This combination of materials has been found to be most satisfactory for minimizing temperature effects in the electrode. This electrode is sturdy and well insulated and has small electrical capacity to the frame of the indicator. Complete electrical shielding is provided by a metal jacket which serves as a unit shield and cable connector, as shown in Fig. 6. This jacket also houses the coil assembly as shown. In order to avoid excessive eddy current losses with this small coil mounted in the metallic jacket it was found necessary to encase the coil completely with a powdered iron core and sleeve. The coil assembly in conjunction with the insulated condenser plate forms the tuned circuit L_3C_3 shown in Fig. 2.

In order to minimize extraneous capacity changes in the indicator assembly and the oscillator-detector unit, it is desirable to have a large capacity change in the pressure-sensitive portion of the indicator. Since the capacity between parallel plates varies as an inverse function, it is necessary to select a diaphragm thickness so that its movement is limited to approximately 15% of the initial air gap in order to obtain approximately linear response. Considering the natural frequency of the diaphragm, its elastic properties and limited diameter, a maximum diaphragm movement of 0.0003 in. was chosen. An air gap of approximately 0.003 in. was selected to obtain adequate initial capacity, sensitivity, and linearity. The initial capacity and



■ Fig. 5 – Tuning characteristic of link coupled circuit

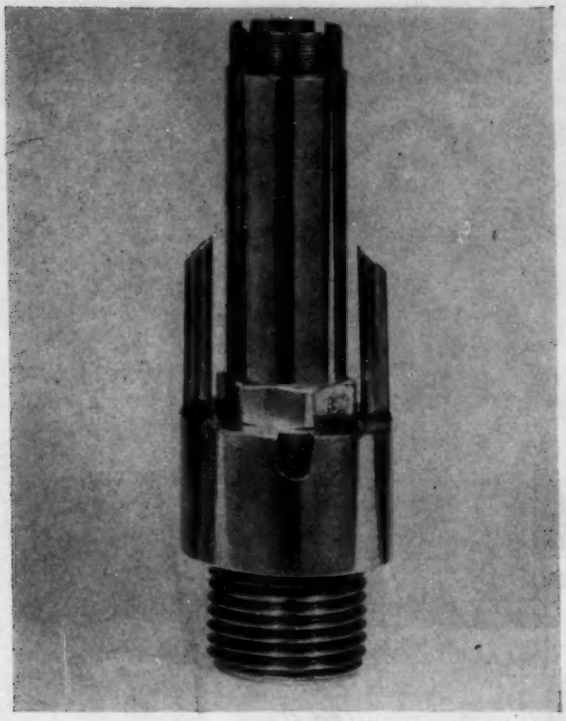


■ Fig. 6 – Cutaway view of water-cooled condenser-plate indicator – high-pressure type

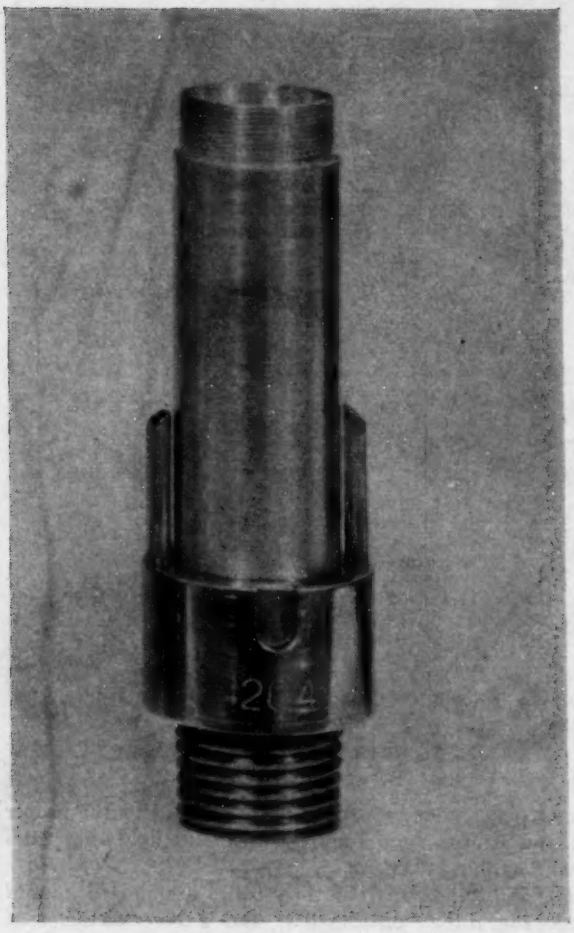
sensitivity of the indicator is further improved by the use of a 0.0015 mica separator between the plates. This mica also prevents shorting of the plates by minute dust or metallic particles.

For successful operation of the indicator, accurate machining of all parts is essential. The surfaces of the condenser plates must be parallel and separated by the proper amount. It has been found practical to hold the plates parallel within 0.0001 in.

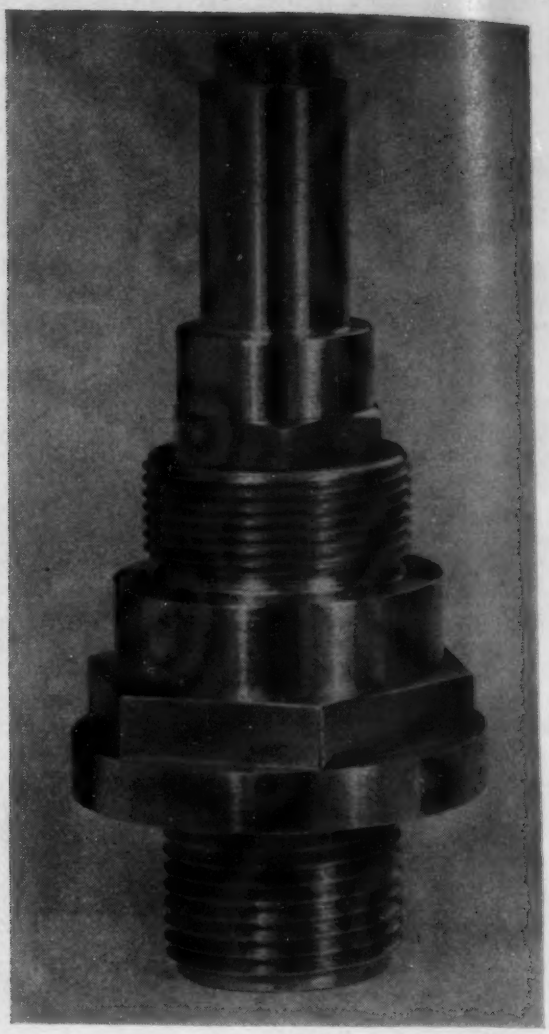
The indicator shown in Fig. 6 has a standard 18-mm spark-plug thread on the base and a bore diameter of 7/16 in. The fixed-plate diameter is held to 0.415-in. in



■ Fig. 7 – 18-mm indicator



■ Fig. 8 – 14-mm indicator



■ Fig. 9 – 1-in. indicator

order to clear the root diameter of the 7/16-60 tapped hole in the frame. Spacing between the 0.415-in. diameter insulated plate and the 7/16-in. bore provides sufficient clearance to assure small electrical capacity between the periphery of the fixed plate and the frame. Small capacity between all parts except between the parallel surfaces of the condenser plates is desirable in order to obtain maximum sensitivity. Fig. 7 is a picture of an assembled 18-mm indicator.

A 14-mm indicator, shown in Fig. 8, has been developed for applications where extreme compactness is necessary and an 18-mm indicator cannot possibly be used. However, the 14-mm indicator is not as rugged or as serviceable as the 18-mm indicator.

A large indicator with a 1-in. diameter thread has been designed to measure pressures up to 15,000 psi. See Fig. 9. Provisions have been made on this heavy-duty indicator for mounting a protective sleeve around the electrode assembly. See Fig. 10.

Since it is desirable to limit the movement of the diaphragm to approximately 0.0003 in., its thickness is determined by the peak pressure to be measured. The curves in Fig. 11 have been obtained by plotting diaphragm deflection of the water-cooled, 18-mm indicator versus pres-

sure in pounds per square inch applied to the indicator diaphragm. A suitable diaphragm thickness for any given pressure measurement may be selected by reference to this chart. For example, with peak pressures of 800 psi, a diaphragm thickness of 0.065 in. should be used.

An indicator can be used over a relatively wide range of pressures by changing the gain of the electrical amplifiers. The sturdy diaphragm construction makes it possible for the indicators to withstand reasonable overloads.

The above mechanical design incorporates the following important features:

1. The physical dimensions of the indicator are small enough to facilitate installation on modern engines.

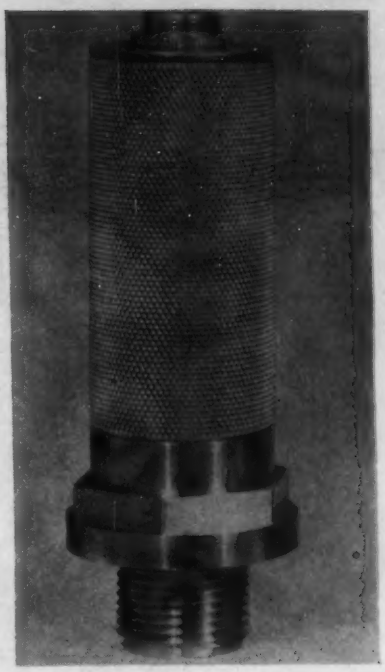
2. The construction is compact and rigid enough to minimize the effects of severe vibration, that is, the inertia forces are small.

3. The natural frequency response of the elastic system is well above any frequency encountered in engine indicator applications. (Pressure changes as high as 6000 cps are common in the combustion chamber of some engines under detonating conditions.)

4. The indicator is constructed so that it can be installed with the diaphragm flush with the walls of the pressure chamber. This type of installation eliminates the possibility of surge effects in connecting passages and avoids a change in combustion-chamber shape and volume.

5. The indicator is constructed so that it can be readily serviced and assembled.

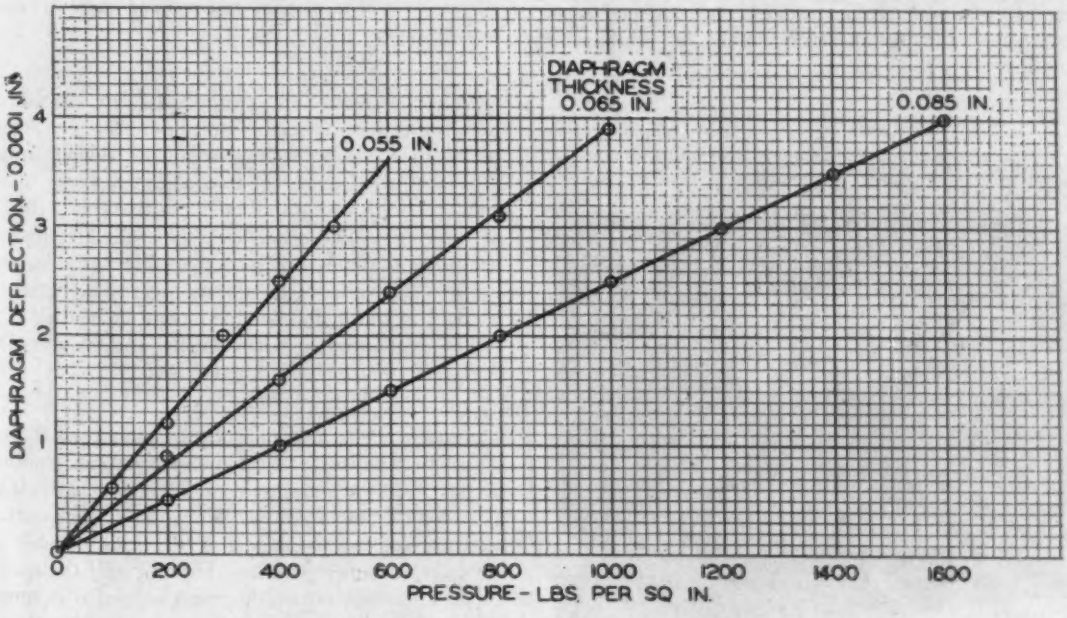
6. The design provides adequate electrical shielding and a convenient means of cable connection.



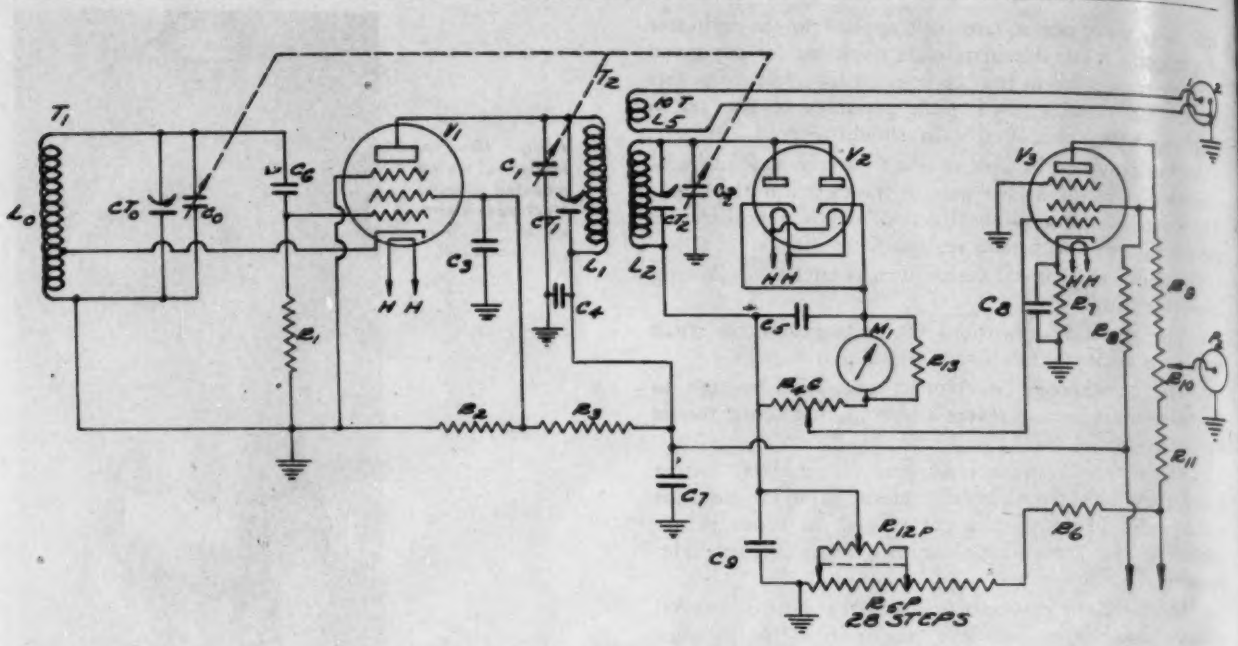
■ Fig. 10 – Protective sleeve mounted around electrode assembly of 1-in. indicator

■ Electrical Equipment

Oscillator-Detector Unit – A complete schematic diagram of the oscillator-detector unit is shown in Fig. 12. This circuit corresponds to the skeleton diagram of Fig. 2. The oscillator-detector unit is housed in a cast-aluminum case to provide rigid construction. See Fig. 13. Tube V_1, in Fig. 12, and its associated circuits form an electron-coupled, high-frequency oscillator. This type of oscillator is used because it provides relatively high output and is inherently stable. The rectifier or diode-detector tube V_2, which is connected as a half-wave rectifier, is coupled to the oscillator through transformer T_2. A microammeter M_1 reads the rectified current flowing through the diode load re-



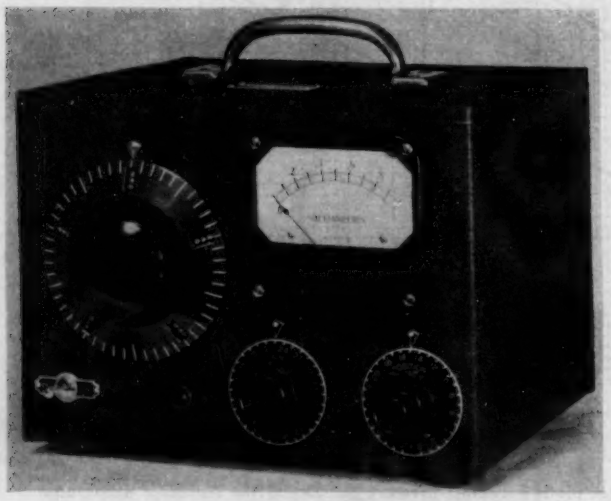
■ Fig. 11 – Diaphragm deflection of water-cooled, 18-mm indicator plotted against pressure applied to the indicator diaphragm



■ Fig. 12 – Schematic diagram of oscillator-detector unit

sistor R_{4G}, therefore, the microammeter reading is an indication of the voltage output across R_{4G}.

The circuits $C_0C_{T0}L_0$, $C_1C_{T1}L_1$, and $C_2C_{T2}L_2$ are all tuned to the same frequency. The tuned circuit $C_2C_{T2}L_2$ is coupled to the indicator by means of the close coupled link coils L_4 and L_5, which are interconnected with a low-impedance cable. The output voltage developed across R_{4G} is opposed by a negative potential source through R_{5P} and R_{12P} to reduce its static component to zero, thereby permitting direct-coupled amplification by means of V_3. This amplifier tube V_3 is primarily used to obtain a low-impedance output source to minimize high-frequency losses in interconnecting cables to cathode-ray or mechanical oscillographs.



■ Fig. 13 – Oscillator-detector unit

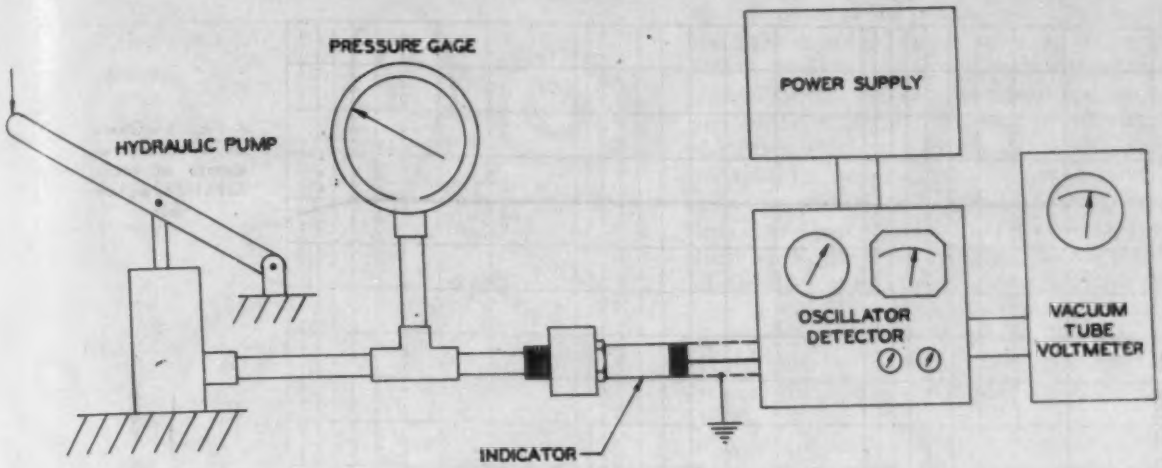
Static calibration tests were made with a condenser-plate indicator and an oscillator-detector unit to determine their overall sensitivity as well as their linearity of response to pressure. Fig. 14 shows the test equipment. A water-cooled indicator of the type shown in Fig. 7, with a diaphragm thickness of 0.065 in., was used for this test. At zero gage pressure on the diaphragm, the oscillator-detector was tuned to the center of its operating range. The pressure was then varied from 0 to 800 psi and the corresponding output voltage measured. The data plotted in Fig. 15 show that the output was approximately linear.

The overall frequency response of the oscillator-detector unit is shown in Fig. 16.

Power Supply – The overall stability of the oscillator-detector and its associated amplifier has been improved by using an electronic voltage-regulated power supply. This power supply consists of two complete units, as shown in Fig. 17. One unit supplies a positive 350 v to the plates of the tubes. The other supplies a negative 350 v for the balancing circuits necessary for direct-coupled amplification. The voltage regulation serves to guard against changes in input line voltage or output loading.

This regulation is instantaneous and, as a result, a high degree of filtering action is obtained.

Amplifiers – In order to observe or record the output of the oscillator-detector unit under dynamic conditions, the output must be amplified to operate a cathode-ray or a galvanometer oscillograph. Since the voltage output of the oscillator-detector is large (10 v), it is possible to use a low-gain amplifier between this unit and the oscillograph. When a cathode-ray oscillograph is used it is quite simple as no external amplifiers are needed besides the one contained in the cathode-ray oscillograph. A gain of 50 in the internal amplifier of the oscillograph is sufficient.



■ Fig. 14 – Test equipment

As previously explained the output of the oscillator-detector must be balanced to zero potential with no pressure on the diaphragm of the indicator. Therefore, it is convenient to use a direct-coupled amplifier to set the zero balance. The Dumont type 208 oscillograph can be readily modified for use with the oscillator-detector unit because it incorporates direct-coupled amplification in the stages immediately preceding the deflection plates.

The really important feature of direct-coupled amplification is the fact that it permits static calibration of the indicator and the measurement of low frequencies. The overall frequency response of the equipment has been built to be linear from 0 to 10,000 cps.

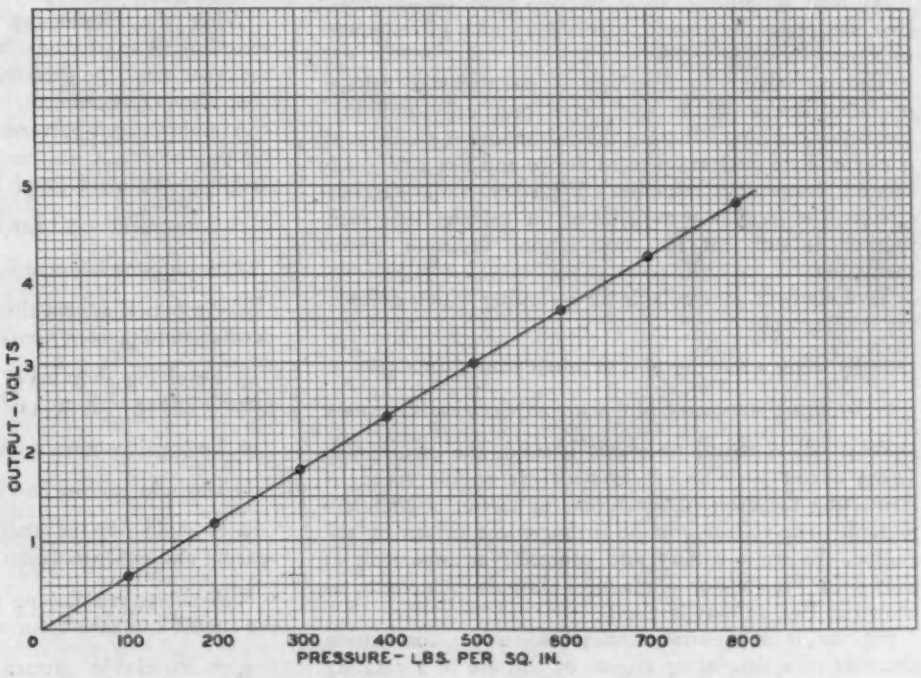
The use of a galvanometer oscillograph requires additional amplification between the oscillator-detector and the oscillograph. A schematic diagram of a current amplifier built for this purpose is shown in Fig. 18.

This amplifier possesses the following features:

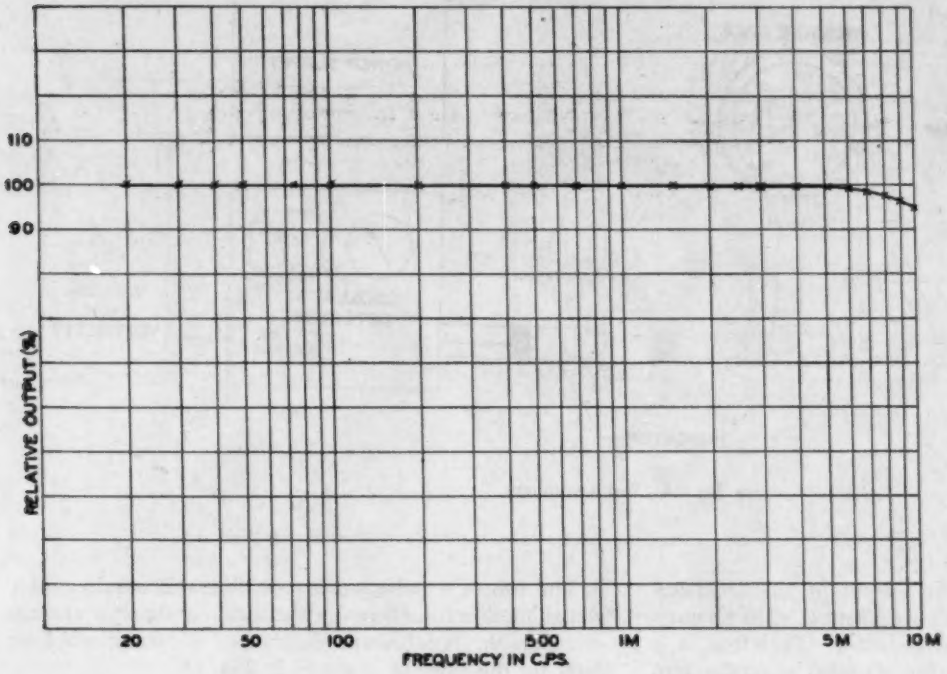
1. Flat frequency response from 0 to 10,000 cps.

2. A power source in the cathode circuit to cancel the initial cathode current flowing through the galvanometer oscillograph.

The complete schematic diagram of the indicator, the oscillator-detector, the amplifier, and the regulated power supply appears in Fig. 17. If desired, these circuits permit



■ Fig. 15 – Variation of output with change in pressure



■ Fig. 16 – Overall frequency response of oscillator-detector unit

the simultaneous use of cathode-ray and galvanometer oscillographs. This arrangement is illustrated in Fig. 19.

■ Recording Equipment

For viewing and recording of pressure phenomena in high-speed internal-combustion engines, equipment must be used that will respond to the highest frequencies to be measured. In general, there are two types of equipment that are practical, namely, cathode-ray and galvanometer oscillographs.

The cathode-ray oscillograph is essentially a voltage-operated device which may have practically unlimited frequency response, whereas, the galvanometer oscillograph is essentially a current-operated device and has a definitely limited frequency response. With practical electrical circuits, the frequency response of the galvanometer oscillograph is sufficient for most engine indicator instrumentation.

Two methods are available for recording with a cathode-ray oscillograph:

1. By using a moving film to provide for a time axis.

2. By using stationary film with an electrical time axis.

When an electrical time axis is used, the oscillograph sweep circuit may be synchronized at engine speed by connecting the external synchronizing circuit of the oscillograph to a voltage-impulse arrangement driven by the engine. Either a battery and contactor arrangement, as shown in Fig. 20, or a magnetic impulse generator, as shown in Fig. 21 may be used.

Fig. 22 is a pressure card taken in the combustion chamber of a detonating engine by the use of a rotating drum for a time axis. Fig. 23 is an indicator card taken with stationary film in conjunction with an electrical time axis.

Under some conditions it is possible to view two indicator diagrams simultaneously on a conventional cathode-ray oscillograph by means of an electronic switch. Two pressure records taken simultaneously by the above method are shown in Fig. 24. The upper and lower records show pressures measured in the combustion chamber and intake manifold, respectively.

While the cathode-ray oscillograph can be used for recording, it is particularly well adapted for viewing. On the other hand, the galvanometer oscillograph is a relatively poor viewing mechanism, but is well adapted for recording.

A multielement galvanometer oscillograph has been developed especially for use with engine indicators. The design of this oscillograph includes the following features:

1. Convenience and simplicity of operation.

2. A tungsten-filament light source.

3. An optical system that gives a uniform time axis and permits simultaneous viewing and recording.

4. Revolving short focal length lenses give high photographic speed with stationary film.

5. Four galvanometers may be used simultaneously.

6. Daylight loading and unloading of standard roll film.

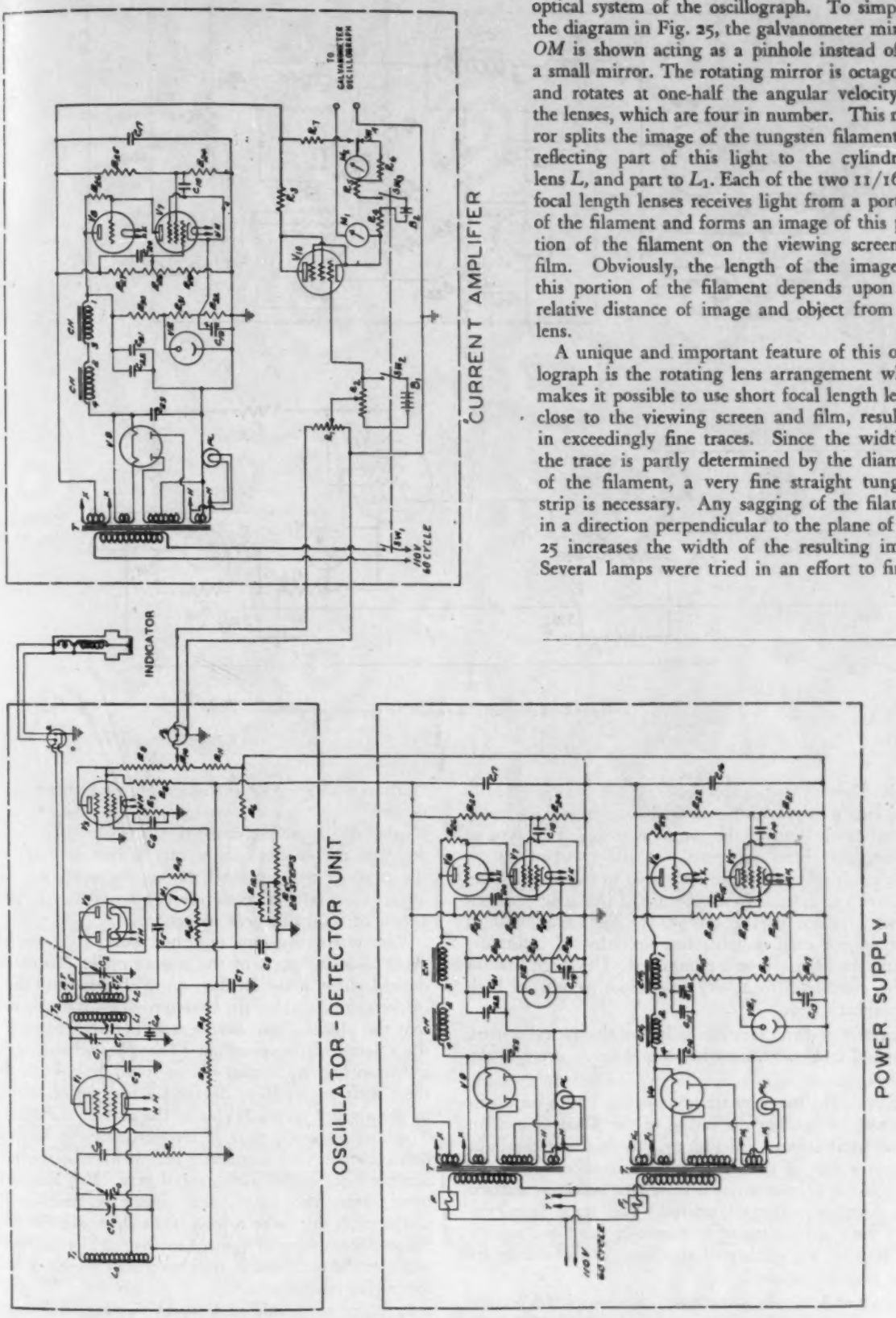
7. A rapid film advancing mechanism with automatic spacing and exposure counting device.

8. A device for phasing the oscillograph time axis with the engine crankshaft during operation.

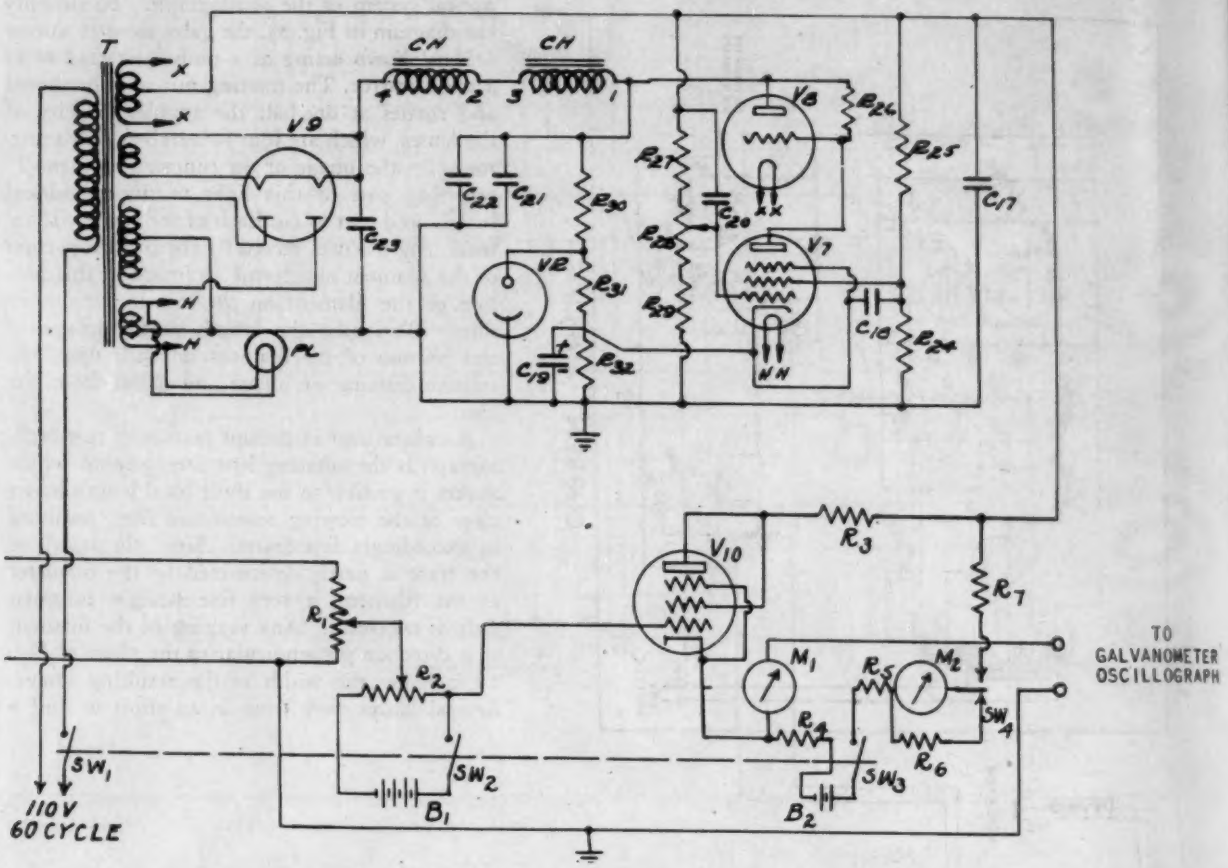
9. A barrel-type shutter with a simple timing mechanism.

Two sketches, Figs. 25 and 26, show the optical system of the oscillograph. To simplify the diagram in Fig. 25, the galvanometer mirror *OM* is shown acting as a pinhole instead of as a small mirror. The rotating mirror is octagonal and rotates at one-half the angular velocity of the lenses, which are four in number. This mirror splits the image of the tungsten filament *T*, reflecting part of this light to the cylindrical lens *L*, and part to L_1. Each of the two 11/16-in. focal length lenses receives light from a portion of the filament and forms an image of this portion of the filament on the viewing screen or film. Obviously, the length of the image of this portion of the filament depends upon the relative distance of image and object from the lens.

A unique and important feature of this oscillograph is the rotating lens arrangement which makes it possible to use short focal length lenses close to the viewing screen and film, resulting in exceedingly fine traces. Since the width of the trace is partly determined by the diameter of the filament, a very fine straight tungsten strip is necessary. Any sagging of the filament in a direction perpendicular to the plane of Fig. 25 increases the width of the resulting image. Several lamps were tried in an effort to find a



■ Fig. 17 – Schematic diagram of indicator, oscillator-detector, amplifier, and regulated power supply



■ Fig. 18 – Schematic diagram of current amplifier

suitable filament but since none of the available lamps proved satisfactory, a lamp was designed with a light spring mounted at one end of the tungsten strip.[1] This type of mounting has been successful, as the constant spring tension tends to remove the slack due to expansion when the filament is heated. With the use of this long tungsten filament in place of the carbon arc light employed in former engine oscillographs, the necessity of constantly adjusting the light source is eliminated. The large amount of light, condensed to a very small spot, results in high photographic speed.

It has been shown experimentally that the optical system, as sketched in Fig. 25, does not introduce an error in the time axis.

Uniform light intensity throughout the entire time axis is obtained by making the image of the filament overlap the cylindrical lenses. See Fig. 25. In this way the light overlaps the face of the lenses enough to take care of the shifting of the light beam as it leaves the reflecting surface of the rotating octagonal mirror. The trace from the number four galvanometer is normally used for timing marks because the quality of the trace is lowered by its extreme angular position.

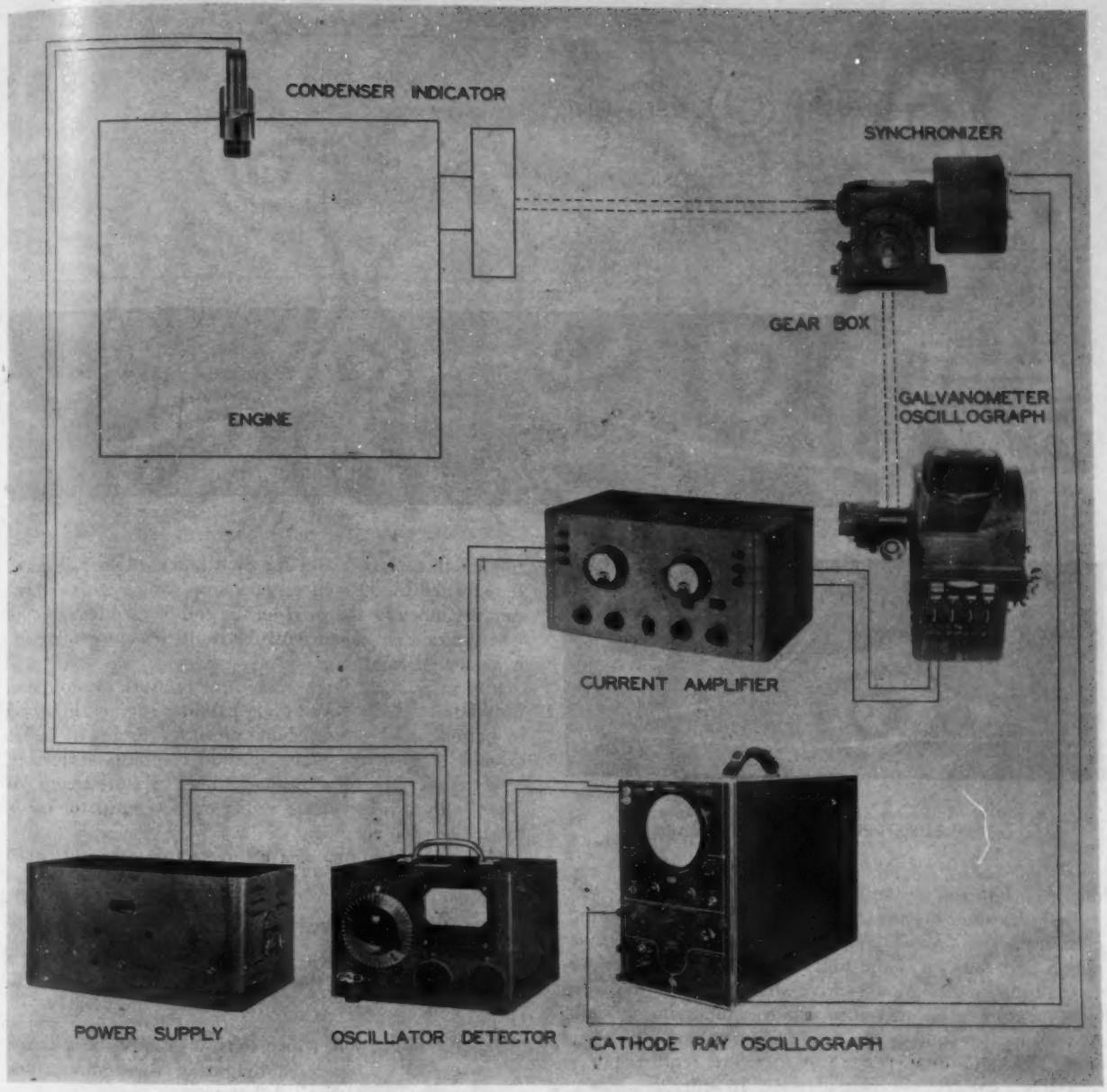
Figs. 27 and 28 are pictures of the completed oscillograph.

[1] This lamp was developed with the cooperation of the General Electric Co., Incandescent Lamp Division.

The viewing screen and recording film each extend over 30 deg of arc in the rotating optical system. In order to record 360 deg of engine-crankshaft rotation on the 30-deg arc, it is necessary to use a 12:1 reduction gear between the engine crankshaft and the oscillograph. Any desired angle of crankshaft revolution can be viewed or recorded by use of a suitable gear reduction.

Very often in engine indicator work, it is desirable to study different parts of the engine cycle. To obtain the desired phase of the oscillograph with respect to the phase of the engine crankshaft, some provision must be made so that the phase of the oscillograph may be changed while the engine is in operation. The phase-changing device shown in Fig. 29 consists of two spiral gears in mesh with their shaft angles at 90 deg and a method of sliding one of these gears perpendicular to the axis of the other gear. The driving spiral gear is mounted on the oscillograph drive shaft in such a manner that it may be moved back and forth across the mating spiral gear. This sliding movement causes the driven gear which is attached to the oscillograph mirror to revolve, thus changing the phase of the oscillograph with respect to that of the engine. The phase of the oscillograph may be changed with or without the engine running.

In order to permit continuous viewing, it is necessary that the shutter mechanism does not interrupt the light falling on the viewing screen. An inspection of Fig. 25 shows that the lower half of the light filament is used for



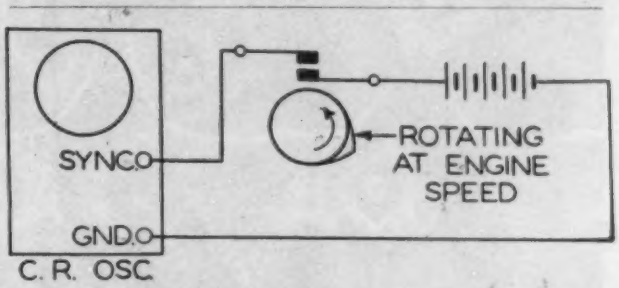
■ Fig. 19 – Complete indicator installation

viewing, whereas, only the upper half is used for recording. Therefore, a shutter is needed only on the upper half of the filament. A barrel-type shutter is shown in Fig. 27. The shutter is synchronized with the engine by means of a suitable timing device located on the mirror shaft.

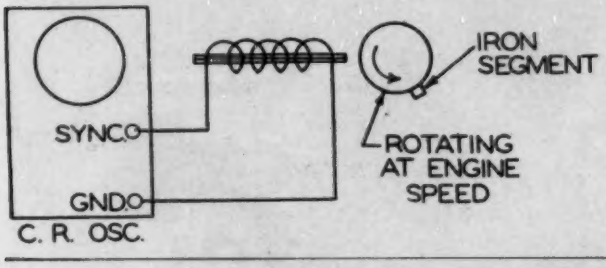
The light intensity is varied to accommodate various engine speeds by a tapped switch which changes the resistance in series with the lamp. An 110-v d-c supply is used to operate the lamp and shutter circuits.

The camera film housing, Fig. 30, is an assembly of a light-tight film holder, film indexing mechanism, and light blind.

The film indexing mechanism shown in Fig. 31 is a rack-and-pinion drive which advances the film 3¾ in. for each index, thus giving 10 single exposures on a standard Kodak daylight-loading No. 122 roll film. The disc carrying the exposure numbers is set to number 1 at the beginning of

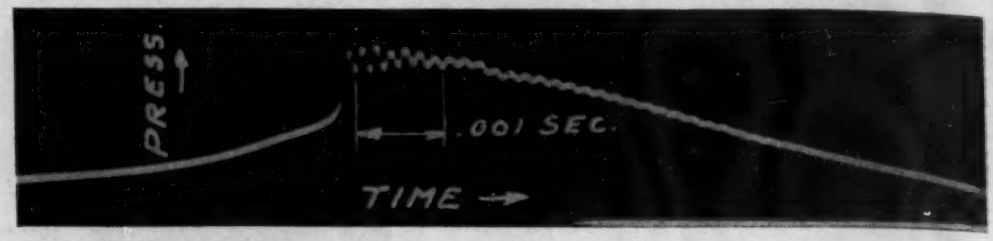


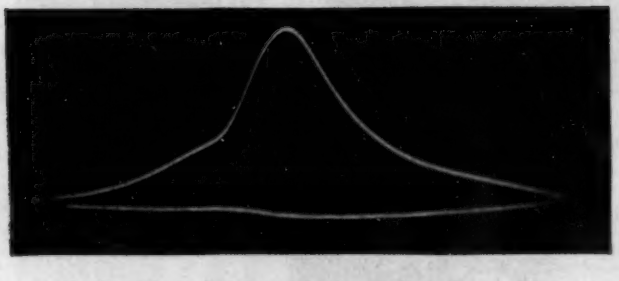
■ Fig. 20 – Battery and contactor arrangement



■ Fig. 21 – Magnetic impulse generator

■ Fig. 22 (right) – Pressure card taken in combustion chamber of detonating engine by the use of a rotating drum for a time axis





■ Fig. 23 – Indicator card taken with stationary film in conjunction with an electrical time axis

each new film roll by pressing the reset button.

To index the film, the rack is pushed down against the rack stop. This forward motion of the rack turns the drive pinion and thus turns the film driving roll. The ratchet mechanism on the pinion drive shaft prevents the film from unwinding on the return stroke of the indexing rack. As the film is wound on the film driving roll, the diameter of the roll naturally increases and, therefore, requires fewer revolutions with each successive index. In order to advance the film the 3¼ in. required for the 3-in. oscillograms, and proper spacing, the number of revolutions is controlled by the length of rack travel for each index. This rack travel is regulated by the stepped rack stop, which is moved one step per index by the pawl on the end of the film rack.

A typical card taken with the multielement oscillograph is shown in Fig. 32.

Fig. 32 is an oscillogram of three pressure records taken simultaneously on a four-cycle gasoline engine. From top to bottom, the records show pressures measured in the exhaust pipe, intake manifold, and combustion chamber, respectively. The combustion-chamber pressure record was taken with a low-pressure indicator developed for use in the combustion chamber.

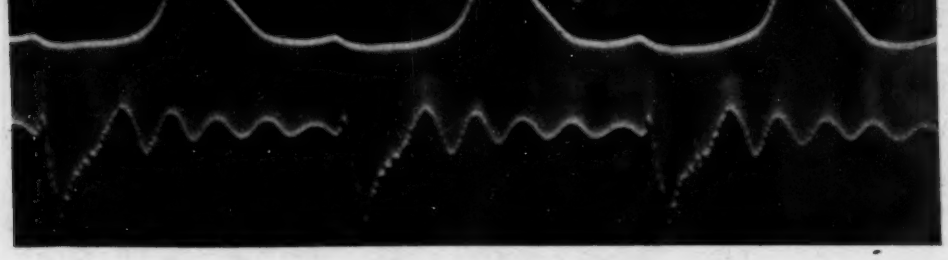
A pictorial diagram of a complete indicator installation is shown in Fig. 19.

■ Indicator Calibration

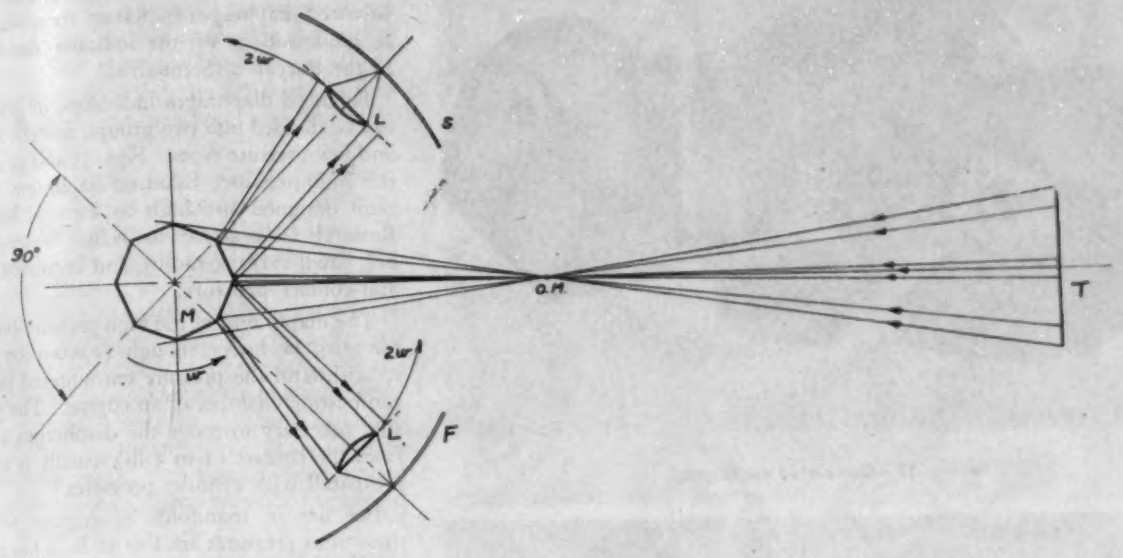
Various methods have been developed for calibrating a single-card indicator. These methods, in general, can be classified as follows:

1. Static calibration, which consists in applying a known steady pressure to the diaphragm of the indicator and recording the oscillograph deflection.

2. Dynamic calibration, which consists of applying a



■ Fig. 24 – Two pressure records taken simultaneously – upper curve: pressure in combustion chamber; lower curve: pressure in intake manifold



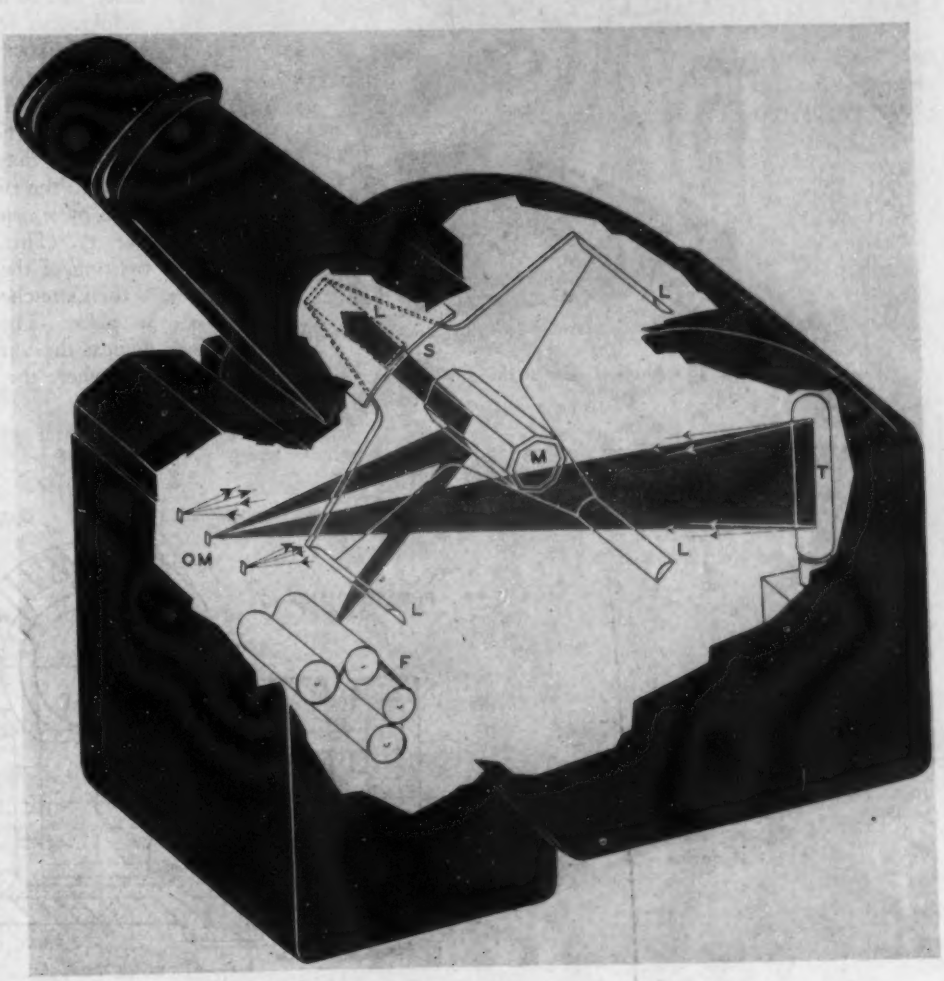
■ Fig. 25 – Schematic diagram of optical system of oscillograph

known dynamic pressure to the indicator diaphragm with a compressor or rotating valve, and recording the oscillograph deflection.

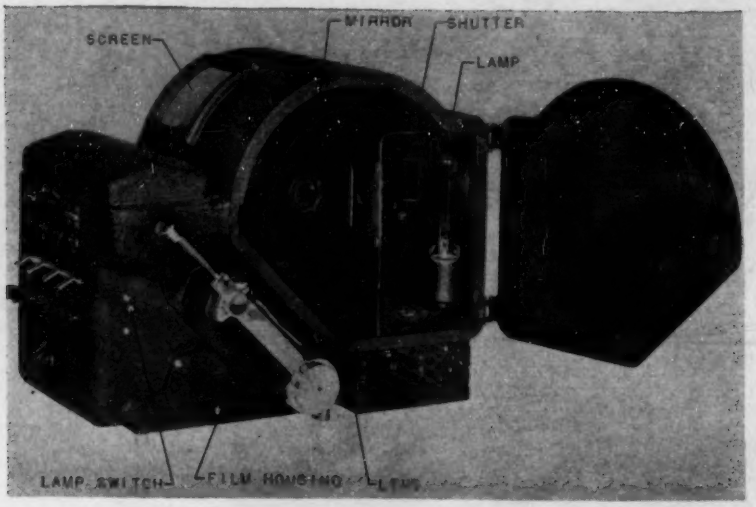
3. Calibration in the engine under firing conditions by a supplementary instrument to indicate a point in the pressure cycle corresponding to a known balancing pressure.

The disadvantage of the first two methods of calibrating is chiefly their failure to simulate actual engine conditions in regard to temperature. After calibrating in a suitable device, the indicator must be transferred to the engine. This transfer may introduce another source of error in the calibration because of the probability that the forces on the indicator frame in the calibrator will not be duplicated when the indicator is sealed in the engine.

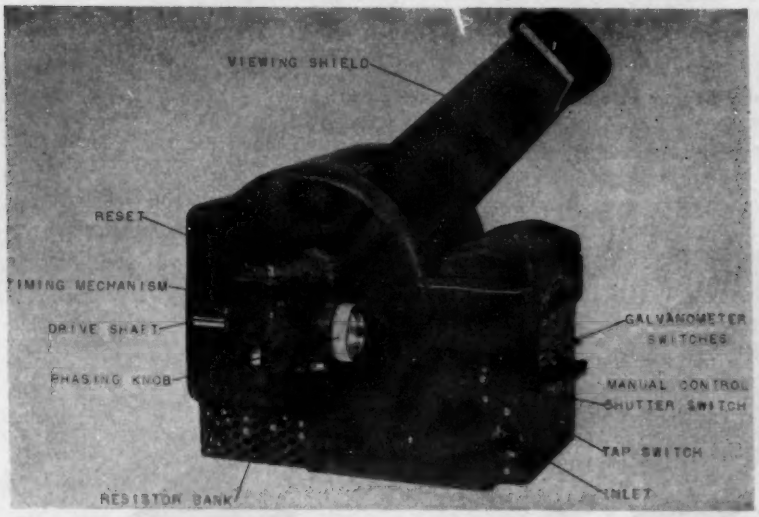
For calibrating under engine firing conditions, two types of balanced pressure indicators are suitable: the balanced valve and the balanced diaphragm. After considering the characteris-



■ Fig. 26 – Optical system of oscillograph



■ Fig. 27 – Completed oscillograph



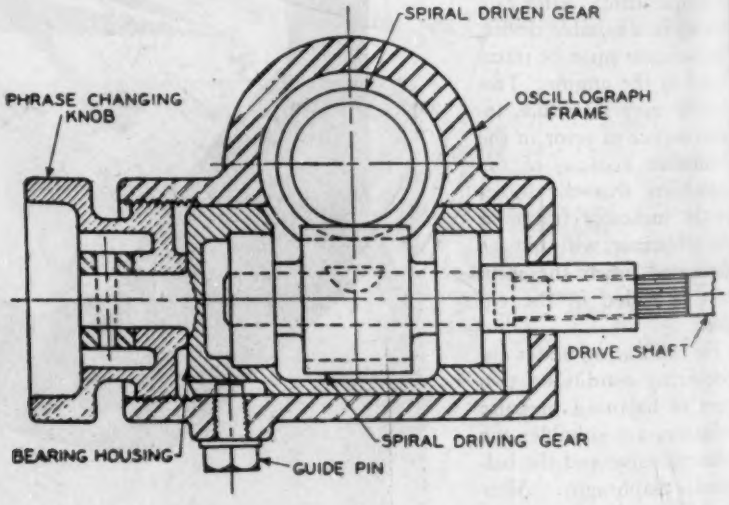
■ Fig. 28 – Another view of the oscillograph

tics of these two indicators, the balanced diaphragm was chosen as the more suitable. All balanced diaphragm indicators are similar to, or modifications of, the indicator developed by the Bureau of Standards.

Balanced diaphragm indicators, in general, can be divided into two groups, namely, high- and low-pressure types. Figs. 33 and 34 show the high-pressure, balanced-diaphragm indicator designed and built by General Motors Research Laboratories to include water cooling, small turning radius, and improved seat and contact structure.

The diaphragm of the high-pressure indicator must be heavy enough (0.002-0.007 in.) to withstand the pressure encountered in the combustion chamber of an engine. The pressure necessary to move the diaphragm away from the contact (1 to 3 lb) usually is small compared with cylinder pressures.

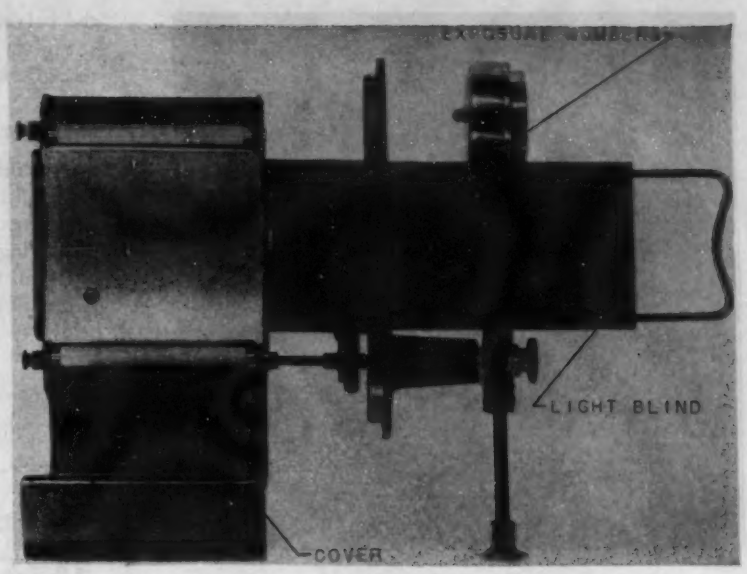
For use in manifolds of engines where maximum pressures are 1 to 10 lb, it has been necessary to build indicators with extremely thin diaphragms (0.0005 in.) in order to decrease the differential pressure necessary to open the contact. A specially designed indicator had to be built for use with these thin diaphragms to prevent wrinkling, buckling, twisting, sticking, and diaphragm rupture. See Figs. 35 and 36. The seat and diaphragm are assembled as a separate unit before installation in the indicator frame. See Fig. 35. The thin diaphragm, 0.0005 in., is placed between the two clamping rings and held in place by six countersunk screws as shown in Fig. 37. This method of clamping prevents twisting of the diaphragm. The diaphragm is then stretched by screwing the bottom seat in place. The upper seat is then screwed against the top of the diaphragm, forming a seal where the bottom stretching seat makes



■ Fig. 29 – Phase-changing device

contact with the diaphragm. See Fig. 37. Both the upper and lower seats contain many small holes for passage of the gas to the diaphragm. The clearance between the upper and lower seats is very small (0.002 in.) in order to limit the maximum movement of the diaphragm.

In Fig. 35, the seat assembly is shown mounted in the indicator frame. After assembly, the insulated adjustable electrode is lowered until it makes contact with the diaphragm. Caution should be exercised when making this adjustment to avoid rupture of the thin diaphragm. The recommended method of adjustment is to close a high-impedance electrical circuit (such as a neon light and resistor) when the insulated electrode contacts the diaphragm. If this adjustment is properly made, the pressure necessary to break the electrical circuit should not exceed 2 in. of water.

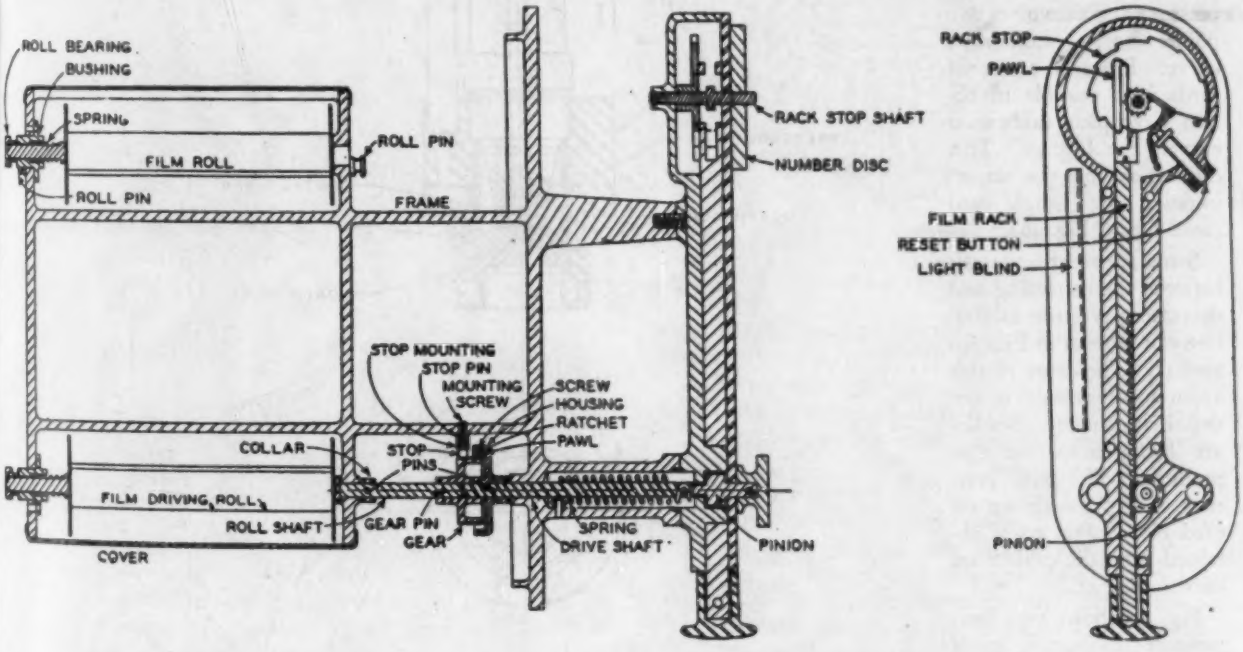


■ Fig. 30 – Camera film housing

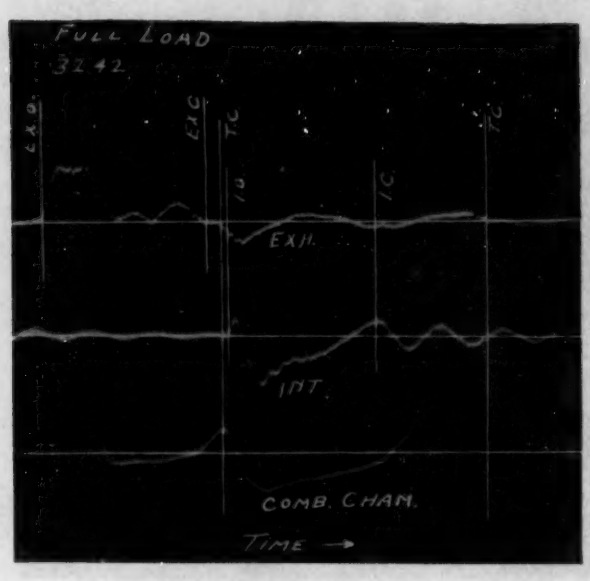
The electrical circuit used to indicate the "make" and "break" of the balanced diaphragm is shown in Fig. 38. The neon light is used to give a visual indication of the balanced-diaphragm operation. The resistor is used to limit the current in the circuit and may be adjusted to any desired value.

A single-card indicator diagram with superimposed calibration marks is shown in Fig. 39. (These marks appear as breaks in the trace of this particular indicator diagram.) The distortion produced on the single-card indicator diagram can be eliminated by using the multielement oscillograph for recording individually and simultaneously the indicator diagram and the breaks produced by the diaphragm indicator.

Fig. 40 shows two of a series of calibrated indicator cards taken with the multielement oscillograph. The method of calibration is as follows: Referring to card No. 6, in Fig. 40, the balanced diaphragm makes contact at *A* when the engine pressure exceeds the known balancing pressure (200 psi) above the diaphragm. This contact remains closed until the engine pressure drops below the balancing pressure. This is shown at point *B* on card No. 6. Since the



■ Fig. 31 – Film indexing mechanism



■ Fig. 32 – Typical card taken with multielement oscillograph

time axes of both diagrams are identical, points A and B may be perpendicularly projected to the points A_1 and B_1 on the indicator diagrams. The vertical displacement of points A_1 and B_1 is measured from the atmospheric datum line. This displacement is produced by a known balancing pressure and thus gives two points on a calibration curve. From a series of cards it is possible to obtain a complete calibration curve. See Fig. 41. This calibration may be superimposed on a single card (card No. 7, Fig. 40).

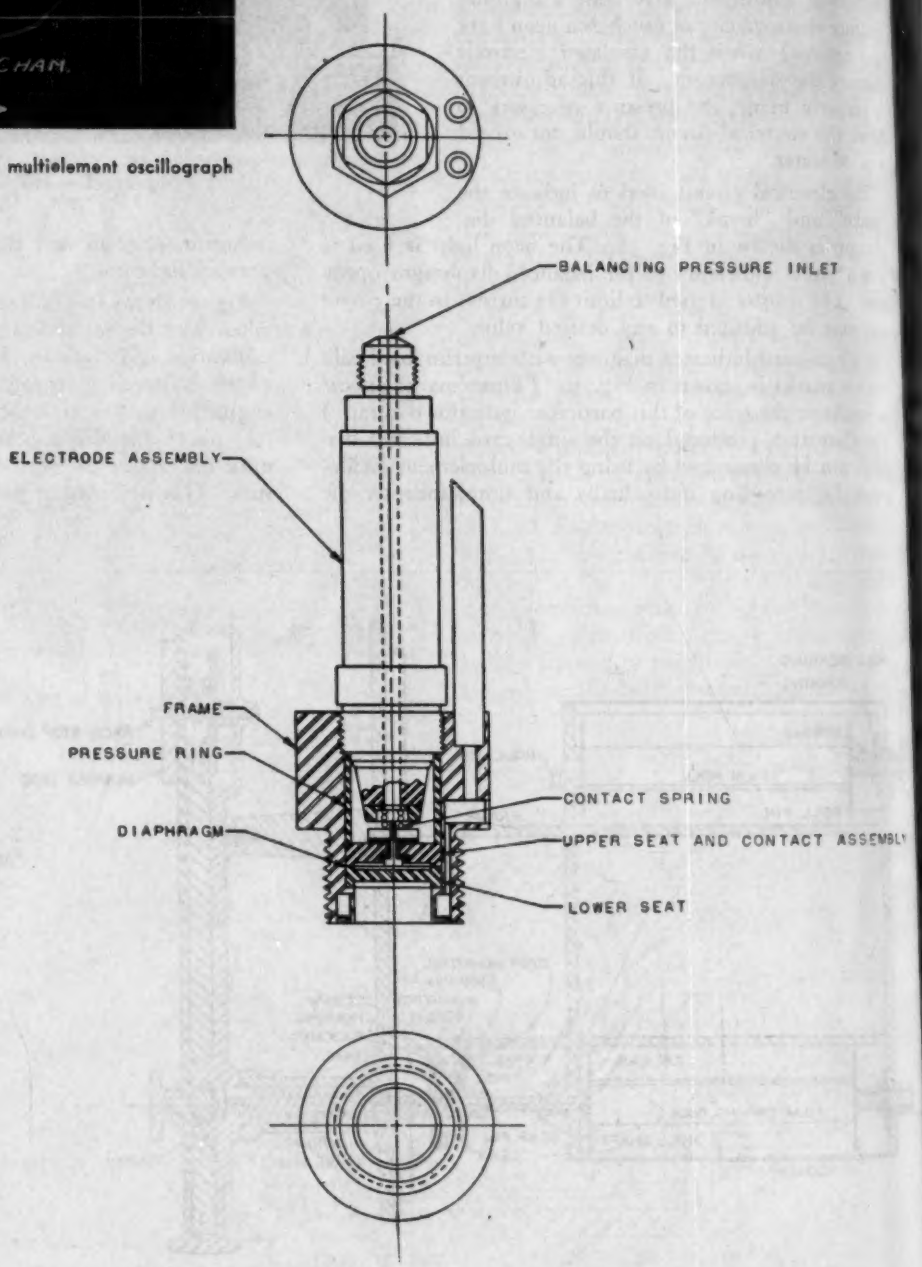
Some apparent hysteresis between the increasing and decreasing pressure calibration can be seen in Figs. 40 and 41. The cause of this apparent hysteresis is not definitely known. Studies are being made on this problem. The peak pressure of 535 psi shown on card No. 7, Fig. 40, is obtained from the calibration curve, Fig. 41.

Fig. 42 shows two low-pressure calibrated cards taken in the manifold of an engine. These cards clearly show the ability of the balanced diaphragm to follow low-pressure waves.

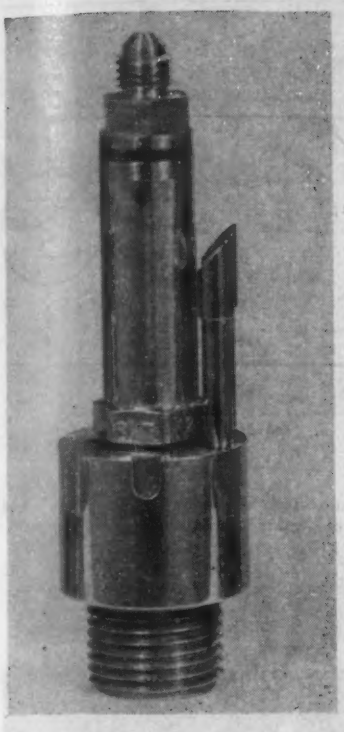
Fig. 43 is a calibrated high-pressure combustion-chamber record taken on an aircraft engine running at 2600 rpm.

■ Other Applications for Indicator

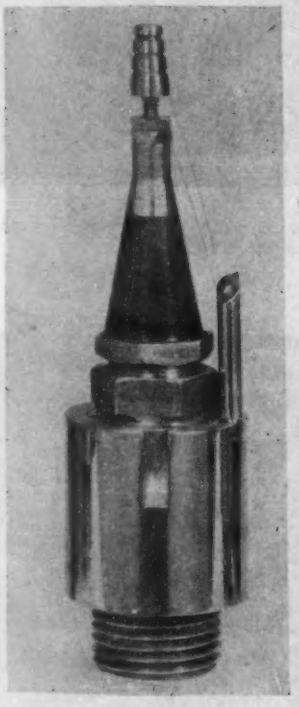
The principle of the condenser indicator and its associated electrical equipment is not limited to pressure measurements. The technique and equipment developed for measuring pressure can be applied to studying the amplitude and wave form of small deflections of moving bodies. A few of the applications in the automotive field are listed below:



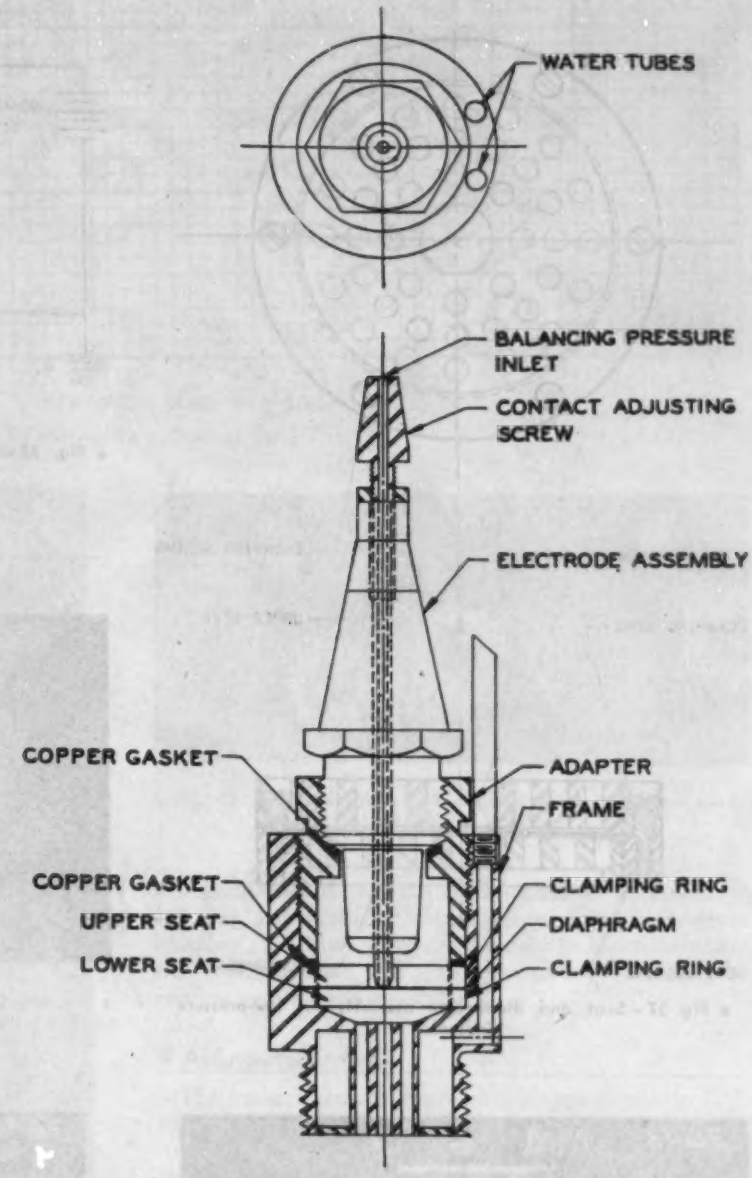
■ Fig. 33 – Diagram of high-pressure, balanced-diaphragm indicator



■ Fig. 34 – High-pressure, balanced-diaphragm indicator



■ Fig. 36 (left) – Low-pressure, balanced-diaphragm indicator



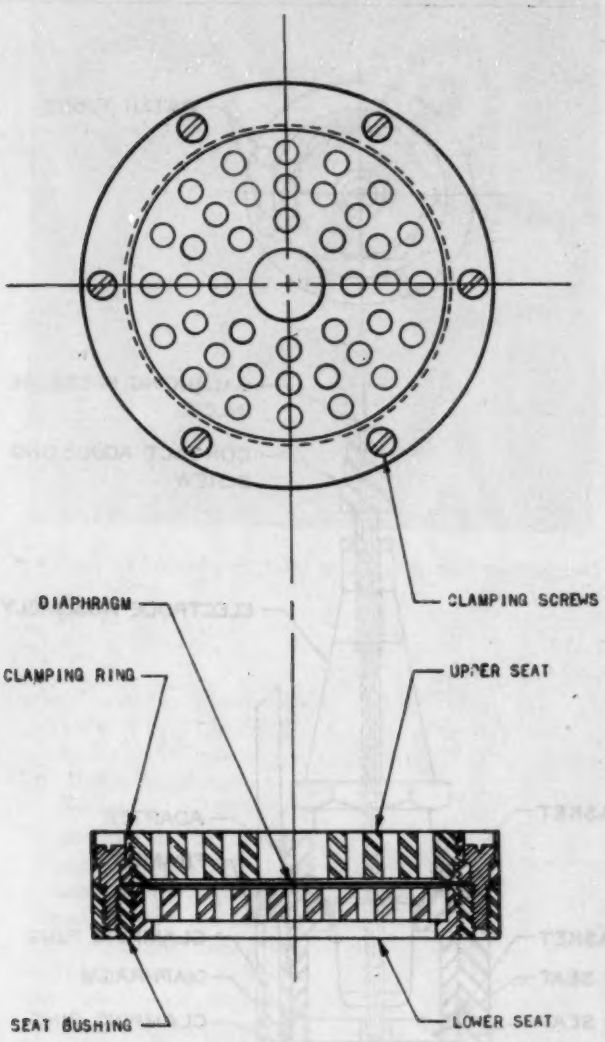
■ Fig. 35 – Diagram of low-pressure, balanced-diaphragm indicator

Measurement of stress and strain.
Measurement of flywheel deflection.
Measurement of cylinder-head and block deflection.
Obtaining valve-lift diagrams.
Use in propeller-shaft balancing machines.

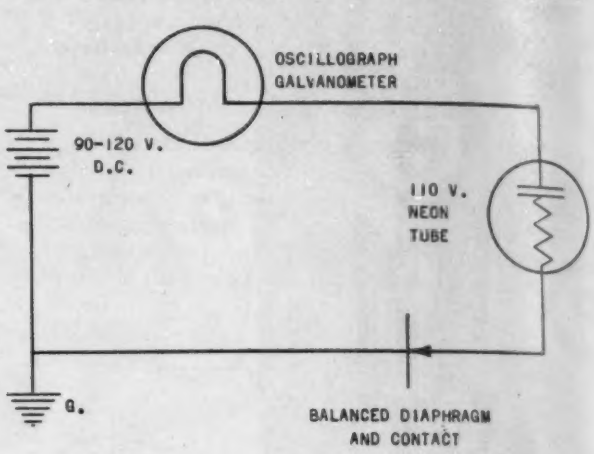
This type of instrumentation might have the following possible uses in other engineering fields:

1. Measurement or indication of unbalance in high-speed rotating machinery.
2. Vibration measurement.
3. Pressure surges in oil-circuit breakers.
4. Measurement of water hammer and other hydraulic surges.
5. Gun pressures.
6. Jet propulsion pressure studies.

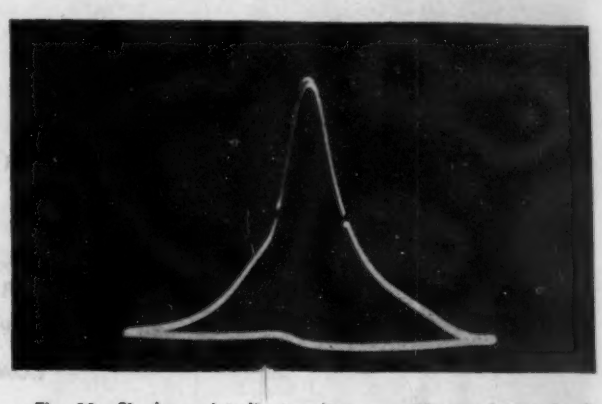
For certain types of detonation studies the indicator equipment has been used in conjunction with a General Radio type 760A wave analyzer to amplify the detonation frequency components of the indicator card. The output from the oscillator-detector is fed through the wave analyzer which has been tuned to the detonation frequency. The output from the wave analyzer can then be viewed with a conventional cathode-ray oscillograph. A high-



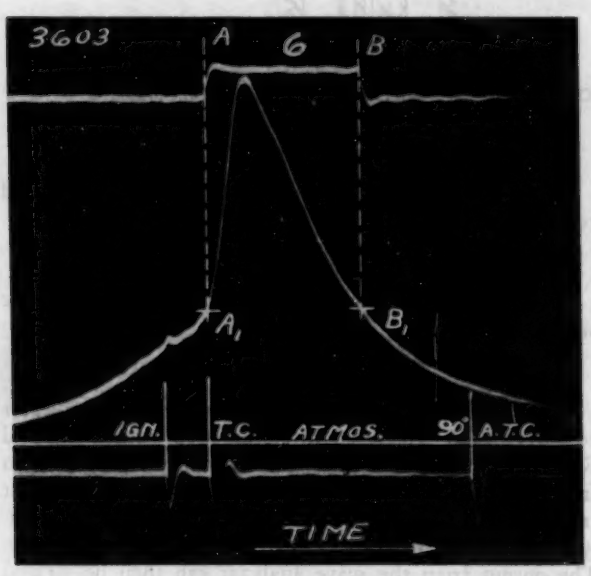
■ Fig. 37 – Seat and diaphragm assembly for low-pressure indicator

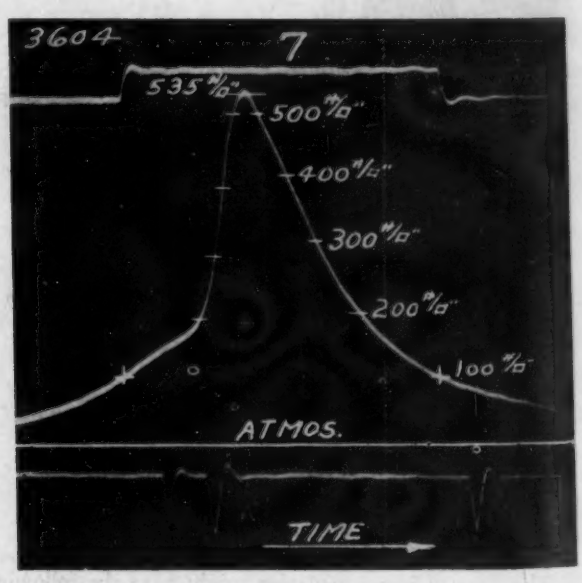


■ Fig. 38 – Electrical circuit used to indicate the "make" and "break" of the balanced diaphragm

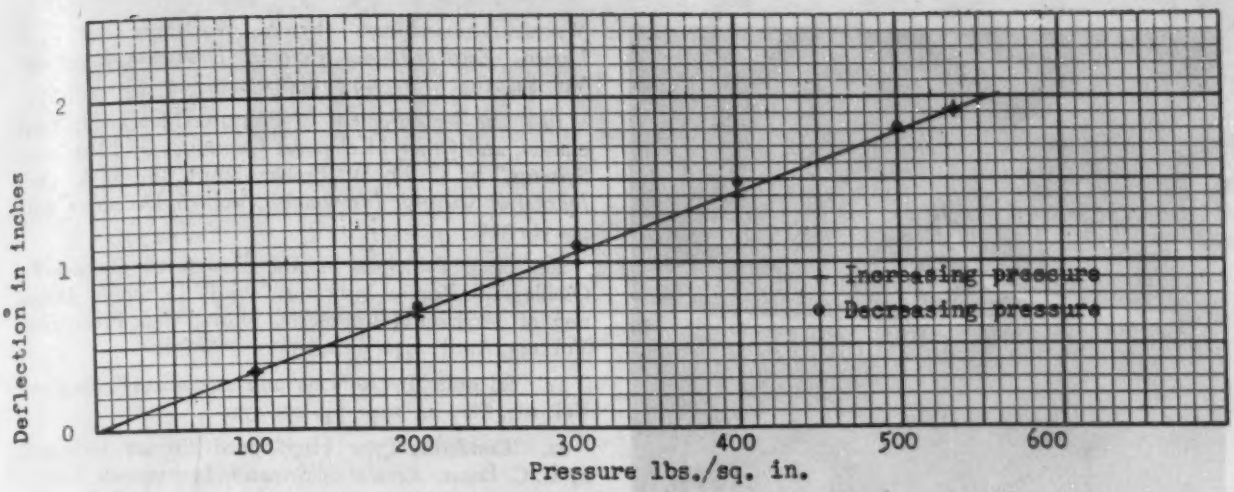


■ Fig. 39 – Single-card indicator diagram with superimposed calibration marks

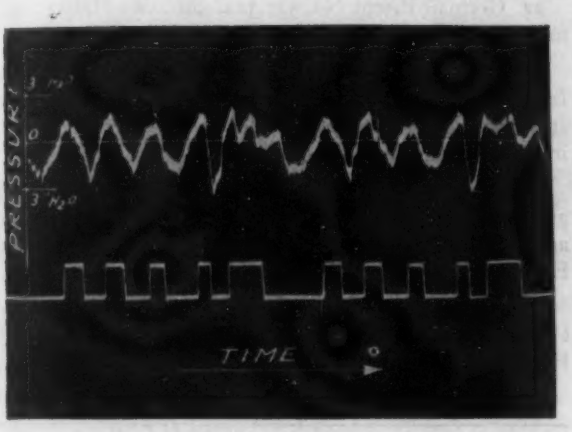


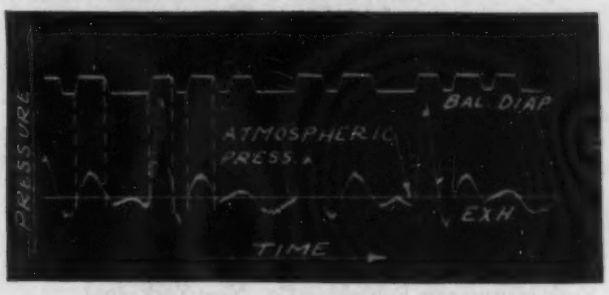


■ Fig. 40 – Two of a series of calibrated indicator cards taken with multielement oscillograph

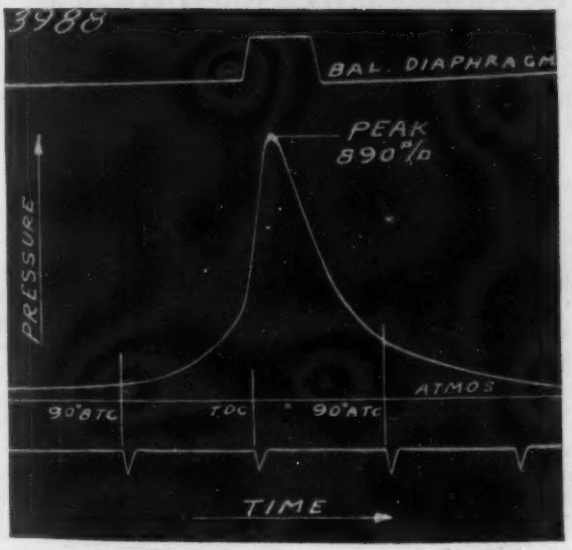


■ Fig. 41 – Complete calibration curve





■ Fig. 42 (left and above) – Two low-pressure calibrated cards taken in engine manifold



■ Fig. 43 – Calibrated high-pressure combustion-chamber record

pressure, combustion-chamber record taken under detonating conditions is shown in Fig. 44. The detonation frequency components of the indicator card under these conditions are amplified approximately 30 times with respect to the lower frequency components of the indicator card itself.

It is understood that the development of the condenser indicator and associated electrical equipment is far from complete. Much work is yet to be done to make it entirely free from vibration and temperature effects that may be encountered in some applications.

■ Acknowledgment

The authors wish to express their appreciation to E. J. Martin and G. M. Rassweiler for their interest and helpful suggestions in the preparation of this paper.

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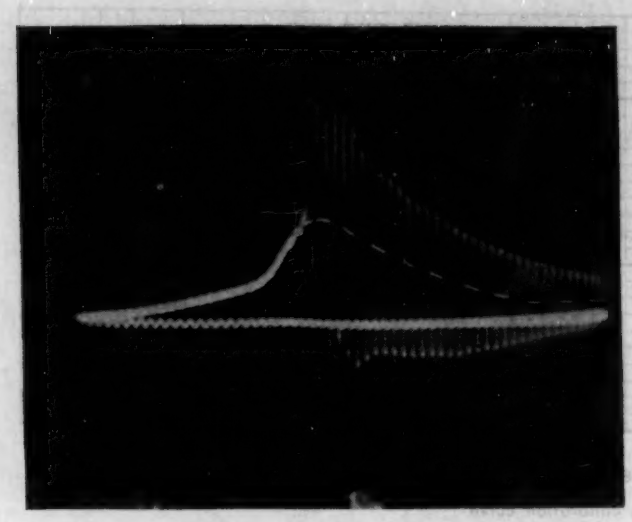

■ Fig. 44 – High-pressure, combustion-chamber record taken under detonating conditions

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Trucks to Come

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the driver holds the indicator at the right spot. When he releases the knob, it will return to the "normal" indication and the buzzer will continue to sound until he opens the ignition switch or remedies the trouble.

It might indicate all or any number of the following:

Low oil.
Low gasoline.
Low water.
Low battery electrolyte.
Low battery charge.
Low oil pressure.
Low air pressure.
Low water temperature.
High water temperature.
High oil temperature.
High speed (over a set limit).
High engine speed (as in coasting in gear).

The INTERCOOLING PROBLEM in AIRPLANE DESIGN

by **HOLLEY B. DICKINSON**
Lockheed Aircraft Corp.

FROM the standpoint of the airframe designer, the purpose of charge-air cooling is to increase the effectiveness of the airplane in accomplishing the operations for which it is designed. This is the ultimate yardstick which must be applied in answering three fundamental questions:

1. Is charge-air cooling to be employed?
2. How is cooling to be accomplished?
3. To what extent is the charge air to be cooled?

It is immediately apparent that these are very complex questions. They involve not only the engine performance, but also the aerodynamic effect of the weight and of the internal and external drag of the coolers. They involve questions of maintenance, accessibility, and servicing. Often, production, tooling, availability of purchased equipment, and cost enter. Certainly, if time were taken to subject all of these factors to careful analysis in a particular case, the design would be obsolete before it flew. A great deal must be left to the individual judgment.

It is not the intention to attempt an original contribution to the problem but to describe and superficially, at least, to analyze current practice. The author does not propose to dismiss any of these problems on the basis of the qualitative discussion and brief quantitative data included, but rather merely to express what must sometimes seem to be the obstinate viewpoint of the designer.

When is Charge-Air Cooling Desirable?

The first problem which confronts the airframe designer is, whether charge-air cooling is to be employed. At first glance, this may seem an almost trivial question, and indeed in most cases the answer is obvious. For example, an installation in a fighter airplane designed for maximum power operation at 30,000 ft may employ an engine which requires a pressure at the carburetor of 35 in. of mercury and, to avoid detonation, a temperature at that point of not more than 100 F. Since the auxiliary-stage supercharger must raise the induction system pressure from about 9 in. to about 37 in. of mercury, the carburetor air temperature without intercooling would be about 300 F under standard atmospheric conditions.[1] Even if detonation did not prohibit such temperatures, the loss in thermal efficiency of the engine and the loss in efficiency of the engine-stage supercharger would do so. In such a case, the only alternative to some form of charge-air cooling is severely reduced power output.

On the other hand, consider a transport designed to cruise at 25,000 ft and employing both auxiliary- and engine-stage supercharging to attain this critical altitude. If a pressure out of the auxiliary-stage supercharger of as much as 23 in. of mercury is required for, say, the power to be used in climbing to 25,000 ft, the resulting carburetor temperature will be about 140 F compared with an outside air temperature of −30 F. Now, this temperature might easily be below the detonation limit at the climb power, so that intercooling could be omitted without reduction of manifold pressure. If intercooling were used, an initial temperature difference between engine and cooling air of 170 F is available, so an engine air temperature drop of

A DISCUSSION of the aircraft-engine charge-air cooling problem is given here from the point of view of the airframe designer.

Many of the factors which enter into the selection of the type and capacity of charge-air cooling installations are difficult to subject to quantitative analysis. Among these are problems of simplicity, maintenance, accessibility, and arrangement of the engine installation. In general, however, the difficulty of installation of the charge-air system is out of proportion to the core size. The weight increase due to the system, for instance, is in the neighborhood of five times the weight of the air-to-air intercooler core.

The use of a liquid-to-air aftercooler system as opposed to an air-to-air intercooler system shows promise only in applications with liquid-cooled engines where the separation of the ultimate cooling unit from the rest of the installation is a definite arrangement advantage.

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[1] Calculated from $T_2 = T_1 + \frac{T_1}{\eta_p}\left[(p_2/p_1)^{0.288} - 1\right]$

where T_2, p_2, and T_1, p_1 are, respectively, the absolute temperature and pressure out of and into the auxiliary-stage supercharger. $\eta_p = 0.60$. Dynamic heating neglected. (See Footnote 2.)

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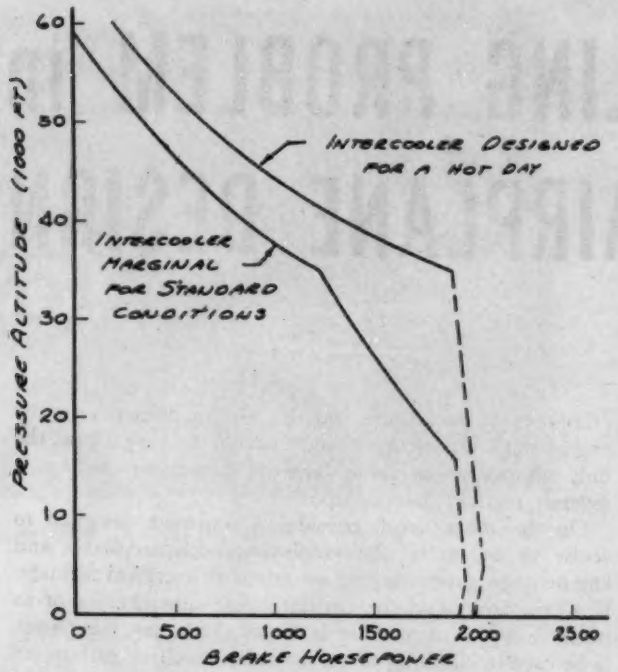
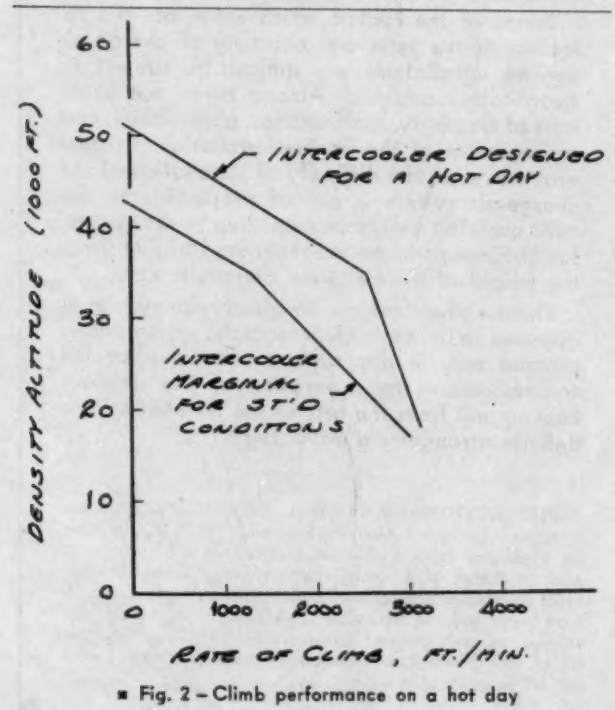


■ Fig. 1 – Power available on a hot day

70 deg could readily be obtained (40% intercooler "effectiveness"). This would produce a gross gain in engine power of the order of 7%. Even though the net gain would be somewhat less because of the weight and drag of the intercooler, the improvement in rate of climb at this altitude would be substantial.

Consider, however, the cruising condition. A carburetor pressure of 18 in. of mercury would probably produce the cruising power desired. Without intercooling, the carburetor air temperature would be only 80 F compared with a detonation temperature of perhaps 150 F. Intercooling could reduce the 80 F temperature to 40 F or less, but this temperature reduction is certainly unnecessary and would be of no advantage to the transport unless it resulted in reduced specific fuel consumption, since the additional power is not needed.



■ Fig. 2 – Climb performance on a hot day

With the fuels of reduced volatility now available, difficulty with poor vaporization is likely to be encountered at low mixture temperatures. The trouble takes the form of condensation of the fuel in the manifold and on spark plugs, cylinder walls, and so on, resulting in excessive leanness, low effective antiknock, and fouling of the plugs. There is, accordingly, a trend towards the use of higher carburetor air temperatures. Under these conditions it is doubtful whether lower fuel consumptions would result from lower carburetor air temperatures attainable with intercooling, even if such temperatures were usable in cruising. When the drag and weight of the intercooler installation is considered, it is apparent that such an installation may well be detrimental at cruising power.

Such being the case, the question arises as to whether it is worth while in transport operation to risk reduced efficiency in many hours of cruising in order to have available more rate of climb at altitude.

It will be well to defer a specific answer to the question of when to employ charge-air cooling with the conclusion merely that high carburetor air temperatures in some flight conditions do not necessarily mean that charge-air cooling is desirable.

■ How Much Charge-Air Cooling is Desirable?

The problem of the extent to which the charge-air is to be cooled is one closely related to that just considered.

There would immediately appear to be two limits to the cooling of the charge, which would aid in core size selection. The minimum amount of cooling is that which will provide a carburetor air temperature just low enough to avoid detonation under the most critical condition. The maximum possible amount of cooling provides a carburetor air temperature equal to the atmospheric temperature increased by dynamic heating. Neither limit, unfortunately, is immediately useful. The maximum condition requires infinite intercooler size, and the minimum depends entirely upon the critical condition selected. Consequently, the selection of the most critical condition must be examined.

In the case of fighter airplanes, there is little question that the power used should be the maximum permissible from the engine, and the altitude should be the maximum obtainable with the supercharger. Whether the atmospheric temperature used should correspond to the hottest day likely to be encountered or to an average day is debatable, but with military airplanes the decision is out of the hands of the airframe designer, since it is specified by the procuring agency. Even if this were not the case, the loss in performance on a hot day would be so severe with an intercooler just large enough for a standard day that the larger cooler would probably always be justified for high-altitude fighters. This is illustrated in Figs. 1 and 2.

Fig. 1, taken from the reference given in Footnote 2, can be interpreted as the difference in power available on a hot day between an intercooler installation designed for a hot

[2] See *Journal of the Aeronautical Sciences*, Vol. 10, October, 1943, pp. 261-269: "Temperature Effects on Turbine Supercharger Installations," by Philip Colman.

day and one designed marginally for a standard day. (A "hot day" is 40 F above NACA standard temperatures.)

The effect of this loss in power on rate of climb is shown in Fig. 2. The airplane with a "standard-day intercooler" loses about 1000-fpm rate of climb on a hot day compared with an airplane with a "hot-day intercooler." This results in about 8000-ft difference in ceiling. Less spectacular losses in speed at altitude would, of course, also be suffered, in return for which there would be some gain under the more typical standard conditions.

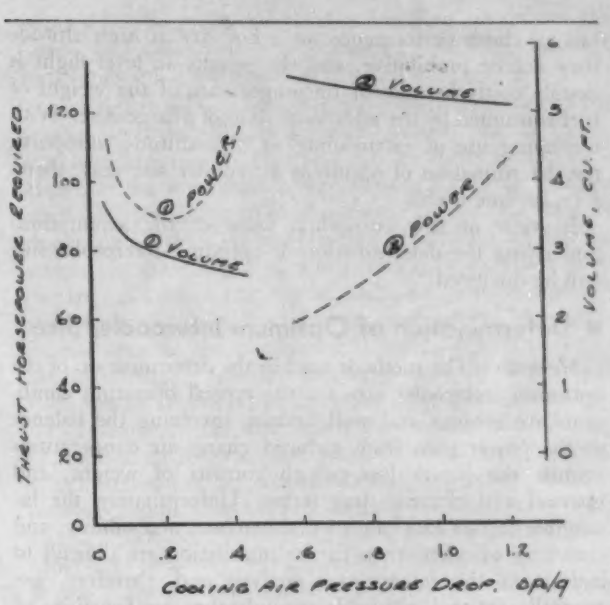
On the other hand, the decision is far less obvious in the case of commercial airplanes and their military counterparts – transports and bombers. With such ships there is not only the question of atmospheric conditions, but also the question of what power and altitude to use. As we saw earlier, it probably will not be desirable to cool maximum power, but only a typical power condition.

Now, considerable attention has been devoted to the determination, for a particular flight condition, of what is called the "optimum" intercooler size, that is, the size giving best airplane performance. Consequently, this determination is now a straight-forward problem which proceeds along well-established lines, under assumptions which will be discussed later. The important point is that the optimum intercooler is often purely of academic interest; the size of intercooler which results in the optimum airplane is generally the smallest size which will squeeze by the most critical *necessary* design condition. This statement assumes, of course, that there is a most critical necessary condition. Stated differently, the optimum intercooler for the typical operating condition would be nice except that it is too small for the most critical necessary condition. Furthermore, the large core required for the most critical condition is enough of a penalty in the typical operating condition without the further penalty which would be imposed by carrying a still larger cooler which would be optimum for the critical condition.

Figs. 3 and 4 illustrate (but do not prove) these contentions for the case of a high-altitude fighter. The variation with core volume of total power consumption of the intercooler installation is shown on the left of Fig. 3 as a function of the cooling air pressure drop across the core necessary at high speed. It is to be noted that there is a volume (about 3 cu ft) which is optimum, that is, which gives minimum power required at high speed.

Considering the curves on the right of Fig. 3, however, it is apparent that a much larger intercooler volume is required to meet the hot-day climb condition. The volume is not very sensitive to pressure drop in this condition, and the designer might be misled into using a core larger than necessary with the cooling air pressure drop available in climb. (The maximum cooling air pressure drop will normally run between 1.0 and 1.2 times the dynamic pressure.)

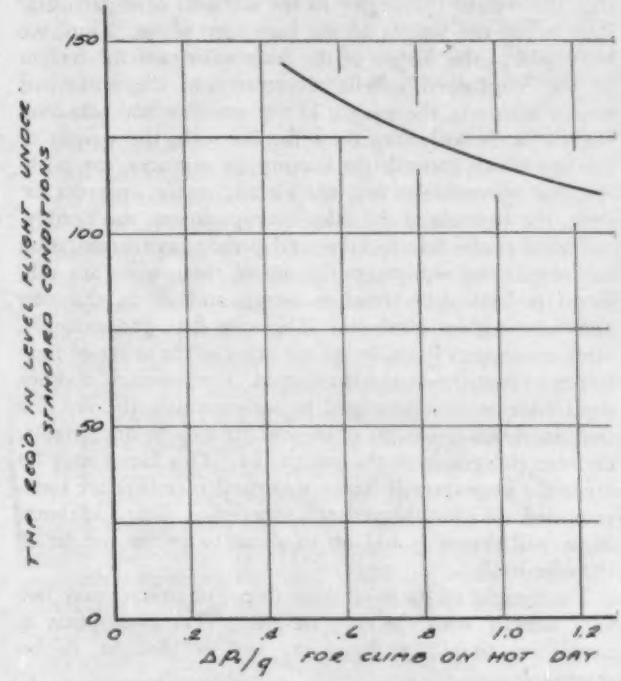
While such a procedure would save power in climb, the effect upon operation in level flight under standard conditions is adverse, as shown in Fig. 4. From this curve it is apparent that the best core for the typical operating condition is the smallest that will meet the critical climb condition. Whereas the optimum core for level flight under standard conditions would have a volume of only 3 cu ft and would consume only 90 thp at high speed, the critical climb condition requires a core of at least 5 cu ft volume, which will consume 110 thp at high speed. If a still larger core is used which requires only 60% dynamic



■ Fig. 3 – Intercooler volume required and power consumption
(1) High-speed level flight – standard conditions
(2) Climb – hot day

pressure drop in climb, the volume increases to 5.5 cu ft and the power consumption at high speed to 135 thp. There may be applications in which the 5.5 cu ft core would be preferable to the 5 cu ft core because it would save about 40 thp in climb on a hot day (considerably less on a standard day), but such cases would not be typical.

In contrast, with transport aircraft the designer may ignore the severe climb condition and select the intercooler to be optimum for a typical level-flight condition. The



■ Fig. 4 – Effect of cooling air pressure drop on power required in level flight under standard conditions

loss in climb performance on a hot day at high altitude may not be prohibitive, and the penalty in level flight is doubly costly because of the importance of the weight of fuel consumed by the additional drag of a large core. With increasing use of intercooling in high-altitude transports, the determination of optimum intercooler size may, therefore, become useful.

In view of this possibility, some of the assumptions underlying the determination of optimum intercooler size will be discussed.

■ Determination of Optimum Intercooler Size

Methods – The methods used in the determination of the optimum intercooler size for the typical operating conditions are obvious and well known, involving the balance of the power gain from reduced charge-air temperatures against the power loss, which consists of weight, and internal and external drag terms. Unfortunately, the intangible factors like ease of maintenance, accessibility, and crowding of other items in the installation are difficult to include in the quantitative analysis and, therefore, are generally omitted, although certainly they are functions of core size, and important factors. As a result, the designer must lean heavily upon experience and judgment in evaluating his answer, and cannot blindly employ the optimum size which results from the analysis.

Carburetor Air Temperature – With the limits set by detonation and the vaporization difficulties mentioned, the charge-air temperature should be at the disposal of the intercooler designer, and considered as one of the primary variables instead of being limited to some arbitrary figure, such as 90 F, at the carburetor. Usually the optimum cooler will result in a temperature above that figure rather than below.

Weight – Another important assumption is the weight chargeable to the intercooler. The light weight of air-to-air intercoolers is deceptive, and it cannot be repeated too often that the weight chargeable to the addition of a particular item is not the weight of the bare item alone. Thus, we must add to the weight of the intercooler core the weight of the lengthened nacelle structure and the additional engine air ducts, the weight of the entrance and exit cooling air ducts, including their flexible seals, the weight of the flap which controls the cooling air exit area, the push-rod that operates the flap, the electric motor or hydraulic strut, the controls to the pilot's compartment, the flexibly-mounted cradle for the core, and perhaps instrumentation and regulating equipment. Some of these parts are subjected to high differential pressures, and all to vibration and temperature extremes. The exit flap, for example, often encounters limit design air loads of the order of 1000 lb per sq ft at the design dive speed. Furthermore, if other units have to be rearranged to accommodate the installation, increases in weight of these units may be appreciable, and are chargeable to the intercooler. This factor may be especially important if heavy structural members are compromised. Within the author's experience, these additional items will generally add up to about twice the weight of the core itself.

The weight of some of these items, of course, may not vary directly with the core weight. That assumption is made for simplicity, however, and is thought to be reasonable.

Beyond that, every pound of additional weight in the airplane requires additional gas if the range is to be maintained without loss of useful load. Depending upon the airplane and upon the range, Fig. 5 shows that from 0.2 to 0.8 lb of fuel and oil will be required per pound added to the weight empty.

Fig. 5 is obtained from the relationship:

$$\frac{\Delta \text{ Fuel Weight}}{\Delta \text{ Weight Empty}} = \frac{\text{Take-Off Gross Weight}}{\text{Take-Off Gross Less Fuel}} - 1$$

for constant range, which is derived from the Breguet range formula, as follows:

$$\text{Range} = K \frac{\eta}{C} \frac{L}{D} \log \frac{W_{TO}}{W_1}$$

where

W_{TO} = Take-off weight

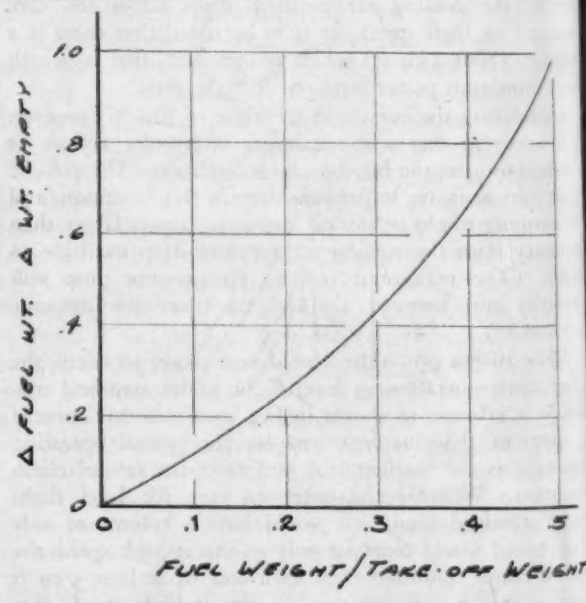
W_1 = Take-off weight less fue

Consider an increase of weight empty, ΔW_E. This will require additional fuel, ΔW_F. Then, for the same range at the new take-off weight, $W_{TO} + \Delta W_E + \Delta W_F$

$$\frac{W_{TO}}{W_1} = \frac{W_{TO} + \Delta W_E + \Delta W_F}{W_1 + \Delta W_E}$$

$$\text{Whence } \frac{\Delta W_F}{\Delta W_E} = \frac{W_{TO}}{W_1} - 1$$

Nor does the penalty end here. Additional tanks must be furnished to carry the additional fuel, and, in the case of military airplanes, bullet-proofing at probably 0.2 lb per lb of fuel must be provided. Lastly, as these weights are added to the airplane, other items will probably be added also, and sooner or later, important components must be increased in size. The landing gear is one such item which often becomes critical, so that it frequently occurs that a substantial weight increase in the landing gear must be charged off to a number of relatively minor increases in the weight empty of the airplane. Further, it is often necessary to maintain the wing loading constant to retain altitude maneuverability, or landing or take-off speed. In this case, wing area at perhaps 4 lb per sq ft must be increased. With a 40 lb per sq ft wing loading, this means another 0.1 lb for each pound added, including fuel. All



■ Fig. 5 – Additional fuel requirement for constant range

this weight in turn requires further increases in the fuel, so that the final increase in weight due to the intercooler could be expressed in the form of an infinite series which does not converge too rapidly.

The above items can be totaled roughly as follows:

Unit core weight	1
Structure, ducts, etc.	2
Fuel = 0.4 × 3	1.2
Tanks and bullet-proofing = 0.25 × 1.2	0.3
Wing area = 0.1 × 4.5	0.5
Landing gear, further fuel, etc. – neglected	
Total, lb	5.0

Briefly, it would be unduly optimistic to assess the weight of an intercooler installation at less than 5 lb per lb of core.

External Drag – In connection with the external drag of the intercooler installation, one point is especially worth emphasizing. The system of ducts which must be provided with the cross-flow air-to-air intercooler makes this unit one of the most awkward to install in the entire powerplant. A comparison of typical high- and low-altitude installations reveals that the difference in simplicity, accessibility, visibility for the pilot, and frontal area are all out of proportion to the size of the intercooler itself. In calculating the optimum intercooler size, these factors, with the possible exception of frontal area, are generally omitted, not because they are unimportant, but because they are difficult to express quantitatively. It is easy to underestimate the increase in frontal area due to the intercooler, moreover, because part of it is in the form of frontal area of other components awkwardly located because of the presence of the intercooler.

In the most fortunate cases, the increase in frontal area due to the intercooler may be limited to little more than the duct entrance area required for the cooling air. This area will be that obtained by dividing the required cooling airflow by about 0.9 of the velocity ahead of the entrance in the climb condition. If the entrance is located in the slipstream, 0.9 of the slipstream velocity can be used. Thus:

A_e = Entrance area

A_c = Area of cooling air face of core

K = Core conductivity, cooling air

$= (Q/A_c) \;/\; \sqrt{2\Delta P/\rho}$

ΔP = Available pressure drop across core

Assume:

$\Delta P/q = 1$

$q = \frac{\rho}{2} V^2$

V = Velocity ahead of the entrance

$K = (Q/A_c) \;/\; V$

$A_e = Q/0.9V = K A_c/0.9$

If $K = 0.2$ is typical,

$A_e = 0.22\, A_c$

so that the entrance area is about 0.2 of the core area. The minimum drag increment is obtained, not with this increase in area, but by using a somewhat larger increase which permits better streamlining. Even this amount is often insufficient to accommodate the core. In some cases, a wing-leading-edge inlet saves frontal area. Within the author's experience, however, this solution entails larger internal losses, great difficulty in avoiding maximum lift losses, and profile drag increases on the wing that are sometimes small but rarely negligible.

The idea that the intercooler can simply be placed in the fuselage or nacelle in a convenient vacant space is, to put it generously, naive. This fact can be substantiated by a glance at the outstanding high-altitude fighters. In some cases, the frontal area of the nacelle or, in the case of single-engine ships, of the fuselage including all cooling scoops but excluding the pilot's canopy, approaches twice the frontal area of the engine itself, an increase which is due primarily to items like the intercooler. A typical frontal area for an aircooled installation with two stages of supercharging and intercooling is 24 sq ft. The engine itself has an area of 17 sq ft. In extreme cases, even larger ratios occur. In one case, an engine which could be concealed behind a 36-in. disc was used in a nacelle design which could not be concealed behind a 62-in. disc. In another instance, a 9 sq ft engine required a 15 sq ft nacelle. Yet, the "density" of these designs is remarkable. One could scarcely hang his hat inside the P-38 boom between the spinner and the prestone radiator. Almost certainly, then, an intercooler installation will add frontal area greater by a substantial amount than the cooling air entrance.

■ Type of Charge-Air Cooling

The airplane designer has the choice of two types of charge-air cooling which are called intercooling and aftercooling, depending upon whether the cooling unit is located between the stages of a two-stage supercharging system, or between the second stage and the intake ports. He also has the choice of accomplishing the cooling by means of air-to-air intercoolers or with liquid-to-air intercoolers. The air-to-air system is the conventional arrangement, and the liquid-to-air system has only recently received favorable consideration. The liquid-to-air system differs fundamentally from the air-to-air system in that two units are required, a primary unit for the transfer of the heat from the engine air to the liquid, and a secondary unit for the transfer of the heat from the liquid to the cooling air.

Since the second stage of supercharging must be integral with the engine in order to facilitate vaporization of the fuel and distribution between the cylinders, the aftercooler must be inserted in a confined region, so that it also is practically integral with the engine. The problems involved in ducting in this arrangement appear to be insurmountable in the case of the air-to-air aftercooler, so that only the liquid-to-air aftercooler needs to be considered.

Likewise, a liquid-to-air intercooling system has little in its favor. The temperature difference between metal and air in each of the two units is approximately the same as in the air-to-air cooler, so that the metal-to-air surface of each unit must be as great as the corresponding surface in the air-to-air core. Hence, the total volume of the two liquid-to-air units will exceed the volume of the air-to-air unit by the total liquid-filled core volume. Thus, the only feature which makes the liquid-to-air system usable is the higher temperature difference which is available with aftercooling.

Finally, then, we need only compare the liquid-to-air aftercooler with the air-to-air intercooler. This means that a discussion of intercooling versus aftercooling cannot be

separated from a discussion of air-to-air versus liquid-to-air systems.

The fundamental advantages of the two systems are immediately obvious. The aftercooler works with higher temperature differences due to the two stages of compression which precede it. The intercooler, on the other hand, makes possible higher blower efficiencies in the second stage of supercharging. As in many other airplane design problems, the deciding factors are not the obvious ones, and a decision as to which arrangement to use in a particular installation can best be reached by consideration of a number of small effects which require judgment rather than analysis to evaluate.

With the liquid-to-air aftercooler, one of the biggest advantages, from the standpoint of the airplane designer, is the flexibility in powerplant installation arrangement which it makes possible. The awkward arrangement of ducts in the cross-flow air-to-air intercooler and their large size make its location very critical and difficult to work out efficiently. It often requires a compromise in a great many other units of the powerplant installation. On the other hand, both units of the liquid-to-air aftercooler system are relatively compact, and require only one pair of air ducts in addition to the relatively small liquid-coolant lines. Furthermore, the secondary unit of the aftercooler system can be removed from the engine compartment altogether and placed in any suitable location in the airplane without concern over the problem of ducting the engine air to the unit and back again. This makes possible its location in thin wings where a cross-flow unit would be quite impractical, or in remote portions of the fuselage or nacelle. This flexibility is especially advantageous in the case of the two-stage geared blower because the auxiliary stage of the blower can be located close to the engine without having to leave room for ducts to a large air-to-air intercooler.

In the case of the turbosupercharger, the auxiliary blower is generally located far enough from the engine so that space can be found for an air-to-air intercooler between the blower and the engine without prohibitive dislocation of the other elements of the installation. The crowding problem of the primary unit of the aftercooler, however, is severe enough so that it is doubtful whether maintenance problems would be alleviated by the use of this arrangement in even a majority of cases. Certainly the weight of the liquid-to-air system will be greater. Likewise, in the upright liquid-cooled engine, the location of the primary aftercooler unit may be such as to interfere seriously with pilot visibility. Another point is that the higher carburetor temperatures encountered with the aftercooler system may prove a very real advantage in improving the vaporization of fuel under adverse conditions. Finally, the duct losses involved in the turns into and out of the air-to-air intercooler subtract from the critical altitude of this type of installation. These losses probably can be avoided in the aftercooler arrangement.

Taking into account the various factors, it is seen that the two systems are comparable from the airplane designer's standpoint. It is doubtful, however, whether it would ever be desirable to suffer the complication and weight caused by the installation of a complete prestone system on an aircooled engine in which such a system was not already in existence for engine cooling. This is especially true since the shape of the aircooled engine tends to create more space behind the engine for the installation of the air-to-air intercooler. Thus, in general, the increase in frontal area chargeable to an air-to-air intercooler would probably be less in the case of an aircooled engine than in the case of an equivalent liquid-cooled engine. On the other hand, the liquid-to-air system requires relatively little additional trouble in a liquid-cooled engine, and its greater compactness and flexibility are of more importance in such installations.

In large airplanes, generally, powerplant interchangeability and maintenance dictate a trend towards concentration of the powerplant in the nacelle. Also, this field has thus far at least been dominated by the aircooled engine. Consequently, the use of liquid-to-air intercooling does not appear probable in such installations.

On the other hand, on smaller airplanes, the decentralization of the powerplant may be extremely desirable for improved aerodynamic efficiency, and the possibility of isolating the secondary cooling in a separate unit located at a distance from the nacelle and probably combined with the prestone radiator may be a big advantage. Therefore, in small airplanes with liquid-cooled engine installations, the liquid-to-air aftercooler appears to offer definite advantages to the airplane designer.

■ Installation Problems

The Crowding Problem – The primary installation problem is, of course, the location of the unit. The air-to-air intercooler is the most difficult of all engine accessories to locate satisfactorily because it is necessary to provide four air ducts to it as against two for the prestone and oil radiators; all four of these are of about the same relative size as the air ducts approaching and leaving the prestone radiator; the charge air and cooling air both must generally travel in a fore-and-aft direction, so that one system must be compromised to the extent of two 90-deg turns in order to conduct the air laterally or vertically through the core. This problem could be avoided to some extent in a counter-flow instead of a cross-flow type of intercooler. A discussion of the relative merits of these types will be left to those better qualified; it is enough for our purposes that a good counter-flow cooler is not now available commercially, and that a good solution to the header problem taxes the imagination.

Considering, then, only the cross-flow intercooler, the frontal area of the core and ducts will always be greater than the core face even if space limitations permit tilting the core. This latter device reduces the overall height or width and reduces the angles through which the crossing air must be turned, but if it crowds adjacent items or compromises structure, the gain is offset by increased frontal area, less accessibility, or increased weight. In any event, the frontal area required for installing the air-to-air intercooler will be from 120 to 200% of the core face, while the liquid-to-air cooler will require a header whose area is perhaps only 20% of the core area and whose shape is not critical, besides coolant lines of an inch or two in diameter. Thus, while it is difficult to put an air-to-air intercooler in a fighter wing, prestone and oil coolers so installed are common.

The header problem for the crossing air is particularly difficult. Large pressure losses are likely, and uniform flow over the core face is rare. It is not uncommon to encounter an area expansion of 100% or more in the width of the core, coincident with a 90-deg turn and a transition from a circular to a rectangular section. The situation is

hardly improved by casual recommendations of 14-deg maximum expansion angles in straight ducts, of ratios of outer to inner radius of not greater than two, and so on. The use of sheet-metal guide vanes has the disadvantage that even if both the duct and the vanes are made very heavy the vanes invariably crack out under vibration and go through the engine.

In cases of restricted header space, it may be worth while to consider increasing the frontal area of the nacelle or fuselage to make possible a better turn into the core. An engineering approach to the problem can be achieved by comparing the loss in brake horsepower due to pressure loss in the duct system if the bad turn is retained on the one hand, and the effective brake horsepower loss due to the higher external drag on the other. This is best illustrated by an example:

Consider an 1800-bhp engine in an airplane capable of 450 mph at 30,000 ft. (All numbers, of course, are purely hypothetical.) First, the drag increase per square foot increase in frontal area is calculated, and this drag increase translated into equivalent loss in brake horsepower. In an actual case, the designer will know how much increase in frontal area will be required to improve the bad spot in the induction system layout. He can also obtain a fair estimate of the reduction in pressure drop which will be realized with the better system. This reduction in pressure drop will provide an equal increase in manifold pressure, which can be translated directly into brake horsepower. The two alternatives are then compared on the basis of brake horsepower.

In the case under consideration, if the frontal area is increased, there will be an effective loss in brake horsepower of about 30 bhp per sq ft increase in area for high-speed operation at 30,000 ft, and about 75 bhp per sq ft at sea level at the same speed. These figures are based on an assumption of 75% propeller efficiency and a drag coefficient of 0.08 based on the increase in frontal area.

In calculating the saving in power loss due to cleaning up a bad turn, the assumptions will be kept on the optimistic side so that the *maximum* probable saving will result. This will assist in evaluating the comparison. Now, the pressure loss in a bad turn may easily equal the dynamic pressure in the duct upstream of the turn, which will be called q, and it is a very good turn indeed which costs less than $\frac{1}{4}\ q$. The turn can be improved by expanding the duct upstream of the turn or increasing the ratio of the inside to the outside radius or both. Either change would probably increase the frontal area. A fairly optimistic figure for the reduction in pressure drop might be about half of the q ahead of the turn in the original set-up. With a specific air consumption for the engine of about 0.12 lb per bhp per min, a velocity of about 200 fps occurs in a 6-in. diameter circular duct, the absolute pressure being about 35 in. of mercury, and the temperature about 285 F. The corresponding q is 80 lb per sq ft, so the likely saving is 0.5 in. of mercury. For an engine which will produce 1800 bhp at 60 in. of mercury manifold pressure, a reasonable estimate of the variation of power with manifold pressure at constant rpm would be about 35 hp per in. of mercury, so that an improvement of 0.5 in. would produce about 18 bhp.

Thus, in the case considered, we can only justify an increase in frontal area of a half a square foot at the most in order to make a bad turn into a good one. An increase in diameter of a typical aircooled engine cowling of only ¾ in. will produce an additional half square foot, however. In the preliminary phases of a design, before tooling and production considerations make such studies purely academic, a powerplant installation can be laid out both ways, that is, first, with all ducts squeezed to the utmost, and second, with the ducts designed for moderate losses. A comparison of internal and external performance for the two arrangements will yield an answer, which, if interpreted with judgment, will serve as a reliable guide. It will seldom be found worth while to sacrifice frontal area to save duct losses.

It should be noted that there is another way of looking at the above comparison which places the increase in frontal area in an even less favorable light. If the frontal area is increased, the speed attainable with a given engine power is reduced at all altitudes and cannot be regained. The effective power loss is especially severe at low altitudes, which are not always insignificant to the high-altitude fighter. Duct losses, and for that matter, intercooler pressure drops themselves can be made up at altitudes below the critical altitude by closing the wastegate of a turbosupercharger or opening the throttle with a geared blower. Thus, duct losses simply reduce the critical altitude without affecting the performance below the critical,[3] whereas increased frontal area affects the performance at all altitudes. In the example cited, the loss of 0.5 in. of mercury in one bad turn would reduce the critical altitude about 1000 ft. This is a severe loss, but below 29,000 ft the turning loss would have no effect.

Miscellaneous Installation Problems – One installation problem which gives considerable difficulty is the structural problem. Not only must the primary structural loads be carried around the intercooler duct system, but also the intercooler must be insulated from the structural deflection. This is usually accomplished with rubber bushings which also help to reduce the vibration to which the intercooler is subjected. The necessity for four ducts to the core makes the problem of routing structure around the intercooler particularly difficult. Accordingly, there is generally a considerable weight penalty chargeable to the intercooler installation which is greater than for the prestone and oil cooler installations. Also, the structural difficulty for the intercooler limits the locations which may be selected.

The problem of protecting the induction system and especially the charge-air coolers from the pressures developed in backfires has not yet been satisfactorily solved. The use of backfire screens is impractical because of the large loss in manifold pressure which results from their use. Various types of relief valves have been tried without success. The only practical answer at the present time seems to be to design the systems for internal pressures of about 100 psi, which appears to be sufficiently high to provide reasonable assurance against damage. This requirement results, of course, in a considerable weight penalty.

The fatigue problem is acute in the charge-air system because of the large variation of temperatures and pressures to which the system is subjected. Also, space, and in many cases, aerodynamic conditions, favor noncircular ducts. Under high differential pressures, these ducts tend to be-

concluded on page 572

[3] Neglecting a slight increase in charge-air temperature and a back pressure from the turbo.

MATERIALS for Preparation and Pr
Component Pa

by
C. E. HEUSSNER and C. O. DURBIN
Engineering Division, Chrysler Corp.

THE present mechanized war requires considerable automotive equipment, trucks, tanks, and marine and industrial engines, the manufacture of which has been considered a major part of the "Battle of Production." Another phase of supply, that of transportation and shipping, has also been considered as a major factor toward eventual victory. The third phase of supply, that of protection against corrosion and physical damage, which occur during normal handling, storage, and export shipping, is of just as great importance, but has not been given as much publicity or thought by the average person.

All production effort, expense, and transportation space which have been given to an article needed on the fighting lines are wasted if that article does not arrive in a usable condition. A truck with a broken transmission, a tank with a blown cylinder-head gasket, or a gun with a broken recoil spring or with badly corroded replacement parts is just as much a casualty as though damaged by enemy action. Another consideration which has spurred industry to develop and maintain adequate preparation and packaging of vehicles and spare parts is the fact that the vehicle or spare part, no matter how insignificant, may be the means of saving a soldier's life and thus more rapidly winning the war.

The conditions of storage and shipping during wartime are considerably more severe than during peacetime and, therefore, require considerably better materials and additional protection. The following are the conditions of exposure for which the parts must be protected:

1. Since the destination of equipment is not known, it must be protected with materials which are resistant to extremes of temperature and humidity which it is likely to encounter; temperatures as low as —40 F and as high as 130 F and constant relative humidities as high as 85 to 90%.

2. Equipment may be stored for long periods of time in open storage, in rain, in ocean fog, and even in several inches of mud.

3. The equipment is handled many times from the time it is packed until it arrives at its destination.

4. Except in special instances and with high-precision equipment, the material may be shipped without preferential loading in any available cargo ship.

5. Equipment must be prepared so that it may be put into service with a minimum of time, depreservation, and assembly.

Summarizing, equipment must be prepared so that it will withstand many handlings, high humidities, ocean spray, and temperatures from —40 F to 130 F, and so that it can be put into active service in the shortest possible time.

Late in 1942, it was found that the preservation of equipment was a critical problem. Too much equipment which had been prepared with materials and methods according to pre-war standards was arriving at the theatres of war in poor condition caused both by corrosion and physical damage. Consequently, new materials and methods were necessary to protect this equipment to meet these severe conditions.

The problem of preparation and preservation of engines, vehicles, and spare parts for storage and shipment was referred to the SAE War Engineering Board, which formed a committee composed of many of the best technical representatives of industry to work with U. S. Army Ordnance in studying this problem and to make recommendations for satisfactory methods and materials to ensure that the equipment would be received in satisfactory condition. The work of this committee was divided into six subcommittees, as follows:

Subcommittee A – Preservation of the Engine.

The efforts of this group included the internal and external treatment of diesel and gasoline replacement engines and the internal treatment of these engines in all types of vehicles for boxed shipment or drive-away.

Subcommittee B – Preservation of Complete Vehicles.

The work of this group included protection of cooling system, accessories, transfer cases, front and rear axles, electrical parts, tires, batteries, sheet metal and engine accessories, and external treatment. Internal engine treatment is covered by Subcommittee A.

[This paper was presented at the SAE National War-Matériel Meeting, Detroit, Mich., June 5, 1944.]

Preservation of VEHICLES and Parts for Storage and Shipment

Subcommittee C – Preservation of Service Parts.

The efforts of this committee were concerned with the operations of cleaning, rustproofing, and packaging, prior to boxing.

Subcommittee D – Corrosion-Prevention Materials to be Used, and Their Application.

Subcommittee E – Wrapping Materials, Cartons, Dehydrating Materials, Tapes, and Sealing Materials, and Their Application.

Greaseproof wrappers, moistureproof barriers, and carton materials are included.

Subcommittee F – Accelerated Test Procedure.

This group was assigned various projects connected with the development of testing procedures, when the procedures available were not adequate.

The success of the committee and the progress made in the packaging of Ordnance material is, in great part, due to the efforts of the packaging section in the offices of the Chief of Ordnance in Washington and Detroit and in the various Ordnance districts. They have done an excellent job and are to be commended very highly. Their unusual degree of cooperation with industry in developing better packaging methods has been outstanding.

The preservation of equipment generally consists of one or more of the following steps:

1. Cleaning; removal of grease, oil, fingerprint residues, and other contaminants.
2. Applying preservative compounds.
3. Wrapping in a greaseproof or moistureproof barrier or a combination barrier.
4. Sealing against penetration by water or water vapor.
5. Cushioning, blocking, or bracing.
6. Cartoning.
7. Boxing or crating.

Steps Nos. 1, 2, 3, and 4 are done primarily for protection against corrosion while steps Nos. 5, 6, and 7 are done primarily for protection against physical damage.

In many instances, it may be necessary to provide protection against physical damage to the preservative materials as well as to the part. In other instances, the preservative materials will supplement the protection against physical damage; consequently, both problems are closely allied.

It is possible that all of the following materials may be used for corrosion protection of a single part:

1. Light preservative film from cleaner or fingerprint neutralization solution.
2. Final preservative, sometimes preceded by temporary preservatives.
3. Wrapping material.
4. Sealing compound.

All of the above must be compatible with each other or with the components which they contact. The compatibility of the packaging materials is one of the most important items in the specifications of materials and is frequently overlooked. Another factor which is important is the compatibility of rust-preventive compounds from different sources used for the same purpose if there is

TRUCKS, tanks, and engines are of no use to an army if they are delivered to the battlefield with parts broken or badly corroded.

The problems involved in protecting these parts from corrosion during storage and shipment are discussed thoroughly by the authors, who also explain the best methods and materials to use in protecting different types of parts under the extremes of temperature, humidity, and corrosive atmosphere likely to be encountered.

■

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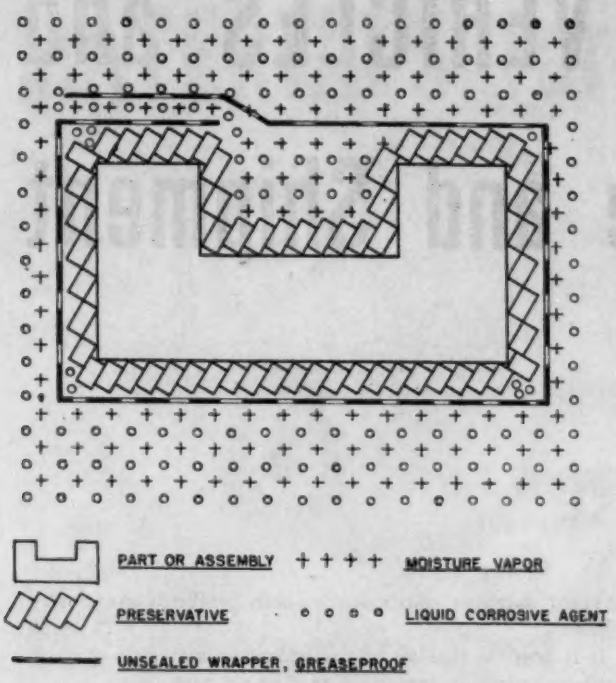


■ Fig. 1 – Functional diagrammatic sketch of Method I packaging

any possibility of their being mixed before or during application.

Incompatibility of packaging and preservative materials will result in one or more of the following:

1. Formation of sludge or gummy deposits on the metal parts.

2. Decrease in protective properties. Incompatible combinations may even become corrosive.

3. Inadequate sealing of wrapper.

4. Destruction of greaseproof and/or moisture-vapor-proof properties of wrapper.

Since failure due to incompatibility in many cases is similar to failure from other causes, it is not always easy to determine. Compatibility may be often predicted from a knowledge of the constituents of the materials; however, this information is very seldom available to the user. It is, therefore, necessary that he determine the compatibility of the cleaning, preservative, and packaging materials which he is using, and use only materials which are known to be compatible.

The preservation of metal equipment for shipment and storage starts as soon as the parts have been completed and inspected whether for assembly into larger units or for shipment as spares.

Metal parts must be clean before any preservative treatment is applied, since many types of contamination are either corrosive or tend to accelerate corrosion in a highly humid atmosphere even underneath the best preservative compounds. Such contamination may consist of oils, greases, cutting compounds, fingerprint residues, or soldering or brazing fluxes.

The removal of oils, greases, cutting compounds, and so on, is important since they may not be compatible with the preservative compounds, or since they may be corrosive in themselves.

Perspiration residues (fingerprints) are left on the parts in handling during manufacture and inspection. The removal or neutralization of these is extremely important because of their acidity and salt content. The damage from fingerprints usually does not occur before the parts are packaged, but is very apparent when they are unpacked and ready for use. This type of contamination will frequently cause sufficient corrosion under most rust-preventive compounds to make the parts unusable. In addition, we must consider soldering and brazing fluxes, which usually cause more damage than any of the above groups, and their removal is frequently scheduled as a part of regular production operation.

The materials which may be used for removal of the contamination depend largely on the nature of the part, the type of contamination, and the equipment available or most readily adaptable. The materials most frequently used are:

1. Alkaline cleaners.
2. Solvent emulsion cleaners.
3. Petroleum solvents, such as mineral spirits.
4. Solvent vapor cleaners (degreaser compounds).
5. Methanol (methyl alcohol).
6. Special compounds for neutralization of fingerprints.
7. Rust-removing compounds.

Alkaline cleaners are used for metallic parts or simple assemblies. They will remove oils, greases, fingerprint residues, and almost any type of contaminant likely to be on metallic parts. They cannot be expected to remove rust or other corrosion. There are many satisfactory formulas and proprietary alkaline cleaners containing combinations of various alkaline sodium salts and inhibitors which are available. In general, alkaline cleaners must be specifically formulated for the type of metal being cleaned. They are not generally satisfactory for nonmetallic parts.

Parts which are cleaned in alkaline cleaners must be thoroughly rinsed. A small quantity of the alkaline cleaner (between 0.02-0.05%) is usually maintained in the rinse water, to prevent corrosion of the parts in drying. If the alkali content is much greater, a heavy residue of alkali will be left on the parts and will tend to accelerate corrosion.

Alkaline cleaners are more difficult to operate satisfactorily than some of the other types of cleaners; consequently their use on high precision parts has been questioned. When the cleaners are properly maintained and operated, however, they may be satisfactorily used for either precision or nonprecision parts. They may be used in simple immersion, spray, or electrocleaning equipment.

Solvent emulsion cleaners usually contain mineral solvents or light oils with suitable emulsifying agents. In use, the cleaner is mixed with water and agitated sufficiently to make an emulsion. Solvent emulsions may be used in simple immersion or spray equipment. This type of cleaner is used for metallic parts and simple assemblies, and will remove oils, greases, dust, dirt, and metal chips (with suitable agitation); however, a thin film of oil from the cleaner will be left on the part. Emulsion cleaners will not, in general, neutralize fingerprints. The emulsion cleaners also require thorough rinsing in hot water. Since this type of cleaner leaves a thin film of oil on the part which is

not removed in rinsing, little difficulty is encountered with rusting during the drying operation. The compatibility of the preservative with the thin oil film left on the parts is often difficult to maintain, because the cutting oils, grease, and so on, removed from the parts may considerably contaminate the emulsion cleaners, and thus the residual film. This contamination may be of such a nature as to render this film incompatible with the preservative.

Petroleum solvents are frequently used for cleaning parts prior to the application of rust-preventive compounds. They may be used in immersion or spray equipment for removal of dust, dirt, chips, oils, greases, and so on. At least two stages are used – a wash and a rinse. They will not neutralize fingerprints. The use of petroleum solvents involves a definite fire hazard, and cannot be tolerated in many plants for this reason. This type of cleaner is easily contaminated with oils, greases, and so on, thin films of which are left on the parts after cleaning. As the solvents are used, the build-up of contamination increases and the film left on the parts becomes greater. As in solvent emulsion cleaning, this film presents a compatibility problem.

Vapor degreasing, using trichlorethylene, perchlorethylene, or other chlorinated hydrocarbon solvents, is one of the most popular methods for cleaning of parts prior to the application of rust preventives. This type of cleaning is the most satisfactory for the removal of oils and greases and, if a spray phase is used, is satisfactory for removal of chips and similar contamination. Vapor cleaning usually consists of three steps:

1. Immersing the parts in the hot vapor of the solvent.
2. Flushing with the cold solvent, either by dipping into the solvent or by spraying.
3. Reimmersing in the hot vapor of the solvent.

When cold parts are placed in the solvent vapor, the solvent condenses on the part, dissolves the oil and greases, and drips back into the boil chamber, carrying the oil and grease contaminants in solution. The chips and dirt are removed by the mechanical action of flushing in the solvent. The final immersion in the vapor is to remove any contaminants left by the flushing solution.

Of the above, only alkaline cleaners will adequately remove fingerprints. Consequently, cleaning with the other materials must be followed by a treatment either to neutralize or remove fingerprint contaminations. The first method proposed was to immerse the parts for 2 min in methanol (methyl alcohol) containing 5-10% water. Considerable ventilation and fire-fighting equipment are necessary to safeguard against the fire and health hazards involved in the use of methanol.

Because of the amount of floor space required for automatic alkaline cleaning equipment and because of the fire and health hazards involved in using methanol, there was considerable effort on the part of industry to develop other materials for neutralization or removal of fingerprints which could be used with vapor degreasing. Several special light-oil rust-preventive compounds have been developed for use after degreasing and prior to applying the final preservative. These compounds are usually mixed with from 5 to 95% water. Some of these special rust-preventive compounds will completely neutralize fingerprints after 15 sec immersion, while others will only inhibit such corrosion when the compound is on the part. Parts which have been coated with the latter type will show definite fingerprint patterns after the parts are cleaned and exposed to highly humid conditions. The use of the special rust-preventive compounds as an immersion treatment after degreasing has permitted cleaning and rustproofing in plants where the floor space and equipment required by other methods were not available.

For simplicity, methods of packaging of Ordnance parts may be divided into three general classifications. These are:

Method I – Breathing pack (not airtight) in which the preservative is applied to the part and the part wrapped in a greaseproof wrapper which will permit the free transfer of air and moisture to and from the part. This type of package is shown diagrammatically in Fig. 1. The preservative is applied directly to the cleaned part and is kept intact by a greaseproof wrapper. Air and moisture have free access to the interior of the pack. Consequently, this type of packaging is used only where the preservative is sufficient to protect the part or where a slight amount of corrosion will not be harmful.

Method IA – Sealed pack in which the part is coated with rust-preventive compounds, wrapped and sealed in a greaseproof and moisture-vaporproof wrapper. This type of package is shown diagrammatically in Fig. 2.

The preservative is applied directly to the cleaned part and is kept intact by the wrapper. This wrapper is then sealed to prevent free access of air and moisture. The sealed wrapper protects the part against such corrosive conditions as ocean spray, rain, and mud. The only function of the preservative, then, is to protect the part against enclosed moisture and the small quantity of moisture which may diffuse through the wrapper. Where possible, however, the preservative on parts in Method IA packages is usually sufficient to protect the parts for a short time without the wrapper.

Method II – Sealed pack in which the part is sealed with a dehydrating agent in a moisture-vaporproof wrapper. This type of package is shown diagrammatically in Fig. 3.

The cleaned part is not treated directly with a preserva-

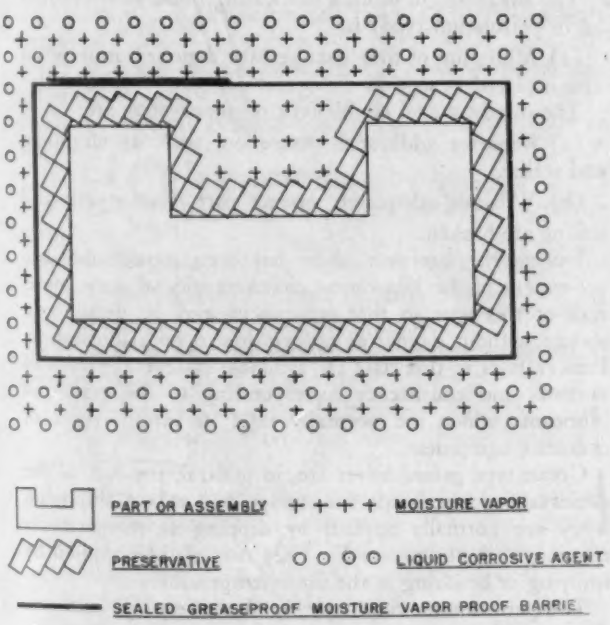


Fig. 2 – Functional diagrammatic sketch of Method IA packaging

tive but is enclosed with a dehydrating agent in a moisture-proof sealed wrapper. The dehydrating agent absorbs the moisture inside the wrapper and the wrapper protects the part from high humidity, rain, and ocean spray.

The type of preservative and packaging needed depends to a large extent upon the nature of the part or equipment, the state of disassembly which will be permitted for packaging, and the amount of unpacking and cleaning which can be done when the part is placed in service.

The following simple chart summarizes the method of packaging usually given to various types of parts:

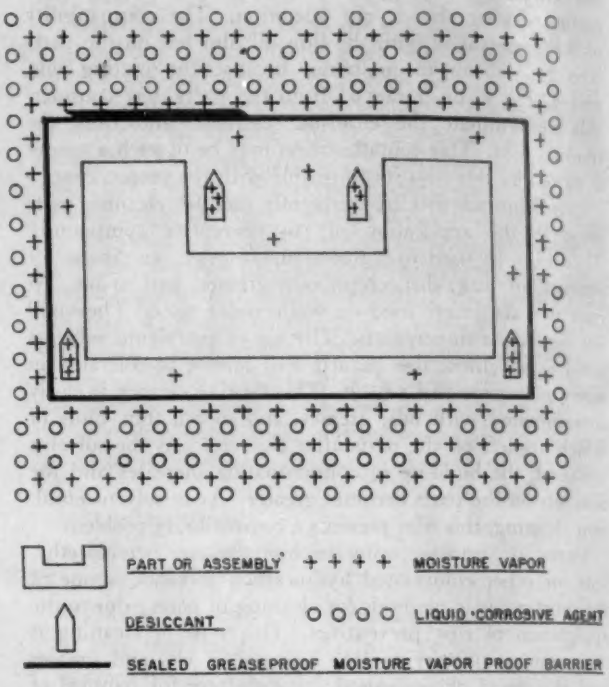
Type of Part	Method of Packaging
1. Simple parts.	
a. Parts with precision and highly finished surfaces.	Method IA (rustproofed and package sealed).
b. Nonprecision parts.	Method I (rustproofed and wrapped).
c. Nonmetallic parts.	Modified method IA (without preservative, but package sealed).
d. Painted parts.	Packaged for physical protection only.
2. Assemblies.	
a. Assemblies with operating mechanisms.	Method IA (rustproofed and package sealed) or Method II (dehydrated and package sealed).
b. Structural assemblies.	Method I (rustproofed and wrapped) or Method IA (rustproofed and package sealed).
c. Assemblies of metallic and nonmetallic materials.	Method II (dehydrated and package sealed).

Most of the materials used in such packaging operations as these are covered by specifications of the various branches of the services. The applicable U. S. Army Ordnance Specifications may be found by referring to Army Service Forces Manual M-406, *Corrosion Prevention Processing and Packaging;* and to U. S. Army Specification No. 100-14a, *Army-Navy General Specification for Packaging and Packing for Overseas Shipment.* The general discussion, therefore, will briefly describe the materials and give the highlights of their uses and their performance requirements. These requirements are usually determined in the laboratory by means of accelerated tests which have been correlated with service performance. Some of the tests which are widely used are humidity, salt spray, water immersion, and cycle tests. Cycle tests consist of daily exposing the same test pieces to high temperatures, low temperatures, water spray, and high humidity. Many of these tests, although reproducible in one laboratory, connot exactly be duplicated by other laboratories, since the results depend to a large extent on the equipment used.

Wherever complicated operating assemblies, such as engines and transmissions, are prepared for storage or shipment, they must be preserved with a compound which will be completely miscible with the lubricant which is to be used in the assembly. The performance requirements of this type of preservative are:

(a) Must protect against corrosion from condensation.

(b) Must be easily applied by filling lubricant reservoir and operating unit.



■ Fig. 3 – Functional diagrammatic sketch of Method II packaging

(c) Must be either compatible with lubricant or be easily removed by flushing with lubricant.

(d) Must be sufficiently fluid to recoat small areas when the preservative film has been removed.

Such lubricating preservatives are usually satisfactory to protect parts against condensation of moisture present in the units when sealed but are not adequate for protecting against extreme conditions of rain and ocean spray.

The advantage of using a lubricating preservative (either oil or petrolatum type) is:

(a) Minimum of time required for depreservation when unit is placed in service.

The disadvantages of this type of preservative are:

(a) Requires additional protection, such as wrapping and sealing.

(b) Will not adequately protect part if wrapping and sealing are broken.

During the last year, there has been considerable improvement of the lubricating characteristics of some materials of this type so that equipment may be given light service without danger of failure due to poor lubrication. Preservatives in this class are available in several grades of viscosity and consistency corresponding to the grades of lubricants which are normally used in various types of ordnance equipment.

Grease-type preservatives are, in general, the best of the preservative compounds for storage and export shipment. They are normally applied by dipping at temperatures ranging from 160 to 200 F. They may also be applied by spraying or brushing at the same temperatures.

The performance requirements of this type of preservative are as follows:

(a) Must protect against corrosion in conditions of high humidity, rain, and ocean spray.

(b) Must maintain sufficient film thickness after heating to 140 F to protect parts.

(c) Must be sufficiently hard to permit handling of parts coated with the preservative without penetration to the base metal.

(d) Must be sufficiently fluid at application temperatures to cover completely the smallest parts which are coated in bulk with the material.

(e) Must be sufficiently soft to permit separation of small parts coated in bulk without removal of coating from either part.

(f) Must be miscible with lubricating oils at operating temperatures.

The advantages of using grease-type preservatives are:

(a) The preservative will fill in over small areas if film is broken in handling.

(b) A maximum of protection is given an unwrapped part.

(c) Application is easy and rapid if hot dipping equipment is available.

(d) If preservative is properly chosen, it may be used for a wide variety of parts.

The disadvantages of using grease-type preservatives are:

(a) They are messy to handle.

(b) Preservative usually must be removed before part is put into service.

Grease-type preservatives are also available in several grades and should be chosen to give the best protection to the average size of the parts to be protected. In general, soft materials will be best for small parts and hard materials for large parts.

Many parts lend themselves to coating with hard-drying rust-preventive films. These films are usually of a waxy or soft resinous nature and are applied by dipping, brushing, or spraying the parts in a mineral spirit solution of the compound. The performance requirements of this type of preservative are:

(a) Must have good protecting qualities against high humidity, rain, and ocean spray.

(b) Must have film sufficiently hard and flexible to resist damage.

(c) Must be easily removable with gasoline and similar solvents.

The advantages of using this type of preservative are as follows:

(a) Coated parts are not messy to handle.

(b) May be easily applied with a minimum of equipment.

(c) Coating not easily damaged during handling.

The disadvantages of using this type of preservative are as follows:

(a) Coating must be removed before parts of actuating mechanisms are placed in service.

(b) If coating is damaged, protection is lost.

(c) Coatings of this type are more difficult to remove than the preservatives.

Experience has shown that most metals will not corrode if the relative humidity is below 20%. Therefore, the use of dehydrating agents which will maintain the humidity inside a moisture-vapor barrier below this value will protect the parts. Since the available wrapping materials are not completely impervious to moisture vapor, sufficient dehydrating agent must be used not only to absorb the moisture present in the container, but also to absorb the small quantities of moisture which will pass through the container in a given period of time. The dehydrating agent most widely used in packaging is silica gel.

The performance requirements of dehydrating agents are:

(a) Must absorb large quantities of moisture in order to maintain a low relative humidity (at least 8.5% of its weight to maintain the relative humidity below 20%).

(b) Must be noncorrosive in contact with metal even when in equilibrium with air at high humidities.

(c) Must be placed in containers which will segregate dehydrating agent from article being preserved.

The advantages of using dehydrating agents are:

(a) No preservative oils or compounds are necessary.

(b) May be used on parts which will be damaged by other preservatives.

(c) Parts and equipment may be put into service immediately after packaging material is removed.

The disadvantages of using dehydrating agents are:

(a) When moisture vapor container is broken, preservation is lost.

(b) Great care must be exercised in proper use and storage for dehydrating agents.

(c) Preservation is good for a finite period only, depending upon the quantity of dehydrating agent used and the effectiveness of the barrier. When the dehydrating agent is completely used, the package must be opened and the dehydrating agent replaced.

The life at various temperatures and humidities of dehydrated packages which are prepared with materials meeting the minimum specifications is shown diagrammatically in Fig. 4. The data presented in this chart are based on the following assumptions:

1. That the moisture-vapor transmission rate of the wrapper is 0.25 g (0.000551 lb) per 100 sq in. per 24 hr with a water-vapor pressure differential across the wrapper of 42 mm of mercury.

2. That the moisture-vapor transmission rate is directly proportional to the water-vapor pressure differential.

3. That the absorptive capacity of the dehydrating agent (silica gel) at 20% relative humidity is 8.5% of its weight.

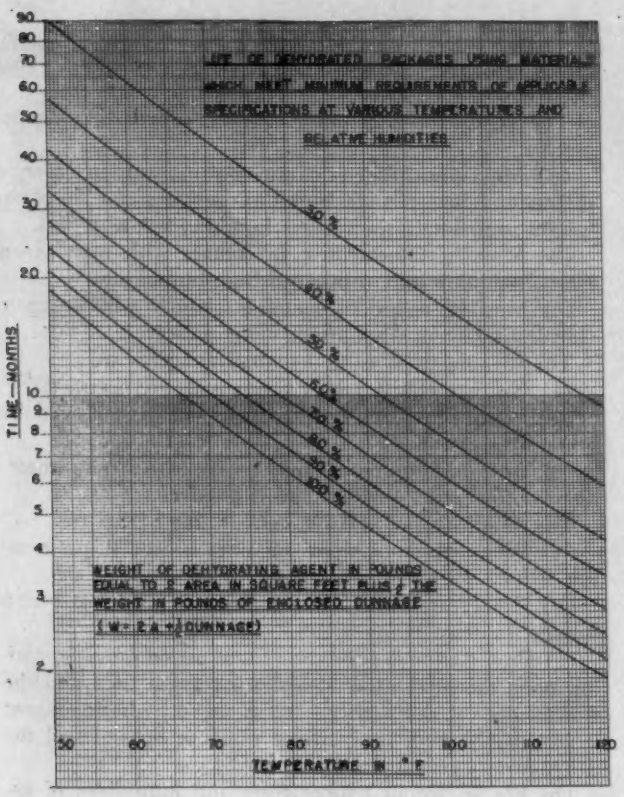
4. That the quantity of dehydrating agent used was calculated from the following formula, which defines the minimum quantity permitted by the U. S. Army Ordnance Department for Method II packs:

$$W = 0.2A + \tfrac{1}{2} D$$

where:

W = Weight of silica gel, lb
A = Area of enclosing wrapper, sq ft
D = Weight of enclosed hygroscopic dunnage, such as cartons

This chart is useful in illustrating the effects of continued exposure to high temperatures and humidities. For example, the average conditions during the hottest month of the year in Detroit are 72 F and 69% relative humidity. If these conditions were maintained throughout the year the life of dehydrated packages in this area would be 13 months; but since this is only for the hottest month, it can readily be seen that the life of dehydrated packages in this area would be considerably greater. For the South Pacific area where average temperatures of 85 F at 90% relative humidity are continuously encountered, the life of dehydrated packages is only six months. This agrees quite closely with the experiences reported for aircraft engines


LIFE OF DEHYDRATED PACKAGES USING MATERIALS WHICH MEET MINIMUM REQUIREMENTS OF APPLICABLE SPECIFICATIONS AT VARIOUS TEMPERATURES AND RELATIVE HUMIDITIES
20%
40%
50%
60%
70%
80%
90%
100%
TIME—MONTHS
WEIGHT OF DEHYDRATING AGENT IN POUNDS EQUAL TO ½ AREA IN SQUARE FEET PLUS ½ THE WEIGHT IN POUNDS OF ENCLOSED DUNNAGE
(W = ½A + ½DUNNAGE)
TEMPERATURE IN °F

■ Fig. 4 – Life of dehydrated packages

stored in this area. These engines were prepared with approximately half the dehydrating agent required by the formula used in preparing the chart and enclosed in a barrier with a moisture-vapor transmission rate near the limit of the specification. It was found necessary to replace the dehydrating agent used with these engines at three-month intervals.

In general, the dehydrating agents available will absorb 25 to 50% more moisture than specified at the 20% humidity level, and most wrapping materials used by Ordnance for dehydrated packs have lower moisture-vapor transmission rates, therefore, the expected life of dehydrated packages will usually be greater than shown in the chart.

After the part has been treated with a preservative, the preservative is kept intact by enclosing in either a Grade A, greaseproof wrapper, or Grade C, greaseproof and moistureproof wrapper. When dehydrating agents are used, the part and dehydrating agent are enclosed in moisture-vaporproof wrappers.

Method I packaging as outlined above requires the use of a greaseproof wrapper which will transmit moisture vapor but which will not transmit oils and greases. This type of wrapper is generally designated as Grade A wrapper. Certain Ordnance specifications for nonprecision parts permit Grade B greaseproof wrappers. This grade differs from Grade A only in the degree of acidity or alkalinity permitted.

Since the Grade A wrappers are superior to Grade B and since the cost and availability are almost the same, the use of the Grade B wrapper for preparation of parts for storage and shipment is not recommended.

1. Performance requirements of Grade A greaseproof wrappers are:

(a) Must be greaseproof (must not transmit or absorb grease).

(b) Must not accelerate corrosion of metals when in direct contact with them.

(c) Must be resistant to deterioration and development of corrosive tendencies due to mildew and mold growths.

(d) Must be sufficiently strong and flexible to permit wrapping of the part without tearing or breaking.

(e) Must be sufficiently flexible to permit speed in wrapping without undue fatigue of operator.

One of the developments of this war in packaging has been the Grade C wrapper. This wrapper is impervious to oils and greases and has a high degree of resistance to water and moisture-vapor transmission. There has been a slight controversy among various packaging groups as to the degree of water- and moisture-vaporproofness required of the wrapper for Method IA pack. Some believe that the wrapper should be waterproof, but must be able to transmit moisture vapor, others believe that the wrapper should be as moisture-vaporproof as wrappers for use with dehydrating agents. Our experience has shown that the latter type does give excellent protection. Performance requirements of these wrappers are:

(a) Must not transmit or absorb preservative oils or greases.

(b) Must not accelerate corrosion of metals when in direct contact with them.

(c) Must be sufficiently strong to permit wrapping of the part without tearing or breaking.

(d) Must be sufficiently flexible and tacky to conform to the contours of the part and to permit speed in wrapping.

(e) Must have a low moisture-vapor transmission rate.

(f) Must lend itself to sealing against moisture vapor.

(g) Must be resistant to deterioration and development of corrosive tendencies due to mildew and mold growth.

The original Grade C wrapper was used to protect underground pipe lines against corrosion and was adapted to the wrapping of parts for export shipment. It consisted of a sheet of cellulose acetate laminated to a sheet of muslin with a microcrystalline wax-base compound. This has been supplemented by other materials, which have since been developed, including cellophane or cellulose acetate laminated to paper with microcrystalline wax and various metal foils laminated to paper with asphalt-base compounds.

As previously outlined in the discussion of the dehydrating agent, Method II packaging depends to a large extent on the quality of the moisture-vapor barrier and the completeness of sealing. The performance requirements of moisture-vapor barriers are:

(a) Must have exceptionally low moisture-vapor transmission rates.

(b) Must be adaptable to sealing operations.

(c) Must have sufficient strength and resistance to wear to resist breakage and puncture in handling operations.

(d) Must be noncorrosive if allowed to contact metal parts.

(e) Must be resistant to deterioration and development of corrosive tendencies due to mildew and mold growths.

Satisfactory materials which have been recently developed for moisture-vapor barriers include:

(a) Certain plastic films, such as Pliofilm and Saran.

(b) Certain of the Grade C wrappers of the microcrystalline wax, cellulose acetate, or cellophane laminations.

(c) Metal foil, asphalt, paper, and cloth laminations.
(d) Certain Kraft paper-asphalt laminations.

The wrapping materials have been divided into the three main classes because of the function which they perform. In general, Grade A, greaseproof wrappers for Method IA packaging may not be used as Grade C wrappers or moisture-vapor barriers, and conversely. Under certain exposure conditions which the parts are likely to encounter, the use of Grade C wrappers for Method I packaging (without sealing) will be considerably inferior to Grade A wrappers because of the entrapment of moisture and consequent slow evaporation. Many of the wrappers may be identified with both the Grade C wrappers and with moisture-vapor barriers, since they may be used satisfactorily for either Method IA or Method II packaging. In addition, certain materials which are good moisture-vapor barriers but are not greaseproof or are corrosive may be used with an underwrap of Grade A wrapper for Method IA packaging.

The sealing of the package is one of the most critical steps in the preparation of highly finished parts for export shipment in accordance with Methods IA or II. If the method of sealing is inadequate, the protection will be inadequate and all the effort of making the equipment and wrapper and in packaging the part is wasted. The method of sealing must be specifically designed for the type of wrapper. The following are the more common methods of sealing:

Wrapper	Method of Sealing
1. Plastic films.	a. Moisture-resistant adhesive tape.
	b. Heat sealing.
	c. Gathering open end (grocery-bag style), threading through a lead ring, and crimping lead ring.
2. Laminations with microcrystalline wax.	a. Dipping in microcrystalline wax compounds.
	b. Lap sealing with special wax sealer compounds.
3. Laminations with metal foil.	a. Special heat-sealing films.
4. Asphalt Kraft laminations.	a. Special asphalt compounds.

The performance requirements of seals for packages are:

a. Must have as low (or lower) moisture-vapor transmission rate as material being sealed.

b. Must be water resistant.

c. Must have sufficient strength to prevent breaking of seal in further packaging and handling.

d. Must be resistant to deterioration of seal by mold growth and mildew.

One of the most recent developments in packaging materials is the plastic-base, stripping compound. These compounds usually contain ethylcellulose combined with various oils and plasticizers. The plastic coatings were developed to give protection which would be equivalent to that given by coating with an oil or grease preservative, wrapping in Grade C wrapper and sealing. These coatings are usually applied by dipping the part in the plastic at a temperature of 325 to 375 F.

Under the present state of development this type of preservation is suitable for simple parts from which the coating may be readily removed, and of such shape that they can be supported or cushioned on sufficient area to prevent damage to the coating by the weight of the part.

The performance requirements of plastic coatings are:

(a) Must protect parts from high humidity, rain, and ocean spray.

(b) Must be sufficiently hard at high atmospheric temperatures to resist penetration due to the weight of the part.

(c) Must be strippable.

(d) Must not crack or break at low temperatures.

(e) Must set within a reasonable length of time after coating.

(f) Must be stable at the application temperatures.

The important advantage of using this type of coating is the saving in man power and floor space. Much less handling and equipment are required in using the plastic coating than in other methods of protection. One of its disadvantages is the loss of a large amount of the protection when the coating is damaged.

The number of parts which can be satisfactorily protected with this type of compound is rapidly increasing with new methods of application.

Of equal importance to corrosion protection is the protection against physical damage. Most of the wrapping materials which are used for Method IA and Method II packaging require physical protection to withstand the number of handlings which the parts receive in export shipment. For the smaller, lighter parts, fiberboard cartons or folding and set-up boxes inside nailed, wooden boxes are satisfactory, the strength of the cartons and boxes being dependent upon the size of the parts.

Frequently, the carton is coated with a greaseproof material, such as wax, which takes the place of the Grade A Wrapper used for Method I packages. Since the start of the war, the industry has considerably improved the quality and water resistance of the carton materials so that materials are now available which will not delaminate and which will maintain more than half of their strength after water immersion for 24 hr.

Additional physical protection for parts with projections and sharp corners may be accomplished by enclosing fillers made of carton materials or by using cushioning materials made of shredded cellulose. Cushioning materials which will not absorb and hold large quantities of moisture should be chosen for packaging, otherwise such materials may maintain corrosive conditions over long periods of time.

Cartoned parts are given further protection for overseas shipment by packing in a nailed wood box which is lined with waterproof paper. This waterproof paper is made from two or more sheets of Kraft paper laminated with asphalt. The materials used for sealing case liners are of the asphalt-base type. Since it is the case liner which protects the cartons against rain and ocean spray, it must be resistant to water penetration but not necessarily to moisture-vapor penetration.

The best material available for construction of exterior packing cases is wood. By using proper design, wood packing cases, wire-bound boxes, and crates will have the necessary strength, resistance to puncture, and cushioning against shock.

Large parts and completed vehicles are usually packed directly in crates, the sides of which are lined with waterproof paper similar to that used for case liners in nailed wood boxes. The crates for the larger parts, such as engines and axles are designed so that they may be reused. That is,

when a damaged unit is replaced, it is shipped to a major repair base in the crate originally containing the replacement part.

Perhaps the best way to sum up the ways in which these materials are used would be to give a few specific examples as to how they are used in the preparation of parts.

(a) Exhaust manifold (nonprecision part).

1. Clean in trichlorethylene vapor degreaser. No fingerprint neutralization necessary.
2. Coat with hard-drying rust preventive.
3. No wrapping necessary.
4. No sealing required.
5. Corrugated fillers for blocking.
6. Place in wax-lined carton.
7. Place case liner in nailed wood box, fill case liner with one or more cartons, seal case liner, close box, and band.

(b) Piston (precision part).

1. Clean in trichlorethylene vapor degreaser. Neutralize fingerprints in special rust-preventive water mixture.
2. Coat with a grease-type preservative (hot application).
3. Wrap in Grade C wrapper.
4. Seal by dipping in wax-base dip sealing compound.
5. Place in carton.
6. Place case liner in nailed wood box, fill case liner with cartons, seal case liner, close box, and band.

(c) Generator (assembly of metallic and nonmetallic materials).

1. Clean metallic materials before assembly.
2. Protect with dehydrating agent by enclosing with part.
3. Place generator in carton with corrugated fillers.
4. Wrap carton in moisture-vapor barrier.
5. Seal (by dipping in wax compound).
6. Place in second carton.
7. Place case liner in nailed wood box, place carton in case liner, seal case liner, close box, and band.

(d) Spare gasoline engine (operating assembly containing minor assemblies of metallic and nonmetallic parts).

1. Clean metallic parts before assembly.
2. Fill lubricating system with lubricating oil preservative and operate unit sufficiently to coat all parts in contact with the lubricating system.
3. Apply preservative to upper cylinder walls by spraying through air intake during the last 5 min of operation.
4. Drain coolant from cooling system and blow dry with air.
5. Coat all exterior bare metal surfaces with grease-type preservative.
6. Close all openings to interior of engine with Grade C wrapper and seal wrapper with special wax sealer.
7. Enclose entire engine with dehydrating agent in moisture-vaporproof wrapper (same as Grade C wrapper), and seal wrapper with special wax sealer.
8. Bolt engine supports to blocks in base of crate.
9. Crate for shipment. Sides, ends, top, and bottom of crate to be lined with waterproof paper.

We wish to take this opportunity to thank the Engineering Division of the Chrysler Corp. for permitting publication of this paper, the staff of the materials testing laboratory for their help in its preparation, and the SAE War Engineering Board Committee on Preparation and Preservation of Matériel for making available a large portion of the data used in this paper.

INTERCOOLING PROBLEM

continued from page 563

come circular, and the large deformations encountered produce cracking.

A number of attempts have been made to employ automatic regulation of the carburetor air temperature by thermostatically controlling the outlet flap on the intercooler cooling-air duct. Unfortunately, there is little thermal capacity in the system, so that with air-to-air intercoolers, extremely rapid response is needed from the control. For instance, if the airplane is cruising at low power at high altitude and the pilot suddenly increases the power, the carburetor air temperature reaches its maximum value as soon as the turbo speed is increased. In general, a few seconds of detonation may cause severe engine damage so that practically instantaneous cooling is needed. Under the circumstances in which sudden power increases are necessary, the pilot should not have to concern himself with the adjustment of intercooler flaps. Unfortunately, thermostatic controls having an adequate rate of response are not yet available.

The problem of preventing excessive leakage in the system has found no entirely adequate solution. Leakage, however, is not as serious as some other difficulties because at altitudes below the critical it causes very little power loss. This follows from the fact that the supercharger is generally working on the flat part of the curve of efficiency against airflow. Moderate increases in the airflow do not have any appreciable effect on the efficiency. However, at the critical altitude, with the turbosupercharger, the output is limited by the maximum turbo rpm and with the geared blower, full throttle is the limitation. In either case, an increase in the airflow due to leakage makes available a reduced pressure rise from the blower with consequent loss in critical altitude.

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FILTRATION of DIESEL FUEL and LUBRICATING OILS

The opinions and assertions contained in this paper are the author's, and are not to be construed as official or reflecting the views of the Navy Department or the Naval Service at large.

by H. V. NUTT*

THE functions of a lubricating oil or fuel oil filter seldom have been clearly defined; probably because of the varying claims of the filter manufacturers, the inability of the petroleum industry, engine builders, and engine users to agree on how "clean" fuel oil and used lubricating oil should be for satisfactory engine operation and the difficulty of obtaining reliable test data. Since the advent of detergent-type lubricating oils and due to their present widespread use, especially by the military organizations, many of the claimed advantages ensuing from the use of lubricating oil filters, such as reduced ring sticking and blowby, reduced valve sticking and reduced bearing corrosion are no longer important. These functions are now generally considered to belong to the lubricating oil.

■ Navy Standard Filter Sizes

Prior to the general introduction of additive-type oils, the Navy initiated tests of lubricating oil filters for the purpose of determining the ability of various commercial filters to maintain the solid content of straight mineral oils in diesel engines at a low value. This investigation was expanded with the increased use of diesel engines by the Navy until a large variety of filter elements varying in size from 60 cu in. to 1400 cu in. were in use. This large variety resulted in a difficult supply and distribution problem, often preventing the renewal of elements, when required, due to lack of availability. Recognizing that these problems would become more acute due to a rapidly expanding Navy and the increased sphere of naval operations, it was decided to standardize on the sizes of both fuel and lubricating oil filters and to limit them to the minimum practicable number of sizes. As a result of this program, two sizes each of fuel and lubricating oil filters elements have been substituted for the 40-odd sizes previously required. The sizes of the standard elements are given in

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ABRASIVE material can be maintained at the lowest concentration by circulating the lubricating oil through the filter element at a high rate.

High flow rates can be obtained by shunt operation, high lubricating oil temperatures, and by maintaining a high pressure drop across the filter element.

For intermittent-duty service, it is important to have rapid warm-up of the oil in the filter body, which can be aided either by attaching the filter body to the engine or by installing it in close proximity to the engine and passing through the filter case all or nearly all the oil flow, while maintaining a safe pressure drop with an internally built relief valve.

For test purposes, it is only necessary to have a method of oil solids content determination that is reproducible and that will remove more of the solids than any of the filters being tested. There is still a need, however, for rigid definition of the important types of solid contaminants found in lubricating oil and standardization of accurate, reproducible methods for the quantitative determination of such contaminants.

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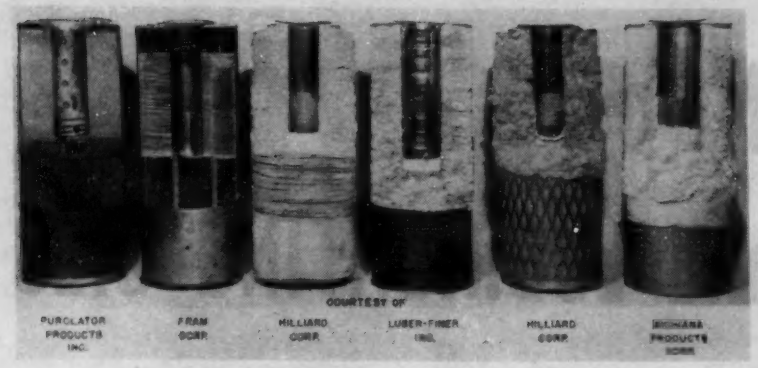


■ Fig. 1 – Typical large Navy standard lubricating oil filter elements – height: 18 in.; diameter: 7 5/16 in.

Table 1. The detailed specifications are available and may be obtained by application to the Bureau of Ships, Navy Department, Washington, D. C.

Table 1 – Navy Standard Filter Element Sizes

	Lubricating Oil		Fuel Oil	
	Large	Small	Large	Small
Overall length, in. (max.)	18 1/16	8.0	8.0	4.0
Overall length, in. (min.)	17 15/16	....	....	....
Diameter, in. (max.)	7 5/16	4.5	3.0	3.0
Installed length, in. (min.)	17 15/16	....	7 5/8	3 13/16
Installed length, in. (max.)	18 1/16	8.0	7 11/16	3 27/32
Type of end seal	Comp.	Radial	Comp.	Comp.
Outside diameter of seal gasket, in.	4.0	....	*	*
Inside diameter of seal gasket, in.	3.0	9/16	*	*
I. D. of center tube, in. (min.)	2 3/32	....	1.0	1.0
I. D. of centering device, in. (min.)	2 3/32	....	1.0	1.0
I. D. of centering device, in. (max.)	2 3/32	....	1 1/16	1 1/16

* Gaskets must be suitable for use with compression sealing washers incorporating two concentric V ridges having diameters at the apex of the V of 1 3/8 in. and 1 7/8 in., respectively.

The Navy standard, small, lubricating oil filter element is identical in size to the Military standard senior element. All makes of elements in any one size classification are required to be interchangeable. Figs. 1, 2, and 3 illustrate the sizes of these elements.

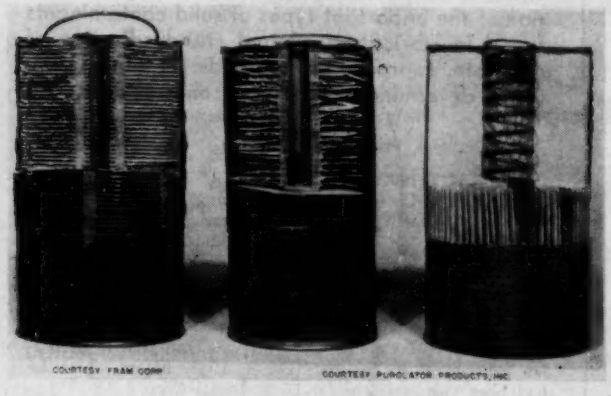
■ Performance Requirements

After fixing the size of filter elements, it was necessary to formulate methods for evaluating their performance. Two important requirements for lubricating oil filter elements are:

(a) Long life.

(b) High solid removal rate.

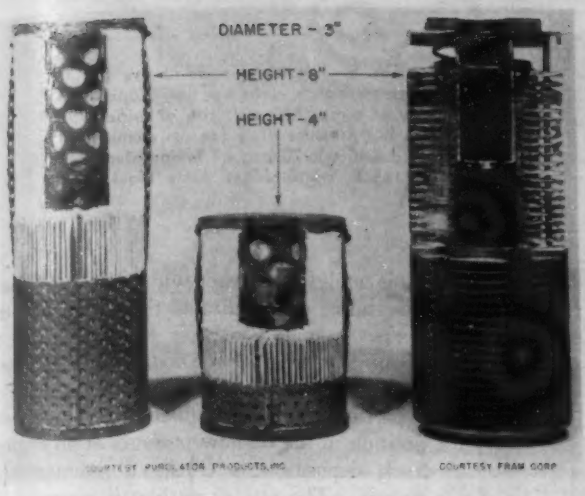
The length of life can be determined by measuring the amount of solids removed by the filter before the rate of removal of such solids falls to an unsatisfactory, low value. This determination was made by means of a batch clean-up test and an engine test. In the batch clean-up test a large Navy standard filter with a 1/8-in. metering orifice in the discharge line was installed in a system so that a batch of 500 gal of N.S. 3065 straight mineral oil contaminated with 0.20%*w* of fine lamp black was constantly recirculated through the filter. A pressure drop of 20 psi across element and orifice and a temperature of 165 F were used. The test was continued until the rate of solids removal dropped below 0.0008 lb per min. This rate of removal was considered to be equal to the maximum rate of increase in the solids content of oil in a service engine. A typical analysis of filter performance on this test is shown in Fig. 4. The solids content of the oil in the 500-gal sump tank and the flow rate through the filter were determined at 1-hr intervals and plotted on semilog paper. The rate of contaminant removal from the oil was obtained from this curve. The amount of dirt stored in the filter was considered to be equal to the amount removed from the oil. The amount of this removal at the time when the rate of removal had dropped to 0.0008 lb per min was called the dirt storage capacity. A minimum dirt storage capacity of 2.5 lb was required to pass the test satisfactorily.



■ Fig. 2 – Small Navy standard lubricating oil filter elements – height: 8 in.; diameter: 4 1/2 in.

The results of tests by the batch clean-up method of five typical Navy standard large filter elements are shown in Fig. 5. Filters 66, 16, and 12 displayed only small reductions in flow rate. In addition, No. 66 removed very little of the contaminant from the batch of oil, indicating excessive porosity. No. 12 reduced the solids content to less than half the original concentration in 50 hr while maintaining a high flow rate. Filter 14 removed very little of the contaminant and the flow rate fell off rapidly, indicating a high rate of initial "plastering" and the formation of a dense dirt bed on the surface of the element which offered a high resistance to the flow of oil.

The second method of determining filter life incorporated operation on a 100-hp diesel engine using straight mineral oil, a temperature of 200 F, shunt installation of the filter, and a pressure drop of 20 psi across the filter element. A typical set of data taken from an engine test is shown in Fig. 6. The minimum required life up to a solid content of 0.3%*w* was 225 hr. In this test, the capacity of the engine lubricating oil pump was approximately 13 gpm. Some of the filters tested, when new, passed this 13-gpm flow at less than 20-psi pressure drop. The pressure drop, therefore, increased up to 20 psi with a constant flow rate, after which the pressure drop was held constant and the flow rate allowed to decrease as the filter accumulated solids.

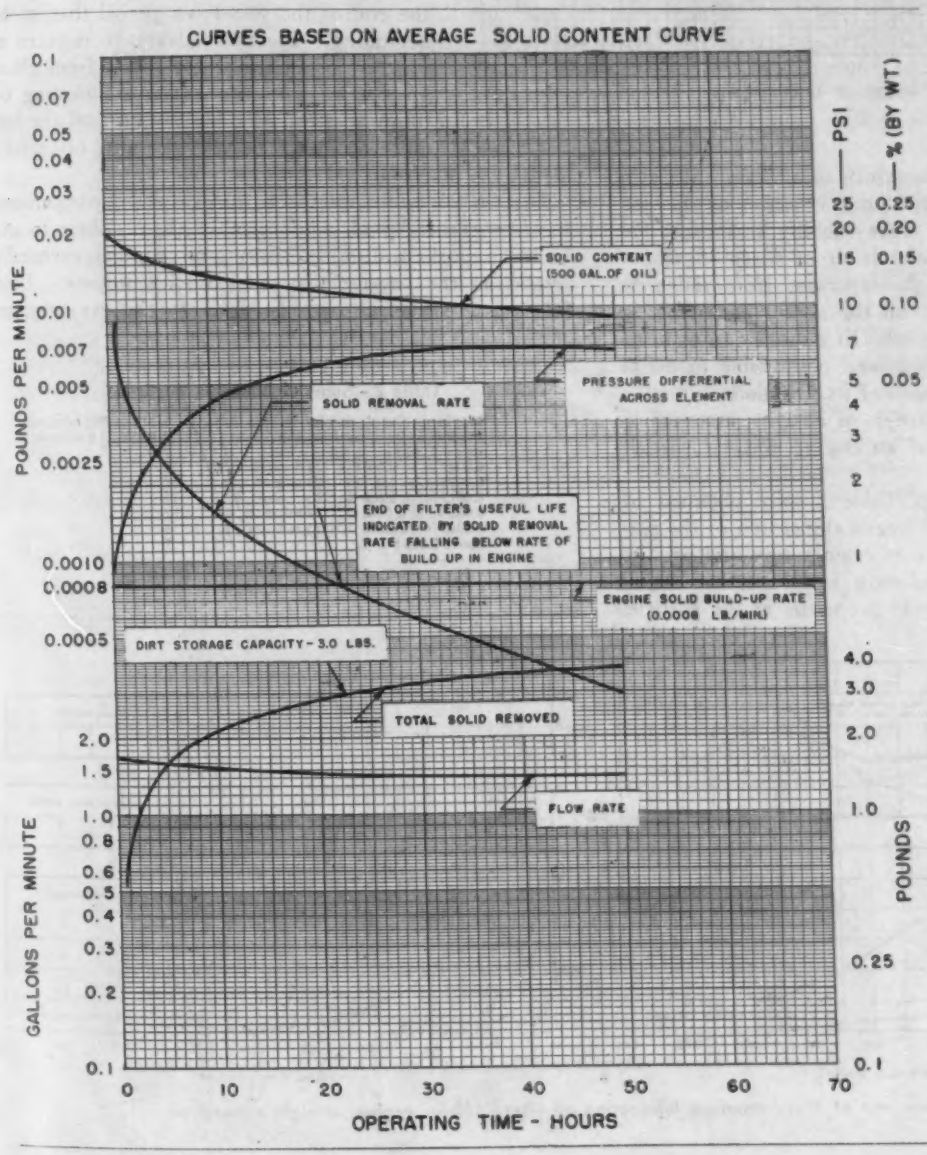


■ Fig. 3 – Typical Navy standard fuel oil filter elements

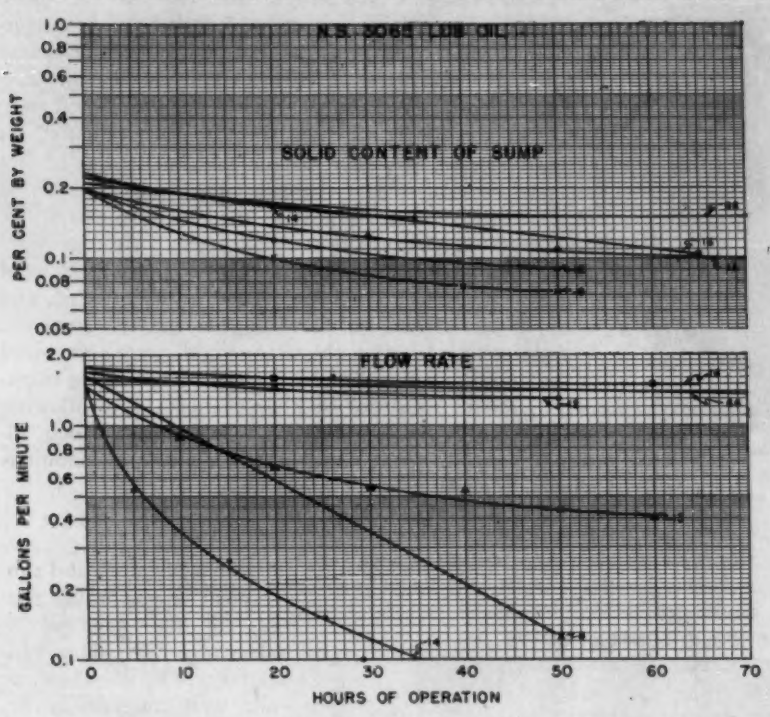
Results of engine tests on four filter elements are shown in Fig. 7. These four elements were of the same construction as like-numbered elements tested by the batch clean-up method, the results of which are shown in Fig. 5. Elements 8 and 16 gave excellent performance in both tests. Filter 13 performed better on the batch clean-up test. It displayed rapid reduction in flow rate on the engine test and at 140 hr started to channel, allowing the oil to pass unfiltered through the cracks in the filter medium. The differences in performance between the two tests were partly attributable to nonreproducibility in filter element manufacture, which has subsequently been corrected, and the inherent differences between the two test methods. The differences between the test methods were recognized at the outset and considered beneficial by indicating markedly different properties of filter elements, thus allowing the selection of elements which would give the best all-around performance under the extremely diverse conditions which are encountered in naval service.

■ Filtration of Additive-Type Oils

During the period of filter-size standardization and subsequent type-approved testing of the standard-size elements, heavy-duty (additive-type) diesel-engine oils were successfully developed to a stage which allowed the Navy to change from straight mineral oil to the newer and better additive-type oils for all diesel-engine use. Assuming that all such oils perform in the manner desired, there still is the need for protection of moving parts against excessive wear caused by abrasive materials. A further requirement is the prevention of stoppages in oil passages due to deposition of the solid contaminants from the oil. This means the elimination of abrasive solids from the lubricating oil as well as from the fuel oil. The definition of abrasive solids depends on the kind of surfaces to be protected, the allowable rate of wear or abrasion, and the type of wear allowed; that is, uniform polishing or lapping action, as compared to roughened or scratched surfaces.


CURVES BASED ON AVERAGE SOLID CONTENT CURVE
POUNDS PER MINUTE
GALLONS PER MINUTE
PSI
% (BY WT)
POUNDS
SOLID CONTENT (500 GAL. OF OIL)
PRESSURE DIFFERENTIAL ACROSS ELEMENT
SOLID REMOVAL RATE
END OF FILTER'S USEFUL LIFE INDICATED BY SOLID REMOVAL RATE FALLING BELOW RATE OF BUILD UP IN ENGINE
ENGINE SOLID BUILD-UP RATE (0.0008 LB./MIN.)
DIRT STORAGE CAPACITY - 3.0 LBS.
TOTAL SOLID REMOVED
FLOW RATE
OPERATING TIME - HOURS

■ Fig. 4 – Accelerated bench test of Navy standard lubricating oil filter element



■ Fig. 5 – Results of tests of five filters on batch clean-up test – capacity of sump: 500 gal; size of orifice: 1/8 in.; pressure drop (across element and orifice): 20 psi; oil temperature: 165 F; contaminant: lamp black

Particle size – Filter performance is affected chiefly by particle size. Size, in turn, is one criterion of the particle's ability to abrade or wear rubbing surfaces. The hardness of the particle, the number of sharp edges and crystalline structure, although important from the wear standpoint, have little effect on the ability of a filter to remove the particle from the oil. In a discussion of filtration, therefore, it is convenient and permissible to define an abrasive particle on the basis of its size alone.

An analysis of a typical sample of abrasive material removed from the fuel line of an engine showed particle sizes as given in Table 2.

The dimensions shown in Table 2 were obtained by microscopic inspection. The largest dimension of the particle was used because of ease in examination and because it was considered to be the most important dimension affecting the scratching or wear properties of the particle.

This analysis, together with many other such determinations indicates that, to be in the abrasive range, a particle will have a size of about five microns (0.0002 in.) or greater.

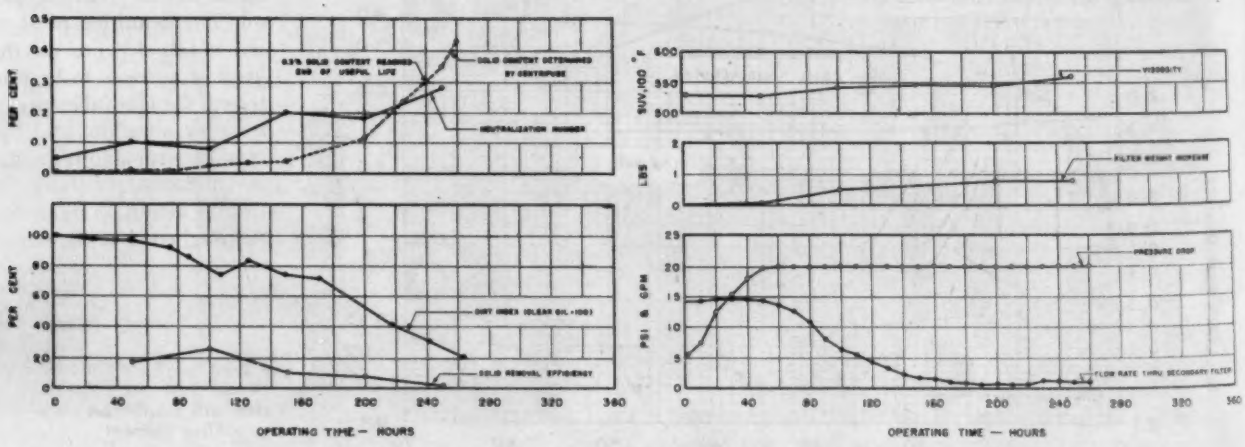
Solids Deposition in Engine – The size of a particle markedly influences its rate of deposition in the oil-flow passages of an engine. Such deposition is chiefly a sedimentation process. The rate of settling is increased by low flow rates and by abrupt changes in the direction of flow. Therefore, in an engine which has been secured for a period of time, the oil will contain less abrasive material at the end of the shutdown period than at the beginning. Practical advantage is taken of this fact by installing the outlet from a fuel oil storage tank or engine lubricating oil sump at a considerable distance off the bottom to allow natural cleaning or gravity settling to take place.

Certain contaminants, such as oxidized hydrocarbons, have properties which tend to make them adhere to and build up on the walls of lubricating oil passages, eventually restricting the passage way an excessive amount. Used lubricating oil which appears clean to the eye may, on

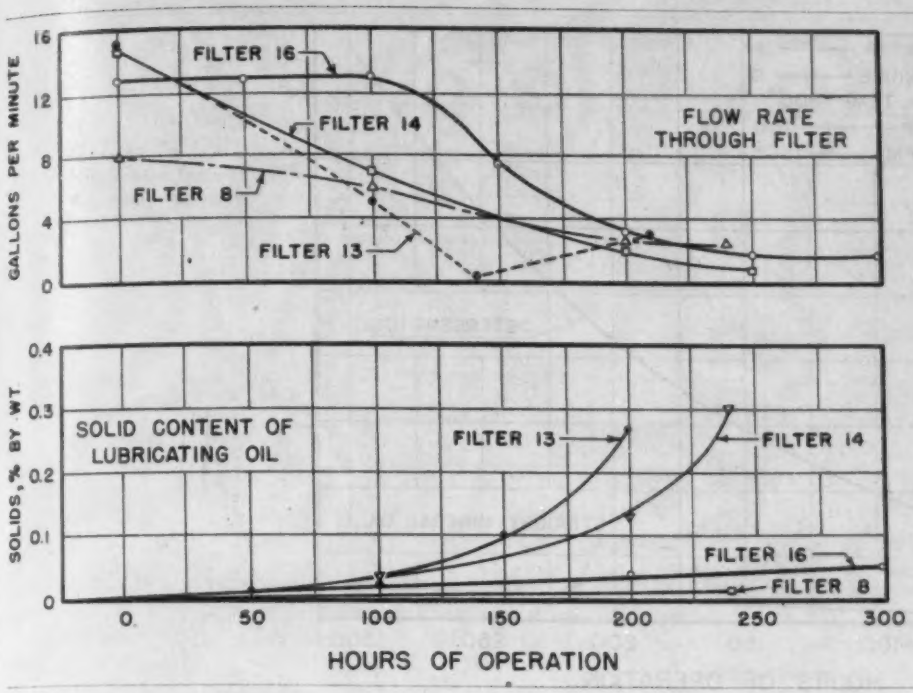
Table 2 – Size[a] of Abrasive Particles

Maximum size of particles, in.	0.013 (330 microns)[b]
Minimum size of particles, in.	0.00006 (1.5 microns)
Below 0.0002 in. (5 microns), %	2
0.0002 in. to 0.0005 in. (12.5 microns), %	18
0.0005 in. to 0.0015 in. (38 microns), %	37
0.0015 in. to 0.003 in. (76 microns), %	22
0.003 in. to 0.0045 in. (114 microns), %	12
0.0045 in. to 0.006 in. (153 microns), %	5
Above 0.006 in., %	7

[a] Size determined from greatest dimension of particle.
[b] One micron = 0.001 mm = 0.0000394 in.



■ Fig. 6 – Engine test of Navy standard lubricating oil filter – 100-hp engine; straight mineral oil

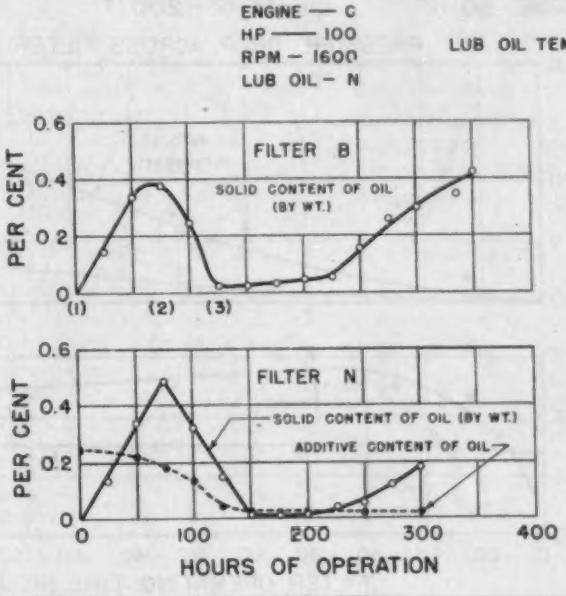


■ Fig. 7 – Life of various filters on 100-hp engine – straight mineral oil; oil temperature: 200 F; volume of each filter: 1000 cu in.

standing, have formed in it, dark-colored, resinous, contaminants which settle out or adhere to the sides of the container. These materials may form deposits on crankcase walls, push-rod galleries, valve gear, under sides of pistons and oil passages, especially when the engine is secured frequently for lengthy intervals. They are seldom washed off by the flow of hot oil during subsequent engine operation and thus never reach the filter where they should be collected. Samples of both straight mineral and detergent-type oils obtained from the discharge side of filters have displayed this characteristic.

Those particles which most easily form deposits inside the engine by the process of sedimentation exist in the size range of abrasive particles and are relatively easy to remove by filtration. The smaller adhesive or sludge-forming contaminants are more difficult to remove by filtration. It is this latter type of contaminant which can be peptized by suitable compounds added to the lubricating oil and thus held in suspension with little tendency to settle out or adhere to the inner surfaces of the engine. This, of course, enhances the difficulty of removing such suspended material by filtration, but the inability to filter these particles is of no consequence, provided the material does remain peptized and suspended in the oil.

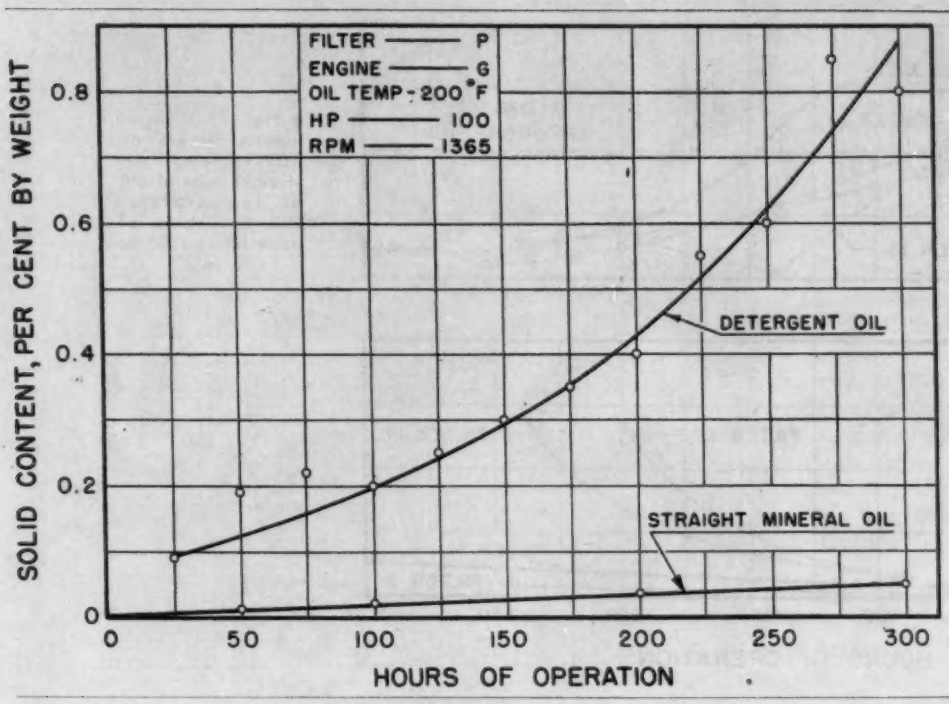
Effect of Detergents on Filtration – Fig. 8 shows the results of attempts to filter an oil to which a detergent compound was added. The solids content of the oil increased rapidly for approximately 75 hr of operation in the engine. During this period, the greater portion of the contaminant, consisting chiefly of carbonaceous material, was passing



(1) ADDITIVE CONTENT - 0.20%
(2) ADDITIVE CONTENT - 0.14%
(3) ADDITIVE CONTENT - 0.04%

NOTE:- ADDITIVE CONTENT INDICATED IS PER CENT BY WEIGHT OF DETERGENT COMPOUND'S METALLIC RADICAL. ACTUAL CONTENT IS HIGHER THAN THAT GIVEN.

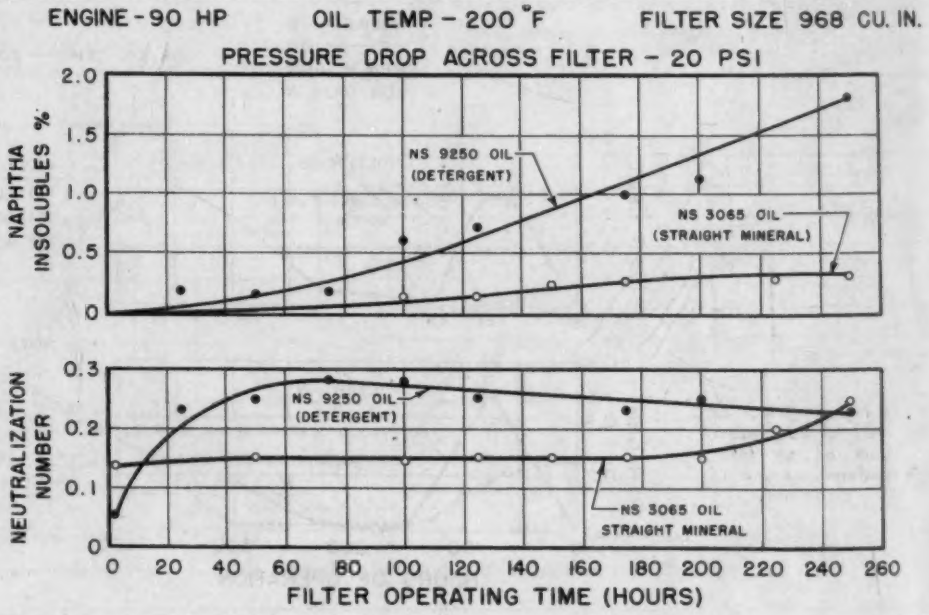
■ Fig. 8 – Effect of loss of detergent from oil on filter performance and oil cleanliness



■ Fig. 9 – Effect of straight mineral and detergent-type oils on performance of filter

through the filter. The amount of detergent compound, indicated by the amount of the metallic radical present, was decreasing during this period. At about 75 hr, the detergent was not sufficiently effective to keep the contaminant peptized and the particles coagulated or grew in size, enabling the filter to remove them. The oil was then cleaned up by the filter and remained in this clean condition until the filter plugged, after which the solid content of the oil increased at a rapid rate. The useful life of the detergent compound as indicated by these tests was short, probably because the lubricating oil consumption was exceptionally low, thus preventing a satisfactory rate of renewal of the additive compound with new, make-up oil. This is one and perhaps the only beneficial result of high lubricating oil consumption.

Tests have indicated that finely powdered silica, fuller's earth, carborundum, and aluminum can be removed from a detergent-type oil with approximately the same ease as from a straight mineral oil. This is not true of the carbonaceous contaminants resulting from incomplete combustion of fuel oil and breakdown of the lubricating oil and may not be true of abrasive materials other than those investigated. Figs. 9 and 10 show the relative ability of filters in removing such contaminants from detergent-type and straight mineral oils. In this respect, each kind of detergent produces different results when an attempt is made to filter it. One type of heavy-duty, detergent oil now commercially available can be cleaned with relative ease with ordinary filters, while some other types can be cleaned only after the addition of a second compound to



■ Fig. 10 – Filter performance with straight mineral and heavy-duty (detergent-type) diesel lubricating oils

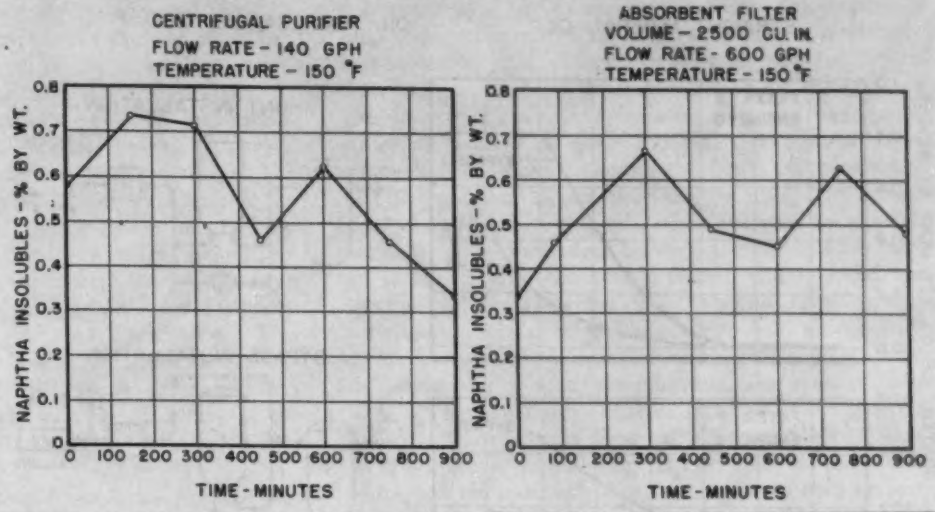

CENTRIFUGAL PURIFIER
FLOW RATE - 140 GPH
TEMPERATURE - 150 °F
NAPHTHA INSOLUBLES - % BY WT.
0 0.1 0.2 0.3 0.4 0.5 0.6 0.7 0.8
0 100 200 300 400 500 600 700 800 900
TIME - MINUTES
ABSORBENT FILTER
VOLUME - 2500 CU. IN.
FLOW RATE - 600 GPH
TEMPERATURE - 150 °F
NAPHTHA INSOLUBLES - % BY WT.
0 0.1 0.2 0.3 0.4 0.5 0.6 0.7 0.8
0 100 200 300 400 500 600 700 800 900
TIME - MINUTES

Fig. 11 – Results of attempt to clean 50-gal batch of highly detergent lubricating oil – oil taken from engine after 560-hr of use

neutralize the peptizing effect of the detergent. This procedure is only practical in laboratory analysis where a determination of the total amount of solids contained in the oil is desired.

Determination of Solids Content

Fig. 11 shows the results of attempts to clean batches of highly detergent oil with a centrifugal purifier and a depth-type filter. The centrifugal purifier apparently removed more solid material than the filter, but the data are sufficiently erratic to prevent rational analysis. These data were obtained gravimetrically using a Gooch crucible, an asbestos filter bed, and standard precipitation naphtha. They indicate how unsatisfactory this method of determination can be. Since these tests were made, a long series of experiments has led to the adoption of a centrifugal method of determining solid content which produces results sufficiently good for the majority of filter tests. The method now in use is described in Appendix I. In conjunction with this method, a rapid indication of the solids content of the oil is obtained by means of a modified light transmission test in which the reduction in light intensity resulting from passing the light through a strip of Whatman's filter paper soaked in new lubricating oil is compared with the reduction in light from the same source passing through a similar paper strip soaked in the used oil under test. The ratio of these two reductions in light intensity multiplied by 100 is called the dirt index. Details of the method are given in Appendix II.

Abrasive Removal Performance of Filters

After analyzing the results obtained by means of engine and batch clean-up tests using straight mineral oil and after various exploratory tests of filters using additive-type oils, a new test for filtering performance has been adopted to replace both of the tests previously used. The batch clean-up test will be used in the future only for the purpose of eliminating those filters having highly objectionable features to save time for testing the better filters by the longer and newer method. This new method is essentially a recirculation process simulating engine operation. It is hoped that more reproducible results will be obtained by eliminating the uncontrolled variables inherent in engine operation and allow selection of filters which will give optimum protection of the engine from abrasive contaminants in the lubricating oil. The system will contain 16 gal of detergent oil, which will be recirculated continuously through the filter under test and back to an oil sump. The flow rate will be the maximum the element will pass at a pressure drop of 20 psi. The test will start with clean oil which will be contaminated at a constant rate of 0.0162 lb per hr with a mixture of carbon black, iron oxide, and asphaltenes in the following proportions:

Carbon black, lb	2.40
Iron oxide, lb	0.24
Asphaltenes, lb	0.60

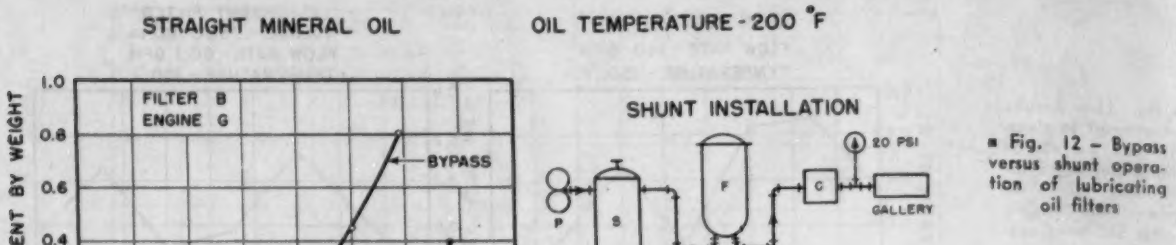
The carbon black will simulate soot from the combustion chamber and particles of highly oxidized lubricating oil; the iron oxide will simulate abrasive material; and the asphaltenes will simulate the lightly oxidized fractions of used lubricating oil. Oil samples will be taken periodically and analyzed for iron oxide content. The amount of iron oxide present in the oil at the end of 200 hr will be the criterion of filter performance.

It is recognized that this test will cause considerable discussion because of the desire of many people to remove the carbonaceous material from used oils. The test is not an argument in favor of allowing such material to remain in the oil, but it is a recognition of the fact that, with commercially available filters, such material cannot be removed from many of the detergent oils now being used without removing or otherwise destroying the additive.

Other Tests

It is necessary that a filter not only perform the functions required, but it must also satisfactorily withstand harsh treatment and in no way be deleterious to the lubricating oil passed through it. Several different methods have been devised to determine these characteristics.

Shock Test – In order to determine if both fuel and lubricating oil filter elements will withstand shock due to gunfire or bombing attacks, the element under test is mounted in the normal manner on a standpipe which is rigidly fastened to the panel of a shock test machine. The panel is struck twenty 300 ft-lb blows by means of a 50-lb hammer dropping through a distance of 6 ft. Elements



■ Fig. 12 – Bypass versus shunt operation of lubricating oil filters

which fail due to these hammer blows are considered unsuitable for naval service.

Water Immersion Test – Some materials used in filter elements disintegrate rapidly when in contact with water, tending to mat or cake and reduce the flow rate. In order to measure this effect and to determine the suitability of elements which may be required to operate under adverse conditions with water present in the oil, the following test has been devised:

(a) Hot oil is passed through the element at various pressure drops and the corresponding oil-flow rates determined.

(b) The element is next allowed to drain for 1 hr, after which it is immersed in a sea-water bath for 16 hr at 100 F.

(c) After removal from the water bath and draining for 1 hr, it is retested with hot oil for flow characteristics.

Excessive changes in flow characteristics render the filter unsuitable for naval service.

Migration of Filter Media – Although the chief function of a filter element is the removal of foreign material from the oil, poorly constructed elements may allow some of the material composing the element to pass into the lubricating oil stream. Such foreign material has actually caused engine failures by collecting in the lubricating oil passages and stopping the flow of oil. In order to eliminate such failures, all large lubricating oil filter element types submitted are subjected to a 200-hr test in which the element is mounted in a filter case containing an internal bypass. Hot oil is circulated through the filter case at a pressure drop of 30 psi. Part of the oil passes around the outside of the element and through the bypass. The remainder of the oil passes directly through the element. The oil flows from the filter to a 0.001-in. spacing, metal-edge strainer, which collects on its surface any of the filter media removed from either the inlet or discharge sides of the element. Fuel oil elements and small lubricating oil filter elements are tested in the same manner, except that the bypass is eliminated.

Effect on Additive-Type Oils – Many of the materials commonly used in the construction of filter elements have adsorptive characteristics which cause them to remove some of the detergent and other types of compounds originally added to the lubricating oil. The same characteristics formerly caused such filters to remove undesirable oxidized hydrocarbons from straight mineral oil. These characteristics are determined by circulating clean, additive-type oils through the element at a temperature of 160 F and a pressure drop of 20 psi for 100 hr. The percentage reduction in the additive content of the oil is determined by analysis of samples obtained periodically during the test. Because of the impracticability of obtaining materials which have no effect on the additive, a removal of 30% is allowed. This applies to all additive-type oils approved for naval service. Some additives are relatively easy to remove even with commercial cotton waste.

Clean, new, lubricating oil is used in this test rather than used, dirty oil. It is not considered that the filter should be penalized for its ability to remove compounds which may be attached to the solid particle. Furthermore, reproducibility of results, using dirty oil, is difficult to obtain because of the wide variations in the power of different carbon particles to adsorb additive compounds.

■ Installation of Lubricating Oil Filters

Shunt Method – Two methods of connecting filters into an engine lubricating oil system are illustrated in Fig. 12. The shunt method of installation is sometimes also referred to as the full-flow or 100% flow system. In this system, all of the oil from the pump passes to the engine oil gallery. The filter is connected into the system in parallel with an automatic pressure relief or bypass valve, which maintains the pressure drop across the filter elements at a constant, fixed value. Naval installations require this relief valve to be inside the filter case. This aids in obtaining rapid warm-up and high oil filtering temperatures.

Large Navy standard lubricating oil filters are not re-

stricted to any one system, but the system used must be approved by the Navy Department's Bureau of Ships. Shunt installations or variations of the shunt installation predominate.

Bypass Method – The bypass method of installation, sometimes referred to as the bleeder system, bypasses a restricted quantity of oil from the pressure side of the lubricating oil system back to the suction side, that is, the oil sump or crankcase. Thus, the amount of oil reaching the oil gallery and ultimately, the bearings, is reduced by the amount flowing through the filter. It is necessary to restrict this flow to a small percentage of the total pump capacity to ensure that the engine is not robbed of oil. This is usually done by placing a metering orifice in the discharge from the filter. All small Navy standard filters have this metering orifice built into the standpipe of the filter case. Such filters, must, therefore, be installed in a bypass system.

High Flow Rate versus Low Flow Rate – For the purpose of maintaining the solid content of the oil at the lowest possible value, it is important that the flow through the filter elements be maintained at the highest practicable rate. A comparison of the performance of two radically different types of large Navy standard size filters, each connected in shunt and in bypass is shown in Fig. 12. A ⅛-in. diameter orifice was placed in series with the filters when connected bypass. In each case, the life of the filter was longest when shunt connected. This means that the rate of solid removal is chiefly dependent on the flow rate through the filter element. It is generally believed that a reduction of flow through a filter results in an increase in solid removal efficiency sufficient to overcompensate for the decreased flow rate and thus produce a net increase in rate of solid removal. Results of tests of fuel oil filters using a contaminant having a particle size down to three microns indicates that such a belief is fallacious. In these tests, no discernible change in the solid removal efficiency was noted, although the flow rates investigated covered a range of five to one.

A second, rather common belief is that high flow rates cause excessive "plastering" of the filter element. To investigate this, two runs were made with elements consisting of dense, absorbent material of uniform consistency, which could be sectioned after use for inspection of the dirt penetration. The element subjected to a pressure drop of 20 psi produced a higher average flow rate, displayed a longer life, and removed the greatest amount of solids from the oil. The inspection also showed the greatest depth of dirt penetration in the high-flow element.

Number of Filter Elements Required – The number of filter elements required for a given engine is a question best solved by trial, except that tests on new engines seldom indicate the filter capacity required for the same engine after it has been run for some time and has reached either an average or poor mechanical condition. Data obtained from many thousands of hours of operation of 16 different 2- and 4-cycle diesel engines using both straight mineral and additive-type oils, when analyzed by statistical methods, show correlations between the rate of solid build-up in the oil and the bore diameter, engine speed, and specific fuel consumption. Figs. 13 and 14 show these correlations. No attempt was made to hold all conditions constant except the one under investigation; thus, to use the curves for a specific application, the rates of solids build-up for the three variables listed must be added together and the average obtained. The correlations shown were utilized in formulating requirements for the number of large Navy standard filter elements to be used on naval diesel engines. These requirements are as follows:

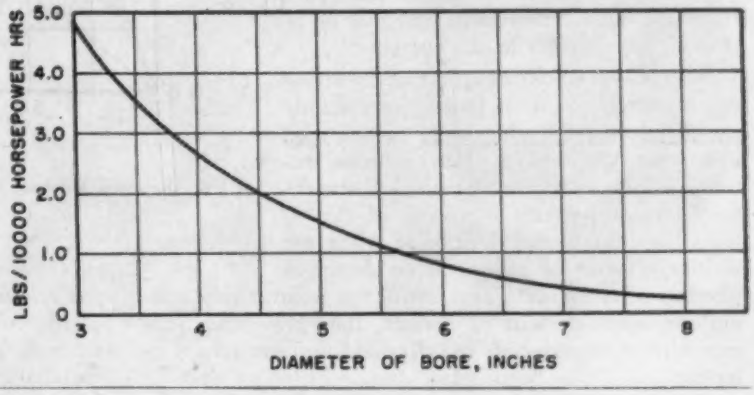
Number of Elements	Horsepower of Engine
8	2000-1400
6	1400-900
4	900-500
3	500-300
2	300-150
1	150-100

■ Rate of Oil Solids Production in Engine

Type of Lubricating Oil – The effect of type of lubricating oil used on the rate of solids build-up in the oil is highly important. Fig. 15 shows the results obtained with both straight mineral oil and various additive-type oils in two different types of 100-hp engines. The bore of engine *C* was ⅜ in. larger than the bore of engine *G*. It was also operated at 1600 rpm, as compared to 1350 rpm for engine *G*.

The solids build-up in the detergent-type oils is indicated to be lower than in the straight mineral oils. The accuracy of solids content determinations on the detergent oils was probably lower than for the straight mineral oils. It will be noted that the rate of solids production tends to decrease with time. It is probably that this is caused by solid material settling out of the oil inside the crankcase and oil passageways.

Engine Condition – The mechanical condition of an engine has an important influence on the rate of oil solids production. One important factor is condition and type



■ Fig. 13 – Effect of cylinder-bore diameter on production of solids in lubricating oil of diesel engines – curve is based on average data from 16 2- and 4-cycle engines using additive-type and straight mineral oils

of piston rings. Fig. 16 shows the change in the rate of solids production caused by changing the type of piston rings used. Several hundred hours of operation with each type of ring proved the superiority of the modified type in keeping the lubricating oil clean.

■ Defective-Construction Failures

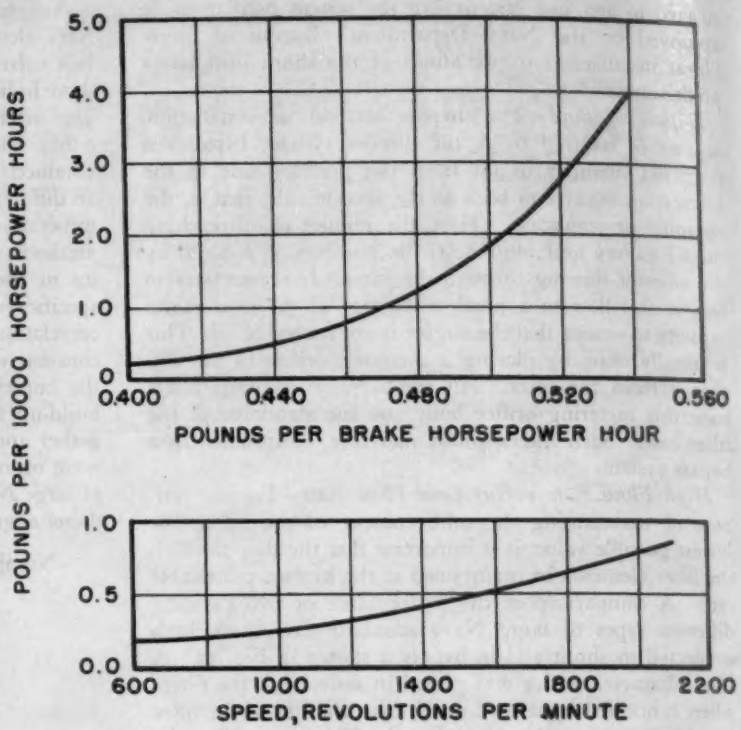
Failures of equipment, especially when in the experimental stage, are quite prevalent. In type-approval tests of filter elements, an endeavor is made to determine those defects in design and construction which may result in casualties in service. Several defects which have subsequently been corrected are illustrated.

Failures of Filter Element Center-Tubes – Fig. 17 shows failures in three different center-tubes. These center-tubes were all subjected to a compressive stress equivalent to that resulting from a pull-down torque of 20 ft-lb on a 1⅞-in. diameter, 12-threads-per-in. sealing nut.

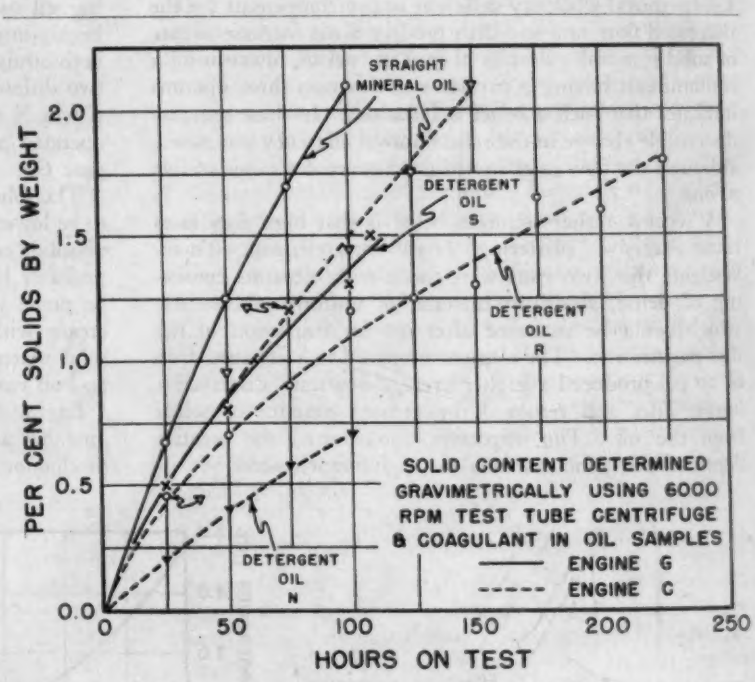
End-Seal Gasket Failures – Fig. 18 illustrates the changes occurring in ground-cork sealing gaskets due to unsuitable binder. Such gaskets take a permanent set, are reduced in diameter, and become brittle and hard. These gaskets, when cemented to the end plate of the element may fail as shown in the right-hand photograph. After the cemented joint was broken, probably due to the hot oil, the pressure differential across the element forced the loosened side radially inward until a free path was provided for the oil to bypass around the element.

Migration of Filter Media – Fig. 19 illustrates the type of material which was caught on the surface of a metal-edge strainer placed in the discharge line from a filter. The filter was constructed to include a spring-loaded relief valve inside the case. In this test, the flow rate was maintained at a sufficiently high value to cause this valve to remain open, thus allowing a portion of the entering oil to bypass the element into the discharge line. The fibers shown came from the outside of the element by way of this bypassed oil. Such fibers are highly effective in plugging or restricting oil passages in the engine.

Filter Element Shrinkage – Fig. 20 shows approximately ¾ in. of longitudinal shrinkage that occurred during 50 hr of operation in another type of filter. Many cellulose materials exhibit such characteristics. If the element design prevents bypassing of the oil from the inlet to the outlet side when the shrinkage occurs, there will be no change in filtering performance. This shrinkage is invariably nonuniform from element to element, thus preventing procurement of reproducible test data and uniform results in service.

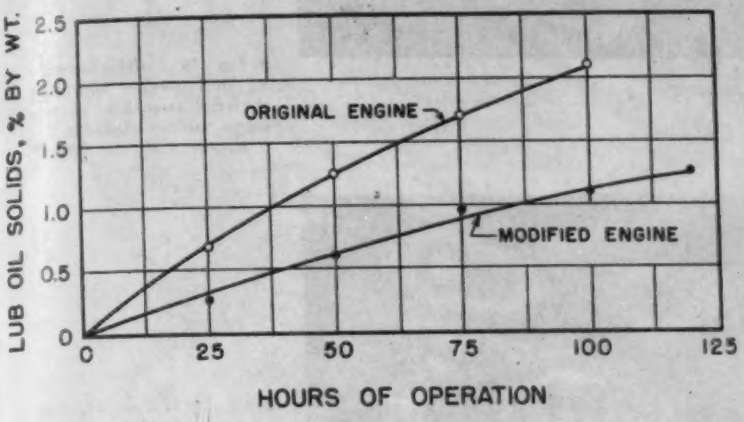


■ Fig. 14 – Effect of engine speed and specific fuel consumption on rate of production of solids in lubricating oil of a diesel engine – data from 16 2- and 4-cycle engines using straight mineral and additive-type oils



■ Fig. 15 – Solid build-up in straight mineral and detergent-type lubricating oils in unfiltered engines

Enclosing the Filter Media – The cotton sock which completely covers the element shown in Fig. 20 was missing from the element shown in Fig. 21. The fibers of filtering material protruded through the perforated metal container



■ Fig. 16 – Effect of modifications in engine construction on rate of production of solids in lubricating oil of unfiltered engine

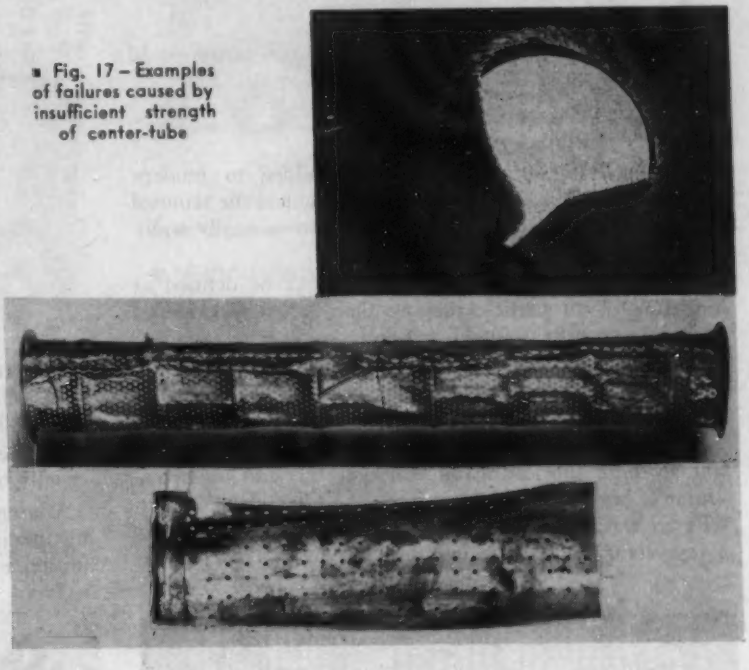
and were easily carried off by the oil passing through the relief valve, as previously discussed.

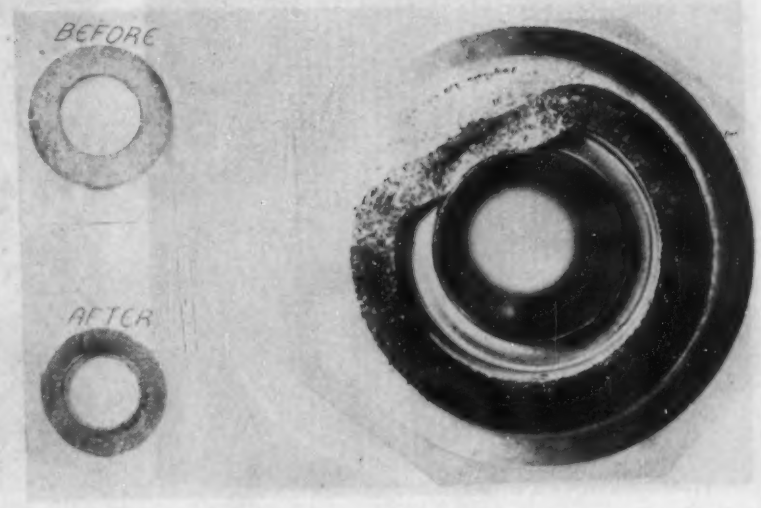
Nonreproducible Filter Media – Stories are numerous about emergency methods of manufacturing filter elements. They include the one which tells of repacking the element container with toilet paper, which performed better than the original filtering medium. Fig. 22 illustrates what may be found in some types of filter elements. This filter medium included a piece of curtain cord, several pieces of cotton underwear, two pearl buttons, a length of cheese cloth, a piece of cellophane from a cigarette package and a piece of blue denim. It is obvious that such material cannot be exactly reproduced from element to element; so that attainment of uniformity in performance is impossible.

Channeling – The flow path through a filter case can adversely affect the life of an element, as shown in Fig. 23. This element was removed from a filter case which had the inlet near the top of the case. When started from cold, a large volume of cold, viscous oil existed in the lower part of the case. The hot, entering oil tended to remain near the top, thus utilizing only the top portion of the element. The photograph shows the upper third of the element completely filled with dirt, but the lower portion only lightly coated. The obvious correction is to place the inlet near the bottom of the filter case.

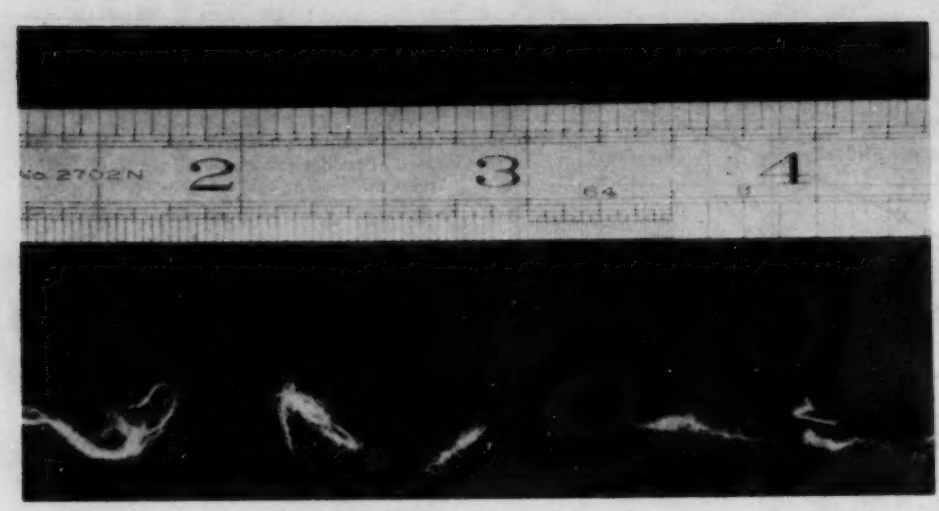
Certain types of elements are constructed by packing several separate and distinct lots of the filtering medium, one on top of the other, into the container. No attempt is made to obtain a continuous and uniform mass. Each separate lot, if shrinkage occurs, will shrink in such a manner that channels will be opened up between the separate lots of filtering medium. An example of this channeling is shown in Fig. 24. When this occurs, the flow rate increases and solid material previously removed may be rapidly unloaded into the oil to the engine in relatively

■ Fig. 17 – Examples of failures caused by insufficient strength of center-tube



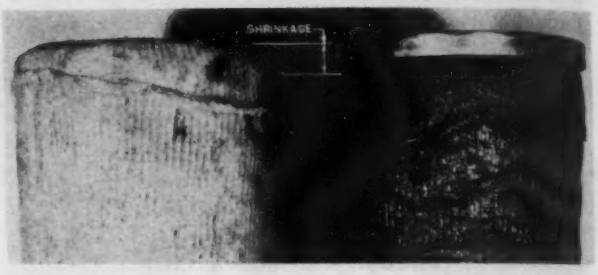


■ Fig. 18 – Effect of use of unsuitable gasket design and material



■ Fig. 19 (left) – Example of filter medium migration material caught on metal-edge strainer placed in discharge line from filter

■ Fig. 20 (below) – Shrinkage of depth-type filter element after 50 hr of use



large quantities. Engines equipped with such filters would probably be better off without any filter at all.

■ Conclusions

The peptizing effect of detergents added to modern lubricating oils, in general, renders impractical the removal of carbonaceous material with filters commercially available.

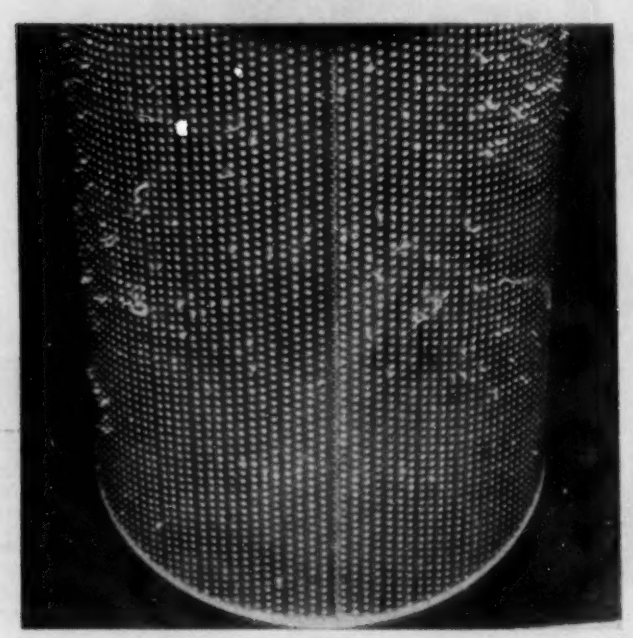
Abrasive material in lubricating oil may be defined as consisting of all particles having their larger dimensions equal to 0.0002 in. (5 microns) and greater.

Abrasive material can be maintained at the lowest concentration by circulating the lubricating oil through the filter element at the highest practical rate.

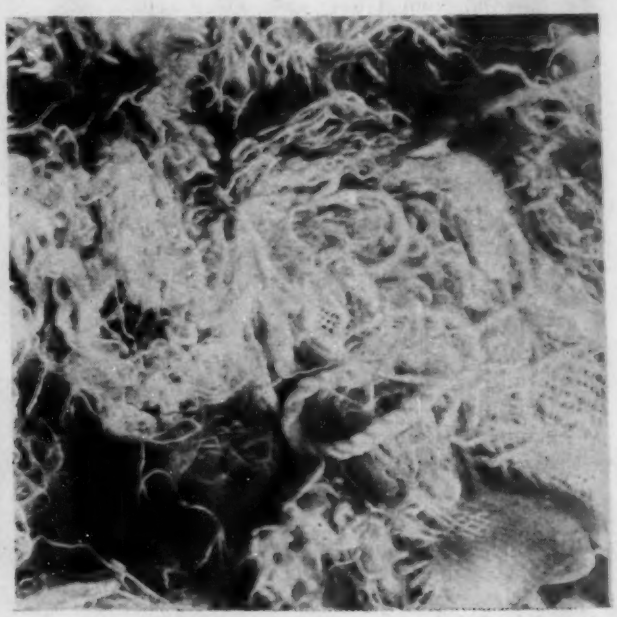
High filtering flow rates can be obtained by shunt operation, by high lubricating oil temperatures, and by maintaining a high pressure drop across the filter element.

Rapid warm-up of the lubricating oil in the filter body is important for intermittent-duty service. This can be aided either by attaching the filter body to the engine or by installing it in close proximity to the engine and passing through the filter case all or a large portion of the total oil flow, while maintaining a safe pressure drop with an internally built relief valve.

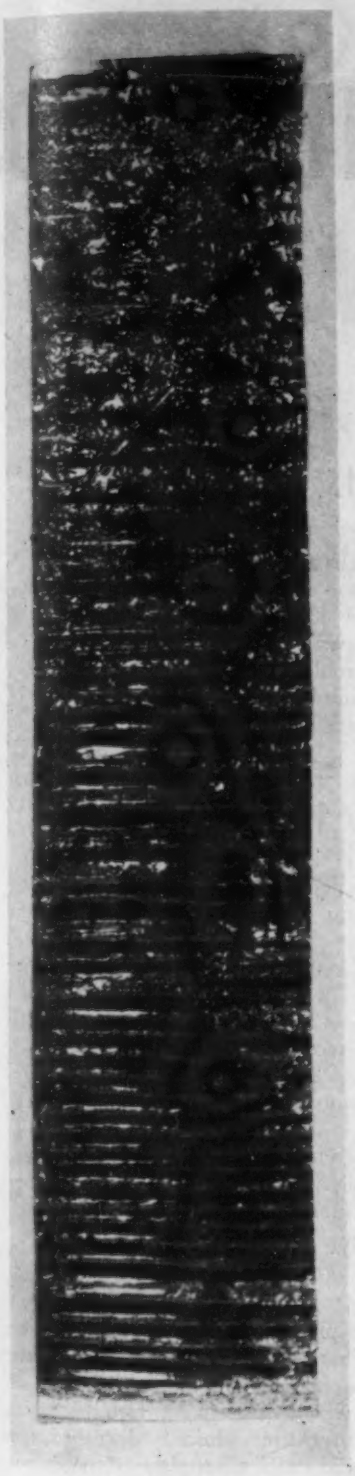
A large portion of the solid, carbonaceous material existing in diesel lubricating oil results from incomplete combustion of the fuel oil. High specific fuel consumption,



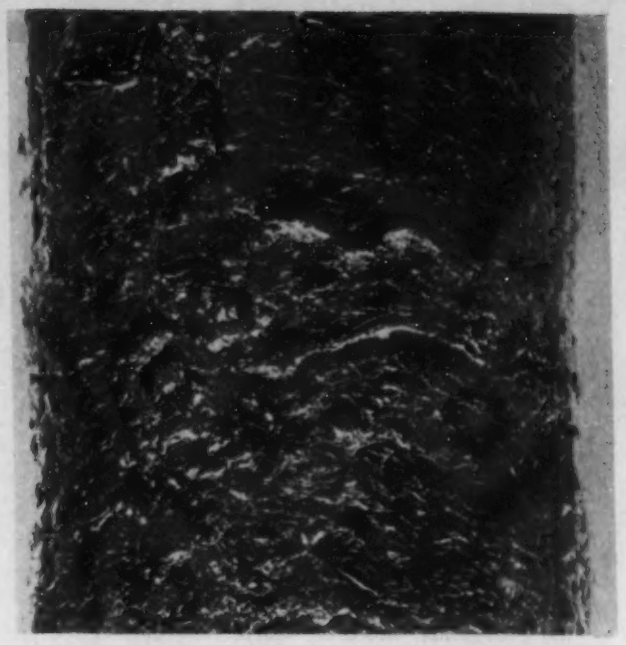
■ Fig. 21 – Construction that allows filtering material to pass into engine lubricating oil system



■ Fig. 22 – Type of filter medium causing nonreproducible performance



■ Fig. 23 (left) – Vertical section of large filter element showing large amount of dirt removed at top (hot) end of element and small amount removed at bottom (cold) end due to unsatisfactory circulation of oil in filter body



■ Fig. 24 (right) – Example of channeling in depth-type filter element

therefore, tends to increase the rate of formation of solids in the lubricating oil.

For practical test purposes, it is only necessary to have a method of oil solids content determination which is reproducible and which will remove more of the solids than any of the filters being tested. There still is definite need, however, for rigid definitions of the important types of solid contaminants found in lubricating oil and standardization of accurate, reproducible methods for the quantitative determination of such contaminants.

■ Acknowledgments

The author desires to express his appreciation to Roger Walsh, who prepared the charts and graphs; to the personnel of the Internal-Combustion Engine Laboratory for their assistance in procuring data; to the various filter manufacturers for permission to use photographs of their products; and to W. F. Joachim for his interest and encouragement in the preparation of this paper.

APPENDIX I

Procedure for Gravimetric Determination of Solids Content of Lubricating Oil by Centrifugal Method

(Method used at U. S. Naval Engineering Experiment Station during tests to evaluate performance of lubricating oil filters)

1. Heat bottle containing sample of oil to be tested to 150 F.

2. Mix sample by shaking bottle in mechanical shaker for 5 min.

3. While shaking sample, pour 2 ml of 2% solution of N-butyl diethanolamine in methyl alcohol into tared 100 ml centrifuge tube. Add 48 ml of precipitation naphtha to bring the contents of the tube to 50 ml.

4. Add 45 g of the oil to be tested to the mixture in the tube.

Note: This should be done immediately after the sample bottle is removed from the shaker.

5. Shake tube for 3 min in mechanical shaker.

6. Whirl the tube in a 6000 rpm, 10,000-*g* centrifuge for 30 min or longer until clear supernatant liquid is obtained. Maintain the temperature at 100 F during this process.

7. Decant the supernatant liquid, but allow a sufficient

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RECENT TRENDS in

by LEWIS A. RODERT*

THREE types of systems are used to give protection against the formation of ice on airplanes: mechanical, chemical, and thermal; the thermal systems being the most successful and the most widely used.

The most satisfactory method of ice prevention, for instance, in that most essential accessory, the carburetor, is by heating the carburetor walls and throttle.

Similarly, surface heating appears to be the best way of protecting the propellers, wings, and empennage from the formation of ice.

Heated air flowing in the passage between the panes of a double windshield is also most successful for keeping the pilot's windshield clear.

Despite the use of such improved methods of ice prevention, however, full benefit cannot be realized from their use until radio facilities and blind landings have been developed further, for flights must now be cancelled because of limited visibility even if icing conditions do not prevent flight.

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A STATEMENT was made in a recent news item that, as a result of extensive research on airplanes flown in icing conditions, ice formations have been prevented by heating the wing leading edges with heat from the exhaust gas.

Such a statement taken alone, although of considerable value to a news publication, does not begin to tell the story of the work which has been undertaken to provide dependable and practical ice protection for aircraft, the success of this work, or its effect on airplane operations of the future.

While great gaps must be left unfilled in the complete development of the topic which has been selected for discussion at this time, a reasonable amount of detail can be assembled here which will clarify the problem, define the activities which have been undertaken and which are being continued, outline roughly the results, evaluate the effects which the results will have on aviation, and suggest to some extent the work still to be done.

Classified documents of the NACA and other Government agencies, which fill in the voids left here for security purposes, are available to authorized firms and persons desiring detailed data.

The importance and need for the provision of protection against ice for aircraft changes with new developments in the design of the airplane and its vulnerable parts, in the type of operations to which the airplane is applied, the geographic region of operations, and other factors. The definition of the icing problem of aircraft as it affects operations today, therefore, must be noted with some degree of reservation. With such reservations, the details of the problem will be outlined.

The popular concept of the icing of aircraft is, that under some unusual weather conditions great accretion of ice forms all over the airplane wings and the airplane crashes to earth because of the great increase in weight. The fallacy of this limited impression will be presently seen. In the development of a research program on the icing problem some years ago, the NACA learned from a widely distributed questionnaire that the formation of ice on the wings was considered of lesser importance than other items in the problem by informed airplane operators. Subsequent information received from these operators, and experience gained through four winters of our own operations in natural icing conditions have sustained this concept of the wing icing problem.

It seems desirable, therefore, even at the risk of stating known facts, to redescribe the problem in detail in order that there will be no misunderstanding of what the *protection* should *protect* against.

■ Meteorology

Due to meteorological phenomena, the details of which will not be examined in this discussion, unfrozen water drops occur in the atmosphere at temperatures of 32 F or lower. When moisture which exists in the liquid state at below freezing temperatures is mechanically disturbed, the change of state from liquid to ice takes place to the extent of the degree of supercooling. The water, which remains unfrozen, is also subject to freezing, the extent of which depends upon the degree of evaporative and convective cooling which may occur before the liquid is mechanically removed by the air stream.

If the degree of supercooling is slight (that is, less than 20 F) and the quantity of water in droplet form is large (in the order of 1 g per kg of air), the ice formation will

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in AIRPLANE ICE-PREVENTION TECHNIQUE

probably be a homogeneous-crystalline, roughly shaped mass which is clear and nearly transparent. This type of ice is usually called glaze ice. The individual water droplets do not retain individual identity in glaze-ice accretions.

If the degree of supercooling is greater (that is, in the order of 20 F or more) and the water content in droplet form is about 1/10 g per kg of air or less, the individual droplets will freeze almost instantaneously on impact and form an opaque, white, rime-type ice. The individual droplets will retain, to a large degree, individual identity and considerable air will be entrapped with the accretion.

Pure forms of ice, as defined above, are not often found. The addition of snow or sleet in the formation frequently occurs; and often rime and glaze ice will be encountered intermittently, which makes identification of the accretion difficult. Rime ice, resulting from very small and highly supercooled droplets, will coalesce in time and assume a crystalline, transparent structure resembling glaze ice.

The temperature at which ice forms affects the physical strength and adhesion to the airplane structure. Rime ice, formed at low temperatures and which has had time to coalesce, usually has the greatest strength.

The effects of ice formations on the performance of the airplane are dependent upon the type of ice. Glaze formations will have a greater effect upon an aerodynamic system because glaze ice usually forms more rapidly and grows across the streamlines. Rime ice, in early stages of formation, grows along the streamlines and does not deform the aerodynamic system as quickly as will the glaze type. Given adequate time and water drop content, however, rime ice will cause about the same aerodynamic effects as glaze ice.

Formations resulting from a freezing rain are usually a type of glaze ice. The variations of conditions which can form ice are infinite and just about every imaginable physical possibility does occur, although some manifestations of the problem are peculiar to certain geographic regions. As an example, the accretion of ice at air temperatures below —50 F has been observed in the colder climates. Such an occurrence would not be likely in the central regions of the United States.

The formation of frost and ice, and the collection of snow on aircraft, when the equipment is at rest on the ground, is an important aspect of the ice-prevention problem and must receive special attention.

The severity of icing types and conditions resulting from the meteorological phenomena depends upon the airplane design, the vulnerability of the part, the method of protection involved, the perfection of the protecting device, the skill of the pilot, and other factors, such as the mission and geography.

■ Development and Research Activities

The manifestations of the icing problem, its effect upon airplane operation, and methods of providing protection have received the attention of the NACA, Army Air Forces, Navy Bureau of Aeronautics, Bureau of Standards, U. S. Weather Bureau, NDRC, many leading educational and commercial research agencies, aircraft manufacturers, equipment manufacturers, and airline operators. These extensive activities have resulted in some knowledge of the problem and its solution. Our knowledge on the subject, while somewhat impressive if judged from the volume of literature available, will probably not reach the satisfactory stage until extensive application has been experienced by the commercial and military operators of the new techniques which are currently being discovered and developed.

This discussion of the problem and ice-prevention techniques will be concerned with the carburetor, propeller, wings, windshield, and air-speed equipment. Attention will also be directed to the need for protection of radio antenna systems, engine cowls, and protuberances. These items have been stated in the order of their relative importance according to the opinions of experienced engineers. There seems to be a general agreement on this rating, and the research work has been conducted accordingly.

■ Induction System

As indicated above, in the opinion of experienced operators, the most essential item on the airplane requiring ice protection is the induction system, which consists of the intake scoop, carburetor, adapter (in some installations), and turning vanes at the eye of the engine impeller. It is also believed that the induction system is the most vulnerable component to complete failure due to icing conditions.

Ice formations in the induction system occur from the meteorological phenomena which produce ice on the exterior part of the airplane and, also, from water at temperatures above 32 F when taken into the system in sufficient quantities with certain humidities and throttle settings. The necessary refrigeration results from the adiabatic ex-

pansion effects in high-velocity regions, and from fuel and water evaporative cooling.

The nature of the problem is altered somewhat by the shape of the intake scoop, the carburetor design, the presence of protuberances into the carburetor air stream, and to a greater extent by the temperature of the air-duct walls. The design of the carburetor, therefore, cannot be expected to alleviate completely the problem, although so-called ice-free carburetors have minimized certain types of icing.

The solution to the induction system icing problem is being attempted by the use of heating the air inducted, injecting alcohol into the carburetor, or heating the carburetor walls and throttle. The first method is employed extensively. Recent design trends have been to define more accurately the temperature rise of the inducted air and to provide a more reliable mechanical control valve by which the air temperature is regulated. Additional attention is being given to the use of automatic air temperature controls for the carburetor.

The use of alcohol was quite extensive for a short time, but as objections developed from practical experience, this method is gradually being discarded.

Heating of the structural parts of the induction system has been the most attractive method to many engineers, because such a system will require less attention from the pilot, and will not limit the brake mean effective pressure at which the engine could be operated, as does the heating of the inducted air. Some carburetors employed in England have oil heating of the induction system and are claimed to be reliable. The trend to such a system is not strong in this country and is not likely to become so until the maze of protuberances in the inducted air stream has been reduced or eliminated.

Some carburetor designers attempt to discharge the gasoline into the air stream at a point where the evaporative cooling cannot result in icing. The discharge of the fuel into the eye of the impeller is an example of this trend, and fuel injection in the cylinder is the logical end to which the trend will lead. It must be noted, however, that the complete elimination of the fuel from the air induction system, such as fuel injection will provide, will not solve the induction system icing problem.

It can be expected that improved equipment for carburetor air heating and control will be available in the future and that less attention by the pilot will be required during operations in icing conditions. Inasmuch as operations at the present time are not limited by the carburetor icing problem, it is not expected that future operations in more severe weather, as may be made possible by advances in ice prevention and blind landing technique, will be limited by this problem.

Work remains on the carburetor problem in establishing data and preparing specifications whereby improved carburetor air-heating equipment may be manufactured and operated. Such results should be forthcoming from present activities. The greater task of establishing data for the heating of the walls and throttle of the induction system has not yet been undertaken with sufficient vigor to ensure satisfactory results.

■ Propeller

The airplane operator is consistent when he places emphasis on the propeller icing problem as the second most urgent item. Like the induction system, the propeller collects ice quickly and unseen. The results are immediate and in some cases catastrophic. The effects of ice formations on the propeller cause loss in thrust and engine vibration. Vibration results from a part of the ice on one of the blades being thrown off by the centrifugal force, thus rendering the propeller out of balance and the thrust of the one blade at variance with that of the others. When ice is thrown off from the propeller, some of the fragments strike the airplane fuselage and cause both damage to the airplane and consternation among the occupants. Rime ice forms only on the propeller blade leading edge in the early stages of an icing encounter, but as the flight is prolonged, the accretions may extend rearward over the thrust face as well. Glaze ice forms over a wider portion of the blade than does rime, and the deformation of the blade-airfoil section results in a greater loss in thrust.

The extent of the ice formation on a propeller blade will depend upon the water-droplet content, the drop size, the forward velocity of the airplane, and the propeller speed. It appears that if the adiabatic temperature rise on the blade leading edge results in a compression temperature above 32 F, ice will not form. The data on this phenomenon are not complete or conclusive. Experience has shown, however, that direct-driven propellers, which rotate at 1800 rpm or greater speeds, are not readily affected by icing conditions. Such experience, it must be noted, has not been obtained in operations of long duration or on aircraft which permit extensive flights in inclement weather, with the exception of one of the NACA ice research airplanes, a Lockheed 12A. This airplane was operated with the propellers turning at 2000 rpm without the use of any protective device, and has not been affected by flights in icing conditions. The reason for this failure to gather ice may be due to the centrifugal forces or the aerodynamic heating or both. The former probably was the predominant factor over the regions near the hub and the heating probably was effective at the high-velocity regions near the propeller tip. Attention is directed to this experience due to the possible relation that the phenomenon may have to the problem of ice prevention on the wings of the extremely high-speed airplane of the future.

The prevention and removal of ice on propellers have been attempted by mechanical, chemical, and thermal means. Experiments with elastic blade-covering devices, which were intended to stretch and thereby break the ice bond, have not given satisfactory results and do not appear practical.

Chemicals to be applied prior to flight have received extensive attention for a number of years. Recent results have justified the use, in service, of some compounds. The objections to this method seem to be in the difficulty in the field under service conditions in getting a proper application prior to flight, the loss in effectiveness due to time elapsed between application and use, rain, and to the effects upon the performance of the airplane produced by applications of certain compounds.

It can be expected, however, that preparations will be available which, if properly applied immediately before a flight, will provide an allowable minimum of protection for certain types and sizes of propellers during operations of limited duration. The further development of the method requires service experience.

Chemicals applied during flight have provided airline operators with an allowable minimum of protection on the propeller types and sizes which have thus far been employed in commercial aviation. Aids toward the efficient

distribution of alcohol, which has been found by extensive service testing to be the most practical chemical, have been made available. The aids consist of a grooved elastic covering, applied to the blade leading edge, which conducts the fluid to outer blade stations; and elastic connecting devices between the collector or slinger ring and the blade leading-edge groove system. The effective operation of the alcohol system depends upon the alcohol being distributed prior to the icing encounter. The water droplets, upon impact, mix with the alcohol and, although some ice may form, the accretions are soft and easily thrown off without damage or great annoyance to cabin occupants.

The objections to the use of the alcohol system for propeller blades are, that skill on the part of the pilot is necessary in order to get reasonably economical use of the fluid, that the system does not function satisfactorily in all types of ice, and that it is doubtful if satisfactory protection can be obtained on the larger-diameter propellers which are anticipated in the future. It is believed that slight improvements will be developed in the alcohol system; however, the aids for the most efficient use of the fluid noted above will receive an increased application in the future, and increased skill of pilots may give better protection than is now obtained.

The electrical heating of the propeller blade through the use of a hub-type, a-c generator and conducting rubber blade shoes has been investigated and found to possess practical possibilities. An accurate estimate of the practicability cannot be made from the limited research flights in icing conditions, but it does appear that this method will give protection at least equal to that obtained by the use of alcohol, and may weigh less and have less maintenance problems; however, this latter statement is open to question. The effectiveness of the thermal method is not as sensitive to icing types as is the alcohol system, and the skill of the pilot is not involved.

The degree of heating intensity is being determined through flight testing, and sufficient data have now been established to enable service testing to be undertaken. While the hub-type generator has been employed in the flight tests thus far, other sources of electric power are also under consideration. Other means of applying electric power than by the use of conducting rubber have also been considered.

In the future, one may expect to see the use of electrically heated propeller blades, that this method will give a better degree of protection than now is available, and that it will be in high favor among the pilots because the operation of the equipment will consist of turning a switch "on" or "off."

It is desirable to avoid depending upon piloting skills to operate ice-prevention equipment because, in icing conditions, which are almost always accompanied by instrument flying conditions, there are many other demands on the pilot.

Work remains in more accurately defining the blade areas to be heated and the local heating intensity, in the development of a serviceable blade heating appliance, in the development of a lower-weight electric power source, and in the development of dependable means by which the satisfactory operation of the equipment may be indicated in flight.

Developments of other means of propeller blade heating for application to large diameter propellers and possibly rotating-wing aircraft are under way. The present state of these developments does not permit a forecast of what form of application these will take. It does seem safe to forecast, however, that the protection for propellers of diameters greater than 15 ft will involve methods not as yet developed for propeller blades.

■ Wings – Empennage

The shape and chordwise coverage of the formation of ice on an airfoil section, which may be employed as a wing or empennage surface, depends upon the type of ice, the magnitude of the accretion, the airfoil section, the Reynolds number of the flow, and the angle of attack. The formations of ice on the airplane wing cause an increase in drag, loss in lift, changes in the pitching moment, and, under certain circumstances, flutter of the wing or the control surfaces.

Glaze ice formations cause a deleterious effect in less time and with less water in the air than will rime ice. Rime ice, in its early stages of growth forms near the stagnation pressure region, which does not greatly change the flow pattern over the wing until a change in angle of attack is made.

Not all airplanes are equally affected by icing on the lifting surfaces, some being known by pilots to have the quality of carrying more ice than others. The problem of ice protection on airplane lifting surfaces is complicated by the use of slotted-wing leading edges, rotating-wing aircraft, gun installations, barrage-balloon cable cutters, landing-light fixtures, the type of wing construction, and the materials employed.

As a result of very extensive investigation by the Ames Aeronautical Laboratory at Moffett Field, Calif., and with the full cooperation of all branches of industry and the military services, data are available by which thermal ice-prevention equipment can be designed. These data are the results of tunnel, model-flight, and full-scale flight investigations which have been conducted under simulated and natural icing conditions.

Although the inflatable-type, leading-edge de-icer has been used extensively, the protection provided by this device has not been entirely satisfactory with existing aircraft. The anticipated use of higher flight speeds and larger airplanes in the future will probably make even less satisfactory the mechanical method of ice removal. It may be expected that the trend will be away from this equipment.

Attempts have been made to employ chemicals which are applied prior to or during flight, but the results of such attempts have not been successful and it is doubtful if the future will bring a change in this regard.

In view of these results, it may be expected that airplanes of the future which are to operate in inclement weather conditions and in regions or at altitudes giving temperatures below the freezing level will be provided with some system of wing and empennage surface heating as protection against ice.

The types of aircraft and the uses to which they are applied today influence the trend of the development of the heated wing. The present designs are closely related to the power wing-area ratio, the wing loading, operating speeds and altitudes, the number of independent powerplants, distances between normal landing places or length of operations, and geographic regions through which operations are to function.

There is some question now whether the source of heat should be the engine exhaust gas or fuel-burning air heaters. This argument may have to be answered by service operations. It appears to many that the most practical and logical source of heat is the engine exhaust, and the rapid advances which have been made in the development of exhaust-air heat exchangers may eliminate the consumption of additional fuel for the ice-protection needs. However, consideration must be given to the rapid developments that are also being made in gasoline-burning air heaters and to the possibility that the actual fuel requirements in practical service operations may not be a serious objection. The NACA has been particularly active in the development of the exhaust-air heat exchanger because of the apparent practicability of this source of heated air for ice protection and cabin heating, and because of the possible relation that such development will have in the provision of flame-suppression equipment. It is believed that the future will see extensive use of the exhaust-air heat exchanger in the protection against ice.

While the consideration of the design criteria and details cannot be undertaken in the present discussion, such data being available through the NACA publications, general characteristics of the thermal ice-prevention method can be stated. It may be anticipated that, in the near future, ice-prevention equipment for the fixed aerodynamic surfaces may be described as follows:

All of the wing and empennage leading-edge regions will be directly heated to a degree which will preclude the loss, due to ice, of aerodynamic efficiency of these parts. Frost removal from the heated regions prior to take-off will be possible.

The weight of the equipment, when combined with cabin heating through the joint use of heat exchangers, will add from ½ to 1½% of the airplane gross weight. The wide range noted may be expected due to the varying degrees of protection desired and the skill and freedom of the designer.

With the exception of the heat exchangers, there will be practically no maintenance and the life of the equipment will be that of the airplane. The heat exchangers may be developed so that the life is equal to or better than that of the exhaust collectors now in service.

The failure of the thermal system, due to inadequate heating, to meet the meteorological condition, will not create an emergency if the delivered heat is within at least 50% of the need for complete ice prevention. The nature of failure of the thermal system prompts this bold assertion. When insufficient heat is employed, ice forms on the surfaces rearward from the leading edge, and after accumulating and covering a strip not more than 2 in. wide and perhaps 10 in. long, the ice has been seen to be blown away. The insulating effect of the ice permits the protected skin temperature to rise, and as soon as a film of water is formed, the formation is blown off.

The use of the equipment will involve between ½ and 1% of the total thrust power in increased drag due to the ducted hot air. When the equipment is not in use, it is possible to discharge the heated air in such a manner as to obtain a slight thrust. However, it is questionable if such refinement will be found in many designs. Air circulation through the system will be by the ram of the propeller stream on tractor-equipped airplanes.

The temperature rise of the wing-leading-edge skin will approach 200 F when the equipment is in operation in air which is free from water droplets. It is believed that such temperature effects will not exceed those experienced from normal operations in the tropics. However, consideration for the location of hot-air ducts with relation to gasoline tanks and fuel lines will probably be necessary.

The designs will probably be based on satisfactory protection at the power and speed of maximum range, minimum weight, and most efficient altitude. An allowable minimum degree of protection may be available from such designs in the descent on single engine with a two-engine airplane.

Skill on the part of the pilot will not be required in the operation of the equipment. Many airplane systems will probably be provided with a simple on-off switch.

Special skill in the design of the duct systems of the equipment is necessary if adequate air is to be delivered to all parts by the ram pressure during all operating conditions.

There does not appear to be any hope of applying to wood or plastic leading-edge structures the air-heating systems as now developed, due to the high thermal resistivity of these materials.

The principal work remaining in the development of thermal ice-prevention systems for the fixed aerodynamic surfaces consists of investigations to define accurately the meteorological conditions which may be encountered, to refine the analytical processes involved in the leading-edge design, to determine the degree of protection which is obtained at operating conditions other than the design condition (such as single-engine operation), and to discover structural arrangements which will minimize the added weight. It is doubtful if these and other minor items of research will change the trend to the use of the thermal system.

The time required to bring the thermal system into wide usage will be greatly influenced by the structural changes that must be made by the manufacturer. In some cases these changes are profound. The improved protection and aerodynamic cleanliness, the reduced maintenance, and the improved life of the equipment will be, however, strong arguments to hasten the change.

Operators with large inventories of present airplane equipment may become interested in the possibility of modifying their airplanes. The values to be weighed in such a consideration involve both financial and technical factors. While the modification probably can be accomplished, as far as the engineering factors are concerned, it is not possible to predict what the economic factors will dictate if and when the issue arises. At present there is no predictable trend in this direction.

■ Windshield

The formation of ice on the airplane windshield usually results in complete loss of vision in a very few seconds after icing conditions have been encountered. Some windshield installations are not as vulnerable to all types of ice as are others. A windshield which is smoothly faired into the surface of a streamlined body, and which is located rearward from the stagnation region, will not collect a rime-type ice unless the conditions are so severe as to cover the entire nose of the fuselage.

Research development activities have been concerned with the use of the wiper which, with alcohol spread over

the swept area, is intended to prevent ice, and with various thermal systems which are supposed either to prevent or to remove ice on the exterior and to remove frost on the interior of the windshield.

The wiper has not been found satisfactory as a means of windshield ice prevention and, although such equipment will be seen in service, it is believed that the trend will be to the use of the wiper only as a means of removing rain, and that the use of alcohol will decrease.

The passage of heated air between the panels of a double glazed windshield has been found to provide protection in airline service; however, it should be noted that the extent of flights in icing weather is limited. A complete service test of the heated windshield will not be possible until other protection on the airplane permits more extensive flight in inclement weather on scheduled operations. There are disadvantages as well as advantages, and while it may be expected that the double glazed heated windshield will receive increased application, continued effort will also be made to discover a more practical solution to the windshield problem. Serious problems arise in the application of the double-panel system in installations employing curved or very thick glass. The high thermal resistivity of plastics almost certainly precludes the use of such materials except in safety glass, in which case the plastic binder thickness must be held to very low values.

A double-windshield design has been prepared by the CAA which was predicated on the problems of both ice-prevention and bird-impact protection. In this installation the exterior panel was specified as a heat-treated plate glass. Safety glass ⅜ in. thick was specified for the inner panel, with a ¼-in. gap between the panels. The arrangement and dimensions of the CAA design are believed to be very good and probably will be the ultimate of future design trends.

Other heating systems involving electric resistance, infra-red radiation, and ultra-high-frequency radiation have been proposed. The use of such equipment will depend upon the availability of electric power on the aircraft. If an abundance of power is available as a result of a peak load not affecting the ice-protection equipment, it may be that practical applications of these types will be developed. The quantity of heating required will be the same as that found necessary in the air-heating system. Still other thermal methods are being examined, and it is possible that the future will find the use of equipment which is substantially superior to the systems described above.

■ Air-Speed Pressure Heads

As is well understood, the formation of ice on the air-speed pitot-static head will block the total pressure port and also cause a false static pressure to be shown. The result is the same, irrespective of ice type. Electric heating of the heads is generally used to prevent such a failure of the air-speed system. This method will probably continue in use.

A further type of failure in the air-speed system which is not well understood often occurs on some installations. The air-speed head on many airplanes is mounted on a short strut which protrudes from the side or bottom of the fuselage. The need for prevention of ice on this strut is not well understood. When ice forms in the region of the air-speed head, but not directly on it, the airflow pattern is disturbed and the static pressure is altered. It is possible for the formation of ice on the air-speed-head mounting strut to cause an error of more than 25 mph within 5 min after encountering icing conditions. Such errors cause both the air-speed meter and the altimeter to indicate low.

Consideration has been given to the location of the static orifice on the side of the fuselage. This location may be either ahead of or behind the wing, depending on the airplane fuselage shape, the wing location, and other shape factors.

If the static orifice is located on the side of the fuselage, the total head may then be located either at the nose of the fuselage or on shorter struts at the side or below. Electric heating of the total head will still be necessary with this system, although it is possible that no heating will be needed for the static orifices. Installations employing the fuselage static pressure vent and a separate total pressure head will probably be extensively used in the future, and such use may be considered a forward step in the development of ice-protection equipment. Work remains in this development through which the type of fixtures and optimum locations for the orifices will be established.

Before aircraft operations can be safely planned without regard to the icing conditions en route, other parts of the airplane not as yet generally considered vulnerable must be provided with protection. These include the radio-antenna system, engine cowls, and all functional protuberances such as engine and fuel-tank vents.

■ Effects On Operations

The effect of the recent developments in ice-prevention technique upon operations of commercial transportation is difficult to evaluate. It is doubtful if the effect will be great until other important problems of inclement-weather flight have also been effectively dealt with. During most encounters with heavy icing conditions and many other encounters by NACA with icing of less severity, radio communication has failed because of precipitation static. Most flights cancelled by the airlines during icing conditions are cancelled, not because of the ice alone, but just as much because of limited or zero terrain clearance. Thus, until radio facilities and blind landings have been extensively developed, the new ice-prevention systems will not actually enable operators to decrease the percentage of cancellations or uncompleted schedules. For the present, the principal advantage of the new equipment will be that the pilot is not faced with an acute emergency if icing conditions are inadvertently encountered.

Since unlimited flight operations in icing conditions may result in encounters with frontal-type inclement weather and other weather phenomena likely to produce vertical gusts of great velocity, the effect of recent developments in ice prevention technique on operations will be determined to some extent also by the skill of the meteorologists in directing the flight path around the storm centers and/or the design of the airplane to encounter safely such gusts.

Recent developments in the protection against ice on airplanes undoubtedly have removed some serious restrictions to the advancement of aeronautics. It is easily understood from the concluding remarks, however, that the usefulness of the airplane is not greatly enhanced by advancements in one field only but depends upon the orderly general advance in all sciences which pertain to the design, construction, and operation of the airplane.

PROPELLER and GE
for Aircraft De

by R. S. SCHAIRER
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FOR aircraft designed for cruising operation it is important that there be high propulsive efficiency at the engine speed for minimum specific fuel consumption for almost all cruising conditions.

For a given airplane with a single-stage, two-speed engine supercharger there is ordinarily more than one combination of propeller diameter and gear ratio which give within 1% of maximum propulsive efficiency for all cruising conditions up to an altitude of 20,000 ft and up

AT present, the competition between airplanes in either military or commercial use is so great that it is very important to design each part of the airplane in the best possible manner. One of the most direct ways to achieve a good design is to utilize the power of the engine as efficiently as possible. This means that the propeller and the propeller drive gear ratio must be properly chosen to fit the particular airplane and engine. The best combination of propeller and gear ratio for any engine depends upon the type and general arrangement of the airplane in which the engine is installed. Since the number of variables involved in the selection of the best propeller and gear ratio for any one airplane is so great and there are so many different types and arrangements of airplane, it is not possible in any short time to make a complete survey of the requirements of all kinds of aircraft. It is the purpose of this paper to discuss the propeller and gear ratio requirements of aircraft designed specifically for cruising operation at a wide range of conditions of speed, altitude, and weight.

The three most important considerations in the choice of propeller and gear ratio are performance, weight, and general arrangement of the airplane. The size and shape of the propeller to give the best performance are dependent upon the performance itself. Weight is of great importance because the range and load-carrying capabilities of the airplane are directly affected by the weight empty. The layout of the airplane often places limits on the size and type of propeller that can be used. It is seldom possible to find a combination of propeller and gear ratio that is the most desirable from all points of view. The final selection, therefore, is a compromise which depends upon the relative importance of the items influencing the selection.

Since the primary purpose of the propeller is to propel the airplane, the most important consideration in its choice must be the efficiency with which it accomplishes its purpose. The first step, therefore, in any investigation to find the best propeller is to find the most efficient one for all of the important performance items, without considering any limits on diameter and weight. The aerodynamic characteristics of propellers are such that for any given propeller configuration the propulsive efficiency depends upon the power being absorbed, the air density, the air speed, the propeller diameter, and the propeller speed. The general nature of the variation of the efficiency with these variables is the same for all types of propellers, but the variation is influenced by the shape of the propeller blades, the number of blades, and the relative positions of the propeller and other bodies near it. There are several different ways in which the five items just mentioned, which influence the propulsive efficiency for any given propeller configuration, can be grouped into two dimensionless coefficients which completely describe the operating conditions affecting the efficiency, except for the effects of compressibility of the air at very high speeds of the propeller blades with respect to the air. Since there is only one set of values of these two coefficients which gives the maximum propulsive efficiency, it is not possible with a fixed gear ratio to have the maximum efficiency for all conditions of flight. It is necessary, therefore, to select those items of performance which are of the greatest importance and to find the combination of propeller diameter and gear ratio which gives high efficiency for those conditions.

The most important items of performance for aircraft designed for cruising operation are cruising, take-off, and climb at the highest desired cruising altitude. Since the airplane is in cruising operation during most of its time in the air, it is essential that it have high propulsive efficiency for almost all cruising conditions which are likely to be

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GEAR RATIO REQUIREMENTS

Designed for Cruising Operation

to maximum cruising speed. Of these combinations, that one is best which has the largest diameter and the lowest ratio of propeller speed to engine speed, because it gives the best take-off and climbing performance.

■ ■

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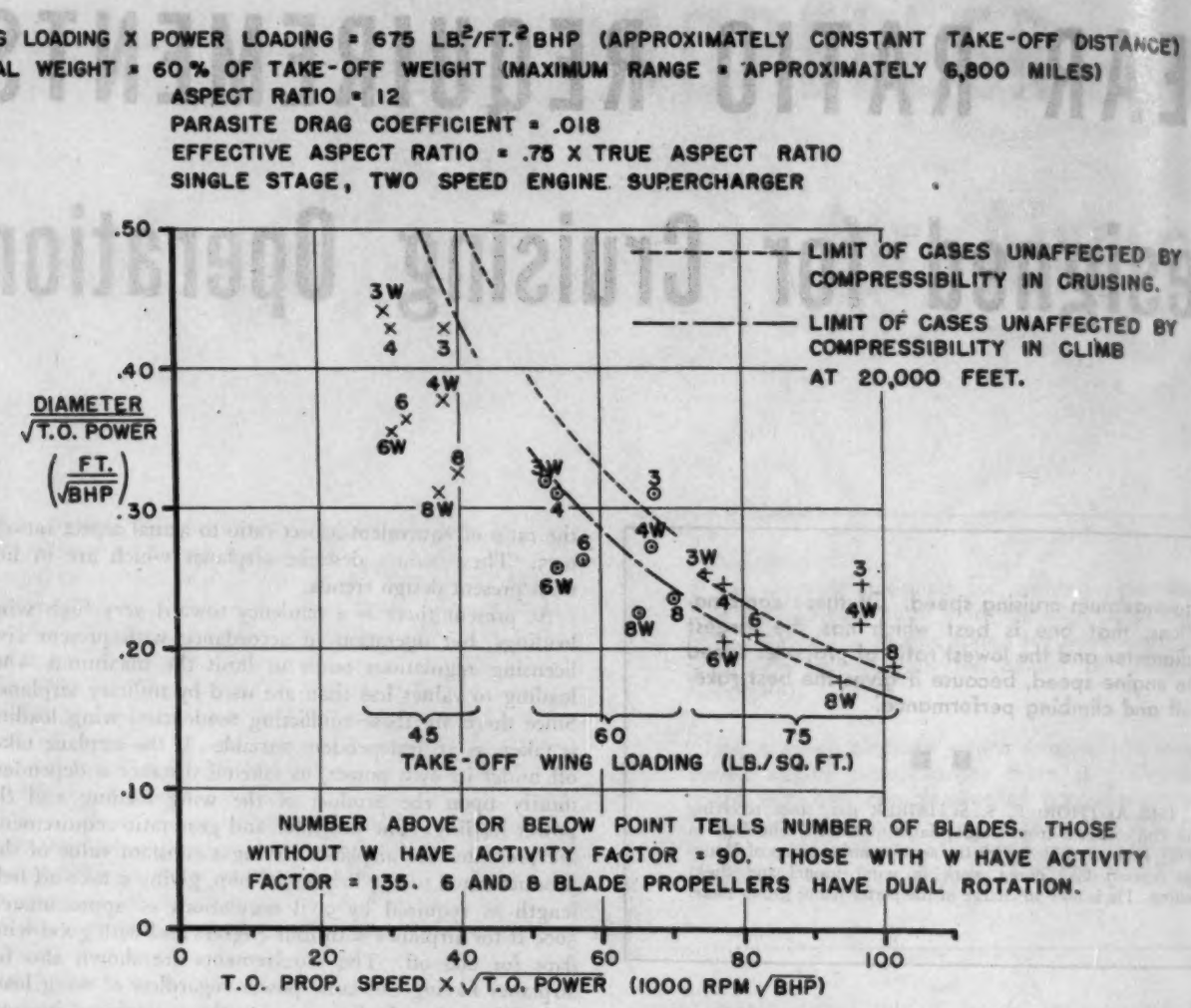
encountered. Small gains in efficiency in cruising can save enough weight of fuel required for a long flight to compensate for the increase in propeller or propeller drive gear weight required to give the increased efficiency. Even for long-range aircraft it is often desirable to cruise at speeds greater than the speed for maximum range. It is important, therefore, to have high propulsive efficiency at speeds as high as the maximum cruising speed. Good take-off characteristics are necessary so that large loads can be carried from the available airports. Improvements in efficiency during take-off can increase the take-off gross weight for a given field length more than the increase in weight empty necessary to make this gain. For cruising at any altitude the airplane must be able to climb well enough so that the cruising altitude can be maintained easily in rough air.

The range of an airplane depends upon the ratio of fuel weight to gross weight, the ratio of lift to drag in the cruising altitude, the propulsive efficiency, and the specific fuel consumption. An airplane designed for cruising operation should have a high ratio of lift to drag. An airplane having a high aspect ratio and a low parasite drag coefficient fulfills this requirement. The upper limit on aspect ratio is usually set by the balance between increased lift to drag ratio and increased structural weight. For the basic cases considered in this paper the aspect ratio is 12, the ratio of equivalent parasite area to wing area is 0.018, and the ratio of equivalent aspect ratio to actual aspect ratio is 0.75. These values describe airplanes which are in line with present design trends.

At present there is a tendency toward very high wing loadings, but operation in accordance with present civil licensing regulations tends to limit the maximum wing loading to values less than are used by military airplanes. Since there are these conflicting tendencies, wing loading is taken as an independent variable. If the airplane takes off under its own power, its take-off distance is dependent mostly upon the product of the wing loading and the power loading. The propeller and gear ratio requirements are presented for airplanes having a constant value of this product equal to 675 lb^2 per ft^2 bhp, giving a take-off field length as required by civil regulations of approximately 5000 ft for airplanes with four engines and with good wing flaps for take-off. The requirements are shown also for airplanes having the same power regardless of wing loading. The take-off distances for these airplanes increase rapidly as the wing loading increases. Take-off wing loadings from 45 to 75 lb per sq ft are considered. For the airplanes with constant take-off field length, the corresponding take-off power loadings vary from 15 to 9 lb per bhp. For the airplanes with constant power, the corresponding take-off power loadings vary from 8.44 to 14.06 lb per bhp, giving approximate commercial field lengths from 2500 to 9500 ft. At a wing loading of 60 lb per sq ft, the take-off field length is approximately 5000 ft for the cases with constant power.

The items mentioned so far determine the shape and general aerodynamic characteristics of the airplane but not its size. The engine can also be considered in dimensionless form by assuming that the ratio of power to take-off power at sea level is dependent upon the altitude and ratio of the engine speed to the take-off engine speed. In this paper only one type of engine supercharging is used. It is assumed that the engines are typical of recent radial air-cooled engines equipped with a single-stage, two-speed supercharger. The maximum cruising powers in low and high blower ratios are 54.3 and 46.4% of take-off power. Maximum cruising engine speed is 80.4% of take-off engine speed. The critical altitudes with maximum cruising power are 11,500 ft in low blower ratio and 24,200 ft in high blower ratio. In most cases the engine speed for lowest specific fuel consumption is the engine speed for


WING LOADING X POWER LOADING = 675 LB.²/FT.² BHP (APPROXIMATELY CONSTANT TAKE-OFF DISTANCE)
FINAL WEIGHT = 60 % OF TAKE-OFF WEIGHT (MAXIMUM RANGE = APPROXIMATELY 6,800 MILES)
ASPECT RATIO = 12
PARASITE DRAG COEFFICIENT = .018
EFFECTIVE ASPECT RATIO = .75 X TRUE ASPECT RATIO
SINGLE STAGE, TWO SPEED ENGINE SUPERCHARGER
LIMIT OF CASES UNAFFECTED BY COMPRESSIBILITY IN CRUISING.
LIMIT OF CASES UNAFFECTED BY COMPRESSIBILITY IN CLIMB AT 20,000 FEET.
DIAMETER / √T.O. POWER (FT./√BHP)
.50
.40
.30
.20
.10
0
45
60
75
TAKE-OFF WING LOADING (LB./SQ. FT.)
NUMBER ABOVE OR BELOW POINT TELLS NUMBER OF BLADES. THOSE WITHOUT W HAVE ACTIVITY FACTOR = 90. THOSE WITH W HAVE ACTIVITY FACTOR = 135. 6 AND 8 BLADE PROPELLERS HAVE DUAL ROTATION.
0
20
40
60
80
100
T.O. PROP. SPEED X √T.O. POWER (1000 RPM √BHP)

■ Fig. 1 – Best propeller diameter and take-off propeller speed

full-throttle operation. This dimensionless handling of the airplane and engine gives the best propeller and gear ratio in terms of diameter divided by the square root of the take-off power per engine and take-off propeller speed times the square root of the take-off power. Thus, the actual diameter and gear ratio are immediately determined when the take-off power and take-off engine speed are given. All of the results are presented, therefore, using take-off propeller speed instead of gear ratio.

The general arrangement of the airplanes is assumed to be conventional. The investigation is carried out using tractor propeller data but the results should hold quite closely for pusher propellers. The result is also independent of the number of engines, except that the commercial field lengths required for airplanes having more or less than four engines differs from the values already mentioned for airplanes with four engines.

The basic criterion in the determination of the best propeller and gear ratio for an airplane designed for cruising operation is that for all cruising conditions there should be high propulsive efficiency and low specific fuel consumption. It is attempted, therefore, to choose a propeller diameter and a gear ratio which give a propulsive efficiency within approximately 1% of the maximum possible efficiency for cruising operation at the engine speed for lowest specific fuel consumption for all combinations of the following conditions:

1. Altitudes from sea level to 20,000 ft.
2. Speeds from 10% above the speed for maximum ratio of lift to drag to the speed with maximum cruising power.
3. Weights from the take-off weight to the final weight.

These wide ranges of values of speed and altitude are considered because these types of operation are frequently used even with airplanes designed particularly for long-range work. The range of the airplane is greatly dependent upon the ratio of final gross weight to take-off weight. The effect of airplane range upon the propeller and gear ratio selection is found by using different values of the ratio of final to initial gross weights. When the final weights are 60 and 80% of the initial weight, the respective maximum ranges are approximately 6800 and 3000 miles. The range with maximum cruising power is from 70 to 80% of the maximum range. It is usually discovered that for any airplane there are a number of combinations of propeller diameter and gear ratio which give the desired efficiency for all of these cruising conditions. When this is

true the conditions of climbing at high altitude and take-off are used, and of those combinations satisfactory for cruising that one is chosen which gives the highest propulsive efficiency for these conditions. The best combination for take-off and climb of those which meet the cruising conditions is the one with the largest diameter and the lowest gear ratio. This combination gives the highest basic propulsive efficiency in climb and take-off and also is least troubled by compressibility. It is assumed that the propulsive efficiency begins to be influenced by compressibility when the helical tip speed of the propeller exceeds 950 fps.

The best combination of propeller diameter and gear ratio depends upon the number of propeller blades, their activity factor, and whether or not there is dual rotation. The effects of these things on the propeller characteristics are best shown by wind-tunnel tests of each of the possible configurations. Such tests, conducted by the National Advisory Committee for Aeronautics, are used as the basis of the results shown in this paper. Sometimes, the trends indicated seem contradictory but at present there are no better data and these must be used. Fig. 1 shows for take-off wing loadings of 45, 60, and 75 lb per sq ft the best combinations of propeller diameter and take-off propeller speed for a number of different types of propeller. All cases considered in Fig. 1 have the product of wing loading and power loading of 675 lb^2 per ft^2 bhp and a final weight of 60% of the initial weight. Lines are drawn which show the limit of cases not influenced by compressibility of the air for the most critical condition of cruising and for climb at 20,000 ft. Any case below these lines is not affected by compressibility. At a wing loading of 45 lb per sq ft, compressibility is of no concern, but at 60 and 75 lb per sq ft, it becomes an important limitation. Fig. 1 shows that increasing the total activity factor by increasing the blade width or the number of blades tends to permit an optimum propeller selection which is less influenced by compressibility. This is true because the region of highest propulsive efficiency occurs at higher values of the propeller advance ratio J for the propellers with higher total activity factor. Therefore, at the same forward speed of the airplane, a lower propeller tip speed is required in order to obtain maximum efficiency.

The best combinations for different types of propeller are not equally good because the general level of propulsive efficiency depends upon the type of propeller. Table 1 compares propellers of different types. It shows propulsive efficiencies for cruising, ratios of best cruising diameter and gear ratio to best diameter and gear ratio for three-blade, 90 activity factor propeller, and propulsive efficiencies at take-off speed for the best diameters and gear ratios for the airplanes having approximately the same take-off distance. It shows that when cruising is the most important consideration it is best to use a three-blade propeller with narrow blades, because this has the highest efficiency and is also the lightest in weight, even when compared with six- and eight-blade propellers of much smaller diameter. The best three-blade propeller is usually small enough so that there is no great weight penalty in designing a long-range airplane capable of using it. Table 1 shows also that, for long-range airplanes, increasing the total activity factor of the propeller by changing from three narrow blades either to three wider ones or to four narrow ones increases the propulsive efficiency for take-off. It is assumed here that take-off is not critical and, therefore, the cases presented are for a three-blade propeller with narrow blades, but the results are easily applied to other types of propeller

Table 1—Comparison of Different Types of Propeller

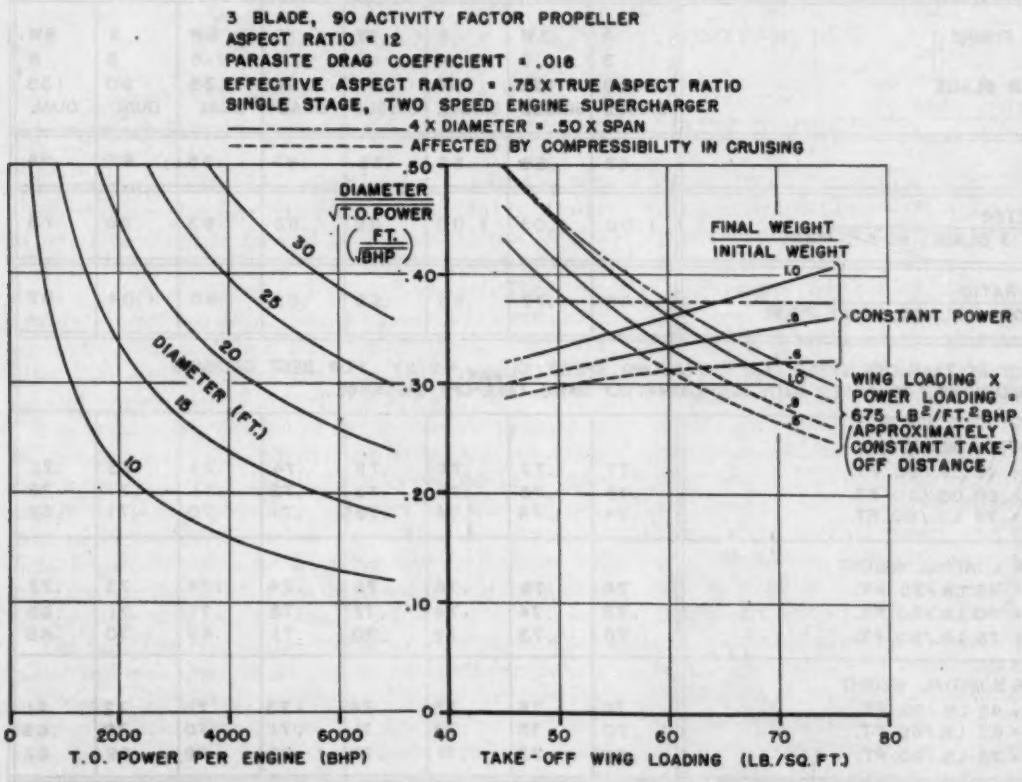
DESIGNATION USED IN FIGURE 1	3	3W	4	4W	6	6W	8	8W
NUMBER OF BLADES	3	3	4	4	6	6	8	8
ACTIVITY FACTOR PER BLADE	90	135	90	135	90	135	90	135
ROTATION	SINGLE	SINGLE	SINGLE	SINGLE	DUAL	DUAL	DUAL	DUAL
CRUISING EFFICIENCY	.87	.86	.86	.85	.87	.85	.87	.84
BEST CRUISING DIAMETER / BEST DIAMETER FOR 3 BLADE, 90 A.F. PROP.	1.00	1.03	1.00	.88	.85	.83	.76	.73
BEST CRUISING GEAR RATIO / BEST GEAR RATIO FOR 3 BLADE, 90 A.F. PROP.	1.00	.77	.80	.99	.85	.80	1.04	.97
PROPULSIVE EFFICIENCY AT TAKE-OFF SPEED (1.15 X STALLING SPEED, $CL_{MAX.} = 2.2$) FOR BEST CRUISING COMBINATIONS FOR CASES WITH APPROXIMATELY SAME TAKE-OFF DISTANCE.								
FINAL WEIGHT = INITIAL WEIGHT								
WING LOADING = 45 LB./SQ. FT.	.77	.77	.76	.75	.74	.73	.73	.72
WING LOADING = 60 LB./SQ. FT.	.75	.75	.74	.73	.72	.71	.71	.70
WING LOADING = 75 LB./SQ. FT.	.74	.74	.74	.72	.71	.70	.71	.69
FINAL WEIGHT = .8 X INITIAL WEIGHT								
WING LOADING = 45 LB./SQ. FT.	.76	.76	.76	.75	.74	.74	.73	.72
WING LOADING = 60 LB./SQ. FT.	.73	.74	.74	.72	.72	.71	.71	.69
WING LOADING = 75 LB./SQ. FT.	.70	.73	.72	.70	.71	.69	.70	.68
FINAL WEIGHT = .6 X INITIAL WEIGHT								
WING LOADING = 45 LB./SQ. FT.	.72	.75	.75	.74	.73	.72	.72	.71
WING LOADING = 60 LB./SQ. FT.	.70	.73	.73	.71	.71	.70	.70	.69
WING LOADING = 75 LB./SQ. FT.	.69	.72	.71	.70	.69	.68	.69	.67

by multiplying them by the ratios of best cruising diameter and gear ratio to best diameter and gear ratio for three-blade, 90 activity factor propeller given in Table 1. Sometimes, compressibility effects, space restrictions, or availability of present designs makes it necessary to depart from what would otherwise be the best combination of propeller and gear ratio. By relaxing the requirement that the propulsive efficiency be within 1% of maximum efficiency for all cruising conditions, the number of allowable cases is greatly increased. In the attempt to avoid compressibility effects by this relaxation of the cruising conditions, the diameter increases so much that it is unlikely that the airplane can be designed to use so large a propeller. In this case it is probably better to use a propeller with higher activity factor which has its maximum efficiency at a higher value of the propeller advance ratio.

Fig. 2 shows the best diameter for a three-blade propeller with narrow blades, plotted as a function of take-off wing loading. For those cases having approximately the same take-off distance, the diameter decreases as the take-off wing loading increases. When the power is held constant, the diameter increases as the wing loading increases. As the range for which the airplane is designed increases, the best diameter decreases. The dotted lines show the cases in which compressibility of the air decreases the propulsive efficiency in at least one of the cruising conditions used in the determination of the best diameter and gear ratio. These cases occur at the smaller diameters because the best propeller speed increases more rapidly than the diameter decreases. Lines are drawn which show the diameter for a four-engine airplane such that the sum of the diameters of the propellers is equal to half of the span. This is approximately the largest diameter on any modern four-engine airplane. Nearly all of the best diameters for the cases investigated are smaller than this.

Fig. 3 shows the best take-off propeller speed for a three-blade propeller with narrow blades plotted as a function of take-off wing loading. The variation of best take-off propeller speed with wing loading and with design range of the airplane is opposite to the variation of best diameter but is at a greater rate. The propeller speeds given in Fig. 3 must be used with the diameters given in Fig. 2, since these combinations are the only ones having the desired characteristics. Both Figs. 2 and 3 show that even for the limited uses of the airplanes discussed here there are wide ranges of diameter and propeller speed (or gear ratio) required to give best performance. It is very important, therefore, that there be available to the airplane designer a sufficient number of propellers and engine gear ratios to enable him to select a combination that closely approaches the requirements of his airplane. The gear ratios available with any engine should be evenly spaced over the whole range of values required by different airplane designs and should not be bunched together to meet the specific requirements of just a few designs.

An investigation was made of the case of an airplane with an aspect ratio of 9 and an initial wing loading of 45 lb per sq ft and in all other respects the same as the cases previously discussed which have a field length of 5000 ft. The best combination of diameter and gear ratio for the airplane with an aspect ratio of 9 is almost exactly the same as that for the airplane with an aspect ratio of 12. With an aspect ratio of 9, however, this best diameter is a greater fraction of the wing span than with an aspect ratio of 12 and, therefore, the layout problem is more difficult. This shows that aspect ratio does not greatly affect the propeller and gear ratio requirements.



■ Fig. 2 – Effect of wing loading on best diameter

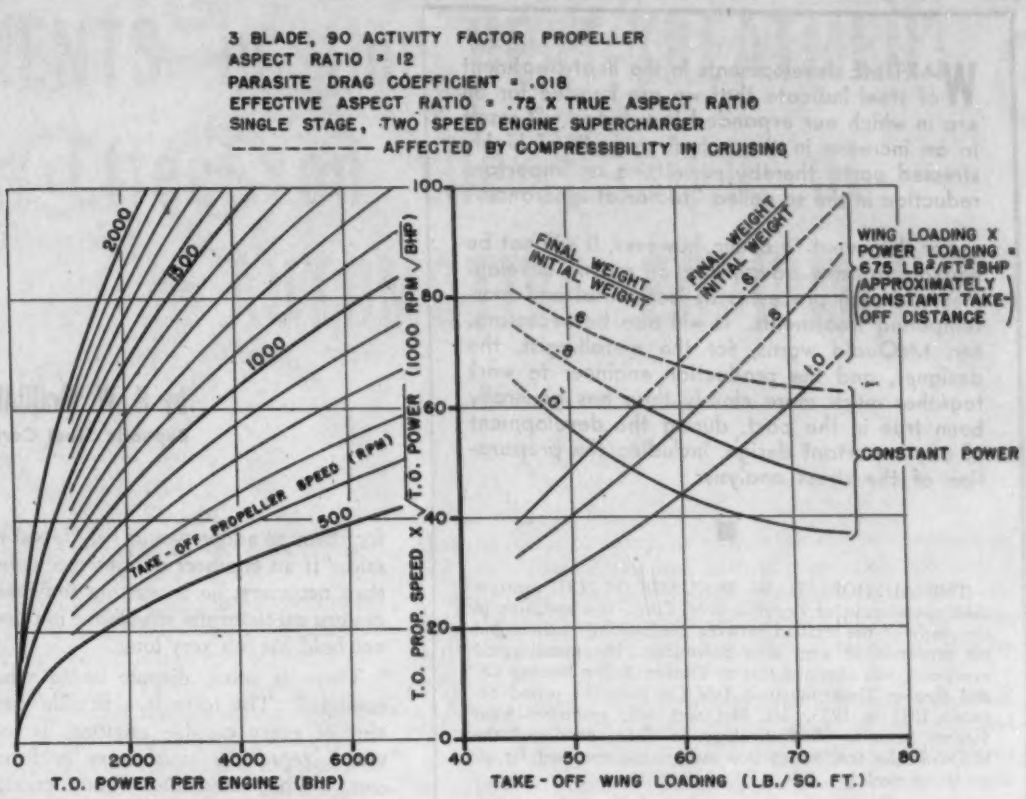


Fig. 3 – Effect of wing loading on best take-off propeller speed

The effects of changing the engine characteristics were investigated by choosing another engine of the same type as used previously but made by another manufacturer. The differences in engine characteristics are in the ratios of maximum cruising powers to take-off power, in the critical altitudes, and in the engine speeds for lowest specific fuel consumption at any power and altitude. The best diameter and gear ratio for this engine are within 2% of the best values for the other engine. This indicates that small changes in engine characteristics do not appreciably affect the selection.

One of the most common occurrences in the course of an airplane design is that larger engines are installed and the maximum gross weight is increased. It is of interest to know if, during this process, it is desirable to change the propeller. This problem is easily answered from the cases presented in Fig. 2, with approximately the same take-off distance, by holding the wing area constant and increasing the power directly with the square of the initial gross weight. When the final weight is 60% of the initial weight, the result is that the best diameter at a wing loading of 75 lb per sq ft is only 2% less than the best diameter at a wing loading of 45 lb per sq ft. In order to avoid compressibility trouble, however, the propeller for the case of higher wing loading must have a higher activity factor. The result is that if the airplane is designed to use the best propeller for the first engine installation, the layout still permits the use of the best propeller for the heavier conditions, when the gross weight and engine power are increased so as to maintain a constant take-off distance.

For airplanes designed specifically for cruising operation at a wide range of conditions of speed, altitude, and weight, the following conclusions are drawn from this investigation of the propeller and gear ratio requirements.

1. The best combination of propeller and gear ratio depends primarily on take-off wing loading, take-off power loading, and the range for which the airplane is designed. With everything else held constant, changes in aspect ratio of the wing do not greatly affect the result. Also, for a given type of engine supercharging, small variations in engine characteristics are not important.

2. It is necessary for any engine to have a number of alternate propeller drive gear ratios evenly spaced over a range of values so that a close approximation to the best combination of propeller and gear ratio is available for any airplane using the engine.

3. The best propeller is usually not too large for airplanes of conventional design.

4. With a single-stage, two-speed engine supercharger there is ordinarily a combination of propeller and gear ratio which gives within 1% of maximum propulsive efficiency for all cruising conditions up to an altitude of 20,000 ft and up to maximum cruising speed with the engine operating at the speed for lowest specific fuel consumption; therefore, for this type of engine installation, there is no need for a two-speed or variable-speed propeller drive.

5. When the power in a given airplane increases proportionately to the square of the gross weight, thereby leaving the take-off distance nearly the same, the best propeller diameter is nearly constant. When the wing loading of an airplane increases but the power does not change, the best diameter increases.

6. For future engines having more power than those available at present it is necessary to have propellers larger than any built to date.

WARTIME DE
of Steel an

by H. W. McQUAID
Republic Steel Corp.

WARTIME developments in the heat-treatment of steel indicate that we are headed for an era in which our expanded knowledge will result in an increase in the actual strength of highly stressed parts, thereby permitting an important reduction in the so called "factor of ignorance."

For this event to occur, however, it will not be enough to take advantage of recent developments in such processes as isothermal and martempering treatments. It will also be necessary, Mr. McQuaid warns, for the metallurgist, the designer, and the production engineer to work together much more closely than has generally been true in the past, during the development of any important design, including the preparation of the stress analysis.

■

THE AUTHOR: H. W. McQUAID (M '23), assistant chief metallurgist of Republic Steel Corp., is a specialist in the study of the relation between steelmaking practice and the properties of steel after processing. His metallurgical experience was obtained first at Timken Roller Bearing Co. and then at Timken-Detroit Axle Co. from the period between 1915 to 1933. Mr. McQuaid, who graduated from Stevens Institute of Technology in 1913, developed the McQuaid-Ehn test, which is a carburizing test used for all grades of steel.

THE preparation of a paper for the SAE on this subject is a pleasant task indeed since in a small way it offers an opportunity to discuss some subjects which have been of interest to the writer for many years.

Being limited by the words "wartime" and "heat-treatment" makes it a little difficult to discuss the effect of the many possible developments in steel-making methods, analysis, or economics on design. We can hardly avoid, however, some discussion of these factors, because of their relationship to the heat-treating operation.

The effect of any new developments on any group of men is often governed as much by their attitude and experience as it is by the importance of the development; therefore, much of this discussion, perhaps too much, will be centered around the present state of the art and what is necessary before the impact of wartime developments can take place. Much of this impact will not be felt for years and many of the changes indicated may never be made because of processing or marketing economics.

■ Some Factors in Design

In many cases the designer is hedged in by limitations of material, processing, or parts, which result in most designs being compromises. There is no doubt that one of the most important decisions of a designer concerns the criterion of a satisfactory design. In many cases the hidden forces which limit his design are those controlled by his customer, his costs, and his competition.

Many years ago a sage of some reputation advised a graduating class of mechanical engineers that a good motto for them to adopt would be, "Good enough is best." He said, "If an engineer designs mechanisms which are better than necessary, he is wasting someone's money, and if he designs mechanisms which are not good enough he would not hold his job very long."

There is some dispute as to what constitutes "good enough." The term is a flexible one. Nevertheless, the aim of every capable engineer is to develop a product which represents satisfactory performance at the lowest cost. What constitutes "good enough" varies with the user, and the more specific the application the easier it is to define "good enough."

Thus, in the design of a special loom for weaving extra-wide rugs, the requirements of the user can be fairly easily determined. Once the operating mechanism is developed, the material and design of the individual members can be worked out to ensure satisfactory performance.

In the design of a mechanism of wide distribution, where the volume is affected greatly by price, the definition of "satisfactory performance" is more difficult to determine. Thus, in an automobile, the designer must be guided always by the effect of any change, however minor, on the cost, and must determine whether the average car buyer will benefit proportionately to the increased cost. To build an automobile that will meet the full requirement of the taxi driver would mean that the average original owner would have to buy a car which was unnecessarily expensive, and the results (to him) not of importance. It is quite possible that such a car might result in improved second- and third-hand value.

But to get to specific instances of what is meant by "good enough," we need only to look at given parts of some mechanism, such as an automobile transmission, to find that the customer is the final factor in deciding what is good enough. The customer indicates his reaction to a transmission that is not good enough by objecting to increase in servicing costs, and finally by changing his mind as to the car of his choice. In a transmission, as in other parts of an automobile, there will always be a certain minimum number of failures regardless of how well it may be designed. These failures come from abuse, poor material, poor inspection, accident, and what not. Thus, there is a minimum percentage of failures which are in no way attributable to design; but as the percentage of failures rises above this minimum, it reaches a point where the customer

[This paper was presented at the SAE National War-Matériel Meeting, Detroit, June 7, 1944.

DEVELOPMENTS in the HEAT-TREATMENT and their Effect on the Design of AUTOMOTIVE EQUIPMENT

reaction (as reflected to the engineer by the sales department) requires some action.

Up to this point the design is considered as good enough. What this point is must be determined for every detailed part of a product, and may be different for different parts of the same mechanism.

For many years the designer, when his design starts to show evidence of obsolescence or inadequacy, had had a tendency to call in the production engineer and metallurgist and insist that they are parties of the first part when it comes to keeping his design in operation. As a net result, there has been an added expense in many parts because of a reluctance to admit that the design is lacking.

There has been little cooperation between the designer and the metallurgist in the past in preparing a new design for production. Even in old, established, well-organized plants, new designs are often approved and sent out for bids or into production without metallurgical check. They are often submitted to the metallurgist in the form of blue prints of finished drawings in large quantities and usually with the expectation of approval or criticism at a time which permits of no critical analysis.

The metallurgical studies should coincide with the initial designing and should include, usually, a careful stress analysis with accent on machining, fabrication, and heat-treating stresses. Tension and compression stresses in the final assembly under operating conditions are often difficult to analyze because of unknown processing stresses and yet no designer can proceed intelligently with a new design unless they can be estimated with some accuracy.

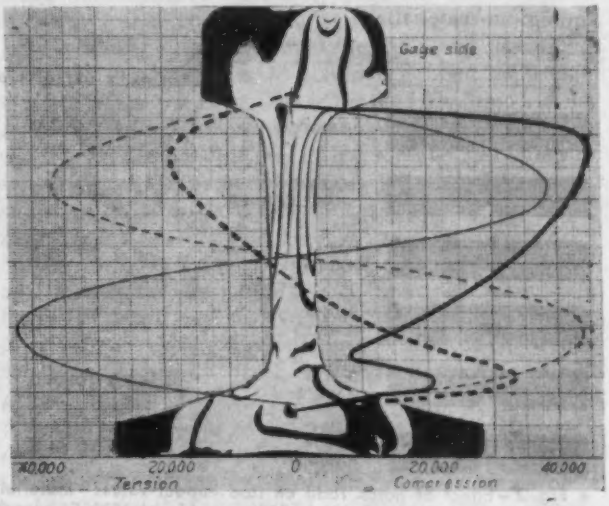
It should be the function of the well-trained metallurgical department to evaluate the type, the intensity, and the location of stresses in a given part resulting from machining, from welding, from heat-treating, forming, and assembling. The designer, on the other hand, should be able to determine the expected stresses in parts due to loading, and the effect on the intensity and location of these stresses in a given part of an assembly because of deflection under load or change in position due to temperature effect, and so on. The metallurgist is also supposed to indicate to the designer the ways and means of improving resistance in a given part to tension fatigue, to deflection, and wear.

Quite often, neither the metallurgist nor the engineer is qualified to supply the information required, and we have a case of the blind leading the blind. The net result is that the factor of safety, as well as the cost is increased, or the new design is put into its first production test with a prayer by the designer. Fortunately, the new design is seldom altogether new and generally represents a change from a previous design which had the "bugs" taken out of it by long use.

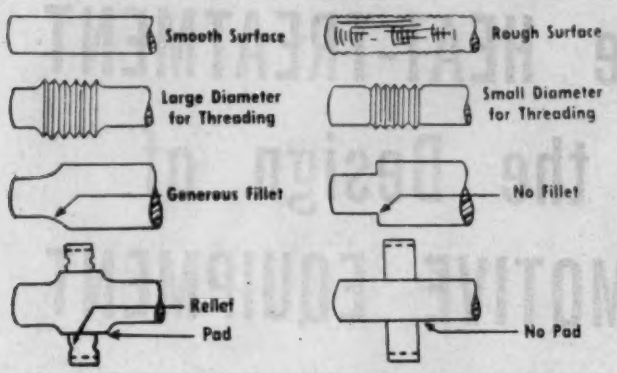
The experience which industry in this country has had in producing aircraft products, in which, by using a steel of known properties and guaranteed uniformity, the factor of safety is greatly reduced as compared to usual commercial parts, should affect the thinking of the average designer.

New parts for automotive and tractor production might well be designed in the future on the aircraft-engine basis with reduced factors of safety. The need for thorough knowledge of stress and strain location and intensity will become increasingly important. The help of the metallurgist will then be essential to the designing engineer as he starts out to evaluate conditions surrounding the performance of a new design, while the metallurgist, on the other hand, will need a thorough training in stress analysis and control.

We use polarized light in connection with a model of celluloid or bakelite as one means of indicating the loca-



■ Fig. 1 – Polarized light study of bakelite model of new rail design (Leaf – Denver & Rio Grande Railroad)



■ Fig. 2 – Design details that result in stress concentration – left: correct practice; right: incorrect practice

tion and, to some degree, the intensity of stresses in a given loaded part.

A transparent model of the part in question is made from either celluloid or bakelite, but usually of bakelite. The model is then placed in the field of a polariscope in such a manner that the rays are perpendicular to the plane of stress. If no stress is present the screen on which the light passing through the model is projected remains dark. If, however, the model is stressed while in the field of polarized light, the image on the screen alters radically. The screen, which is, normally, uniformly dark, now shows a pattern, which we will call a stress pattern, enclosed by the outline of the model. The shape of the curves in the stress pattern, their number, and complexity depend on the loads, dimensions, shapes, and material of the model and upon the nature of the light source. See Fig. 1.

By studying the intensity and characteristics of the stress patterns in the bakelite models under loads similar to the loads applied in service to the original part, it is possible to develop considerable knowledge of the location and intensity of the stresses incidental to service loading.

These studies have been very beneficial and deserve the most serious consideration on the part of the designer and metallurgist. They should be part of the development of new designs and should serve to indicate to the designer and the metallurgist where to concentrate their attention in the endeavor to obtain maximum performance from the minimum material.

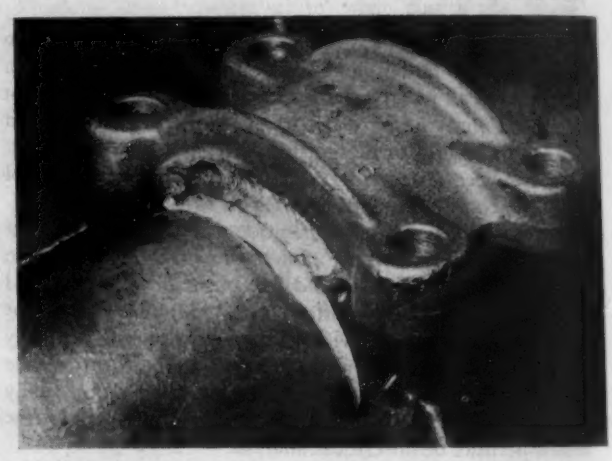
	IMPACT VALUE, F.P.
V NOTCH	2.5 TO 3.0
SQUARE NOTCH SHARP CORNER	2.5 TO 3.0
1 M.M. RADIUS	8.0 TO 10.5
2 M.M. RADIUS	8.0 TO 11.0
3 M.M. RADIUS	10.0 TO 14.0
4 M.M. RADIUS	10.0 TO 14.0

■ Fig. 3 – Effect of notch radii on izod impact values – material: SAE 2345; treatment: 45 min in 30% cyanide at 1500 F, quench in oil, draw for 45 min at 400 F

In modern mechanism, with the emphasis on close tolerances, accurate workmanship and finishes, as well as on hard, wear-resisting surfaces and high-strength parts, the most important single factor is rigidity. The part played by the modulus of elasticity in the performance of gears, bearings, shafting, and almost any highly stressed part is quite generally overlooked. Much more important than factors affecting performance, which the metallurgist can control, are the factors over which he has little control. These factors are deflection and localized stress concentration.

The performance of most modern, accurately made mechanisms is so governed by the factor of deflection that the rigidity of the mounting and the inherent deflection due the operation of Hooke's law must be always uppermost in the designer's mind. As the construction of mechanisms becomes more accurate; as we work to a tenth of a thousandth instead of a thousandth of an inch; as contacting surfaces become higher in strength and hardness; then, the modulus of elasticity becomes of increasing importance and must be considered at every step.

The steel maker or heat-treater cannot help because the modulus of elasticity of steel is, commercially speaking, independent of analysis or treatment. It is a factor in cast iron which can be varied considerably, and it is an important consideration in determining the selection of one metal (such as aluminum) against another (such as steel).



■ Fig. 4 – Failure due to unrelieved stresses set up in welding

Thus, the success or failure of a given mechanism is quite often governed by the design of its housing, or the distance between the supporting bearings. A few ribs added to a gear case are often more important in improving performance than a radical change in the type of steel or treatment of the gears or shafts themselves.

Deflection of the housing imposes unpredictable stresses in bearings, gears, and shafting. These loads are concentrated at points which are correspondingly overloaded. Thus, loads on roller bearings where shaft or housing deflection is present are concentrated on one end of the rollers in spite of the best efforts of the bearing designer. Many a good roller bearing has been condemned because of deflection in the housing or shaft. The same is true of gears when a weak housing or insufficient beam section in the supporting shafting causes bearing movement and a concentrated tension stress at the root of the teeth. This is a common cause of gear failure. Many a good gear has been condemned because of deflection in the housing or shaft.

The factor mentioned above is that of localized stress concentration. Here the metallurgist can exercise some slight control, but not nearly so much after a part is designed and in production as he could have by working with the designer when the part was still on the drawing board.

Notches or sudden changes in section, parts which tend to heat or cool unevenly, and many other design factors which tend to increase stresses or distortion in heat-treatment are important subjects for discussion before the final approval is given to any design which involves heat-treatment.

The effect of notches and surface defects is fairly well known to the average metallurgist, but not too well known to the designer. The war and, especially, aircraft requirements have brought home to many designers the importance of highly stressed parts of smooth surfaces, of an ample radius in corners, and of all possible avoidance of sudden changes in section. These factors in performance have been stressed for many years, yet they are still common causes of failure. The use of welding, with the necessity of relieving the tension stresses set up in the cooling of the weld, calls for cooperation between the designer and the metallurgist. Unrelieved tension stresses set up in welding, when added to the working load are often sufficient to cause failure in a short time. See Figs. 2, 3, 4, and 5.

The stresses incidental to heat-treating are often far greater than those due to calculated loading, and their extent and location must be predicted to secure sensible basis of design. Cold-working stresses, straightening effects, decarburization, machining stresses, stresses incidental to localized temperature changes, and many others, introduce uncertainty in design that would seem to make the development of the best design extremely difficult without the closest cooperation between the metallurgist, the designer, and the production engineer[1, 2].

Many cases have been known where the elimination of decarburized surfaces have more than doubled the life of a highly stressed part. The changing from cold straightening to hot straightening of heavily stressed axle shafts has, in some cases, much more than doubled their performance.

A full knowledge of the adverse effects of these causes of increased stresses or localization of stresses is necessary to understanding in designing. This is true in parts that are not heat-treated, as well as those which are heat-treated. Some very important reductions in failures of railway equipment have been made by recognizing the stress concentrations which are due to cold working and microscopic ruptures in rail and motive power equipment. Here also important improvements were made by recognizing the adverse effects of cold straightening and decarburization[3].

■ Developments in Heat-Treating

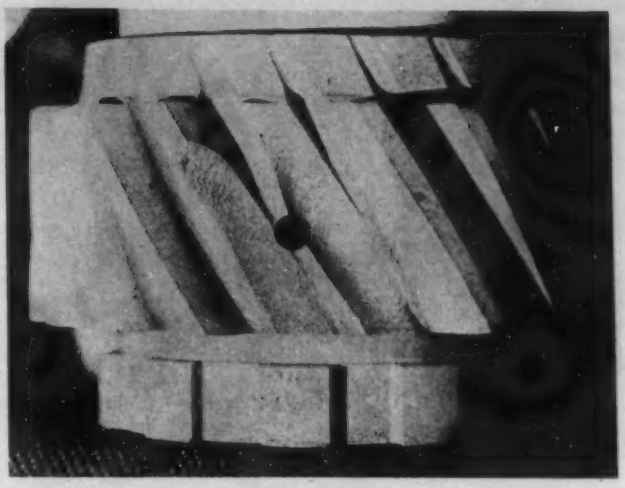
In discussing the strictly wartime developments in the heat-treating of steel, we might think for the moment of what has already been accomplished.

Many developments which have been under way for some time have made great progress under the pressure of wartime requirements and have reached, as a consequence, the commercial-application stage. These we feel must be considered as wartime developments. A partial list would include:

1. Induction and flame heating.
2. Isothermal heat-treating.
3. Improved quenching.
4. Shot blasting, and so on.
5. Controlled atmosphere.

The strictly wartime developments might be limited to the following:

1. Use of multiple-lower-alloy steels to replace simpler steels of higher alloy content.
2. Water quenching of alloy-steel armament.



■ Fig. 5 – Gear-tooth failure caused by concentrated stress of drilled hole

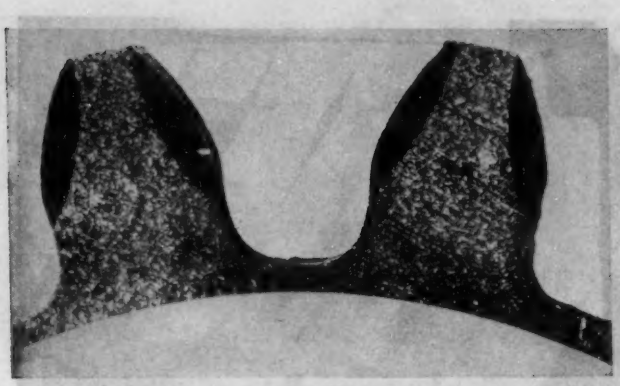
■ Induction and Flame Heating

The induction-hardening technique, in which heat is generated by induced electrical currents, is one of heating rather than hardening. These currents are controlled by their frequency and the power supplied. The surface being heated acts as the secondary of a transformer with the depth of the zone affected controlled by the frequency. The higher the frequency the shallower the zone affected, and the greater the power supplied the greater the power in the induced currents, hence, the greater the absorption of energy and speed of heating.

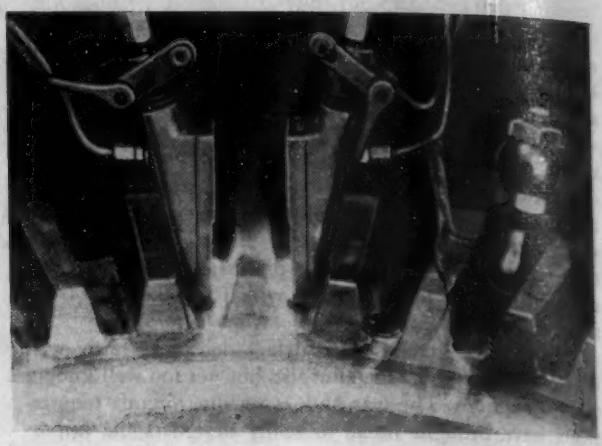
By designing appropriate induction coils, and controlling the frequency and power to give the required depth and speed of heating, it is possible to obtain areas heated to the depth desired in a matter of seconds. After making certain that the temperature and carbon solution and diffusion are satisfactory, it is only necessary to apply the quench and draw to obtain a heat-treated part within the power zone.

This method of heating has, of course, metallurgical and mechanical limitations which must be considered. Properly worked out, however, induction heating promises to give more leeway to the designer, and its scope and value should, consequently, be of interest to every designer. It permits in objects of simple symmetrical shape, such as pins and bushings, the use of lower alloy and higher carbon combinations, which can be water quenched without excess distortion or internal stresses. This means higher hardness and more useful available load-carrying capacity. The use of induction hardening should be primarily limited to shallow-hardening steels, that is, lower alloy steels, other-

[1] See SAE Transactions, Vol. 50, February, 1942, pp. 52-61: "Facts and Fallacies of Stress Determination," by J. O. Almen.
[2] See SAE Transactions, Vol. 51, July, 1943, pp. 248-268: "Shot Blasting to Increase Fatigue Resistance," by J. O. Almen.
[3] See *Metal Progress*, Vol. 40, August, 1941, pp. 189-191: "Railroad Failures in Track and Cars," by R. McBrian and P. A. Archibald.



■ Fig. 6 (above) – Flame-heated gear tooth



■ Fig. 7 (right) – Flame heating the gear tooth

wise the severe quench may result in a highly stressed surface, which, if it does not crack, has very much decreased load-carrying capacity[4, 5, 6].

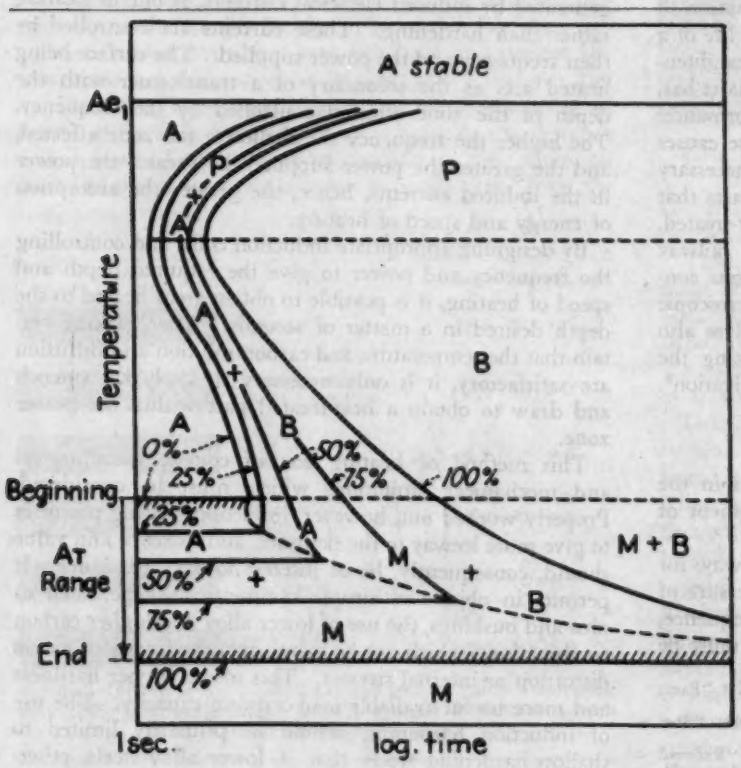
The application of induction heating is rapidly spreading as we become more familiar with its use and limitations. It can be used on internal as well as on external surfaces, and lends itself to "in line" applications. Its use can be made automatic with simple push-button control, and where properly applied it provides a finished product of very satisfactory performance.

The principal advantage of induction heating to the designer lies in its application to localized zones. It makes possible the design of parts in which it is not necessary to heat the whole piece, and thereby eliminating distortion from this cause. Parts can be designed so that the stresses in a finished piece, at a given point, can be relieved by local heating.

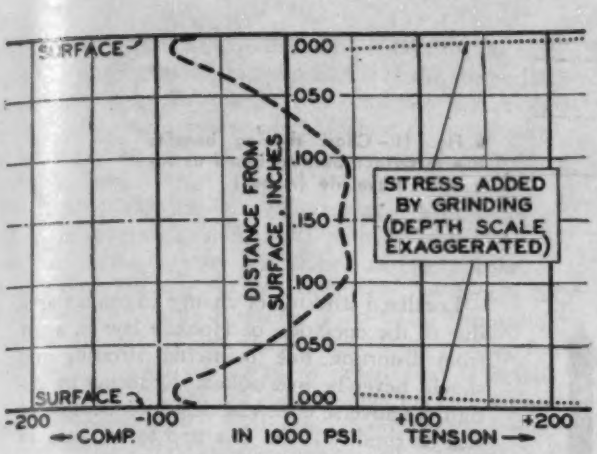
Parts can also be designed in which only internal surfaces or projections are to be hardened, so that welded or brazed assemblies can be built up prior to heat-treating. Induction heating has valuable applications in brazing operations, and the distortion incidental to overall heating of welded assemblies in heat-treating can be eliminated. The use of induction heating of gears with the heat applied only to the rim section permits wide latitude in built-up webs and easily machined hubs. The wide range of applications of induction heating make that subject an important one for the designer, and indicates the changes in design that might come from one who profits by a thorough study of its possibilities.

Much that has been said about the use of induction heating can also be said about flame hardening. Where there are fewer limitations due to the external nature of the heat application, as in large gears, a concentrated high temperature flame can be used which is brought directly in contact with the surface to be heated. This surface very rapidly reaches a temperature which is determined by the heat of the flame and the heat conductivity of the metal being heated. The rate of heating from then on proceeds at a uniform rate, and is governed by the same factors. The tendency is for somewhat deeper heating than by induction. See Figs. 6 and 7.

Not all the advantages possible with properly applied induction heating are available in flame hardening, but for some applications there is little choice between the two methods. Every possibility should be carefully studied. If flame hardening will satisfy, there is the advantage of much lower cost of installation. In any event, the metallurgical and operating aspects involved in the use of induction- or flame-heated parts should be obtained by the designer before deciding on the design or the equipment.



■ Fig. 8 – Modified "S" curve – full lines represent per cent of austenite transformed; *A*-austenite; *P*-pearlite; *B*-Bainite; *M*-martensite or tempered martensite (Shepherd)



■ Fig. 9 – Magnitude and depth of residual stress due to carburizing and hardening (Almen)

The advantages of induction heating in permitting localized softening of hardened parts should be familiar to every designer in connection with the attachment of hardened parts to other parts, in the relief of stresses set up in fabrication, and in the possible forming of heat-treated parts. As an example of the first case, we might take an operation in which a hardened flange is softened to permit drilling and tapping. As an example of the second case, we might take an operation in which the induction heating is used to soften the bases of armor-piercing shot. In the last case, we can use, as an example, the heating of the nose on the heat-treated high explosive shell to permit nosing in. Similar applications can be found throughout industry which permit changes in design to facilitate processing or assembly.

■ Isothermal Heat-Treatments

While isothermal heat-treating was practiced many years ago, its possibilities were largely undeveloped until the advent of the constant-temperature transformation curves of Bain and Davenport,[7] about 1930. This work led to the development of constant-temperature transformation heat-treating practice as a commercial operation. This practice, because of limitations of size and analysis, was usually confined to small sections and carbon or low-alloy steels.

The publication of studies for many grades of alloy steels showing the time required at all temperatures for the transformation of the austenite to ferrite and the precipitation of the carbon in form of a carbide to form martensite or other structures led to application of this knowledge in commercial heat-treating. Thus, it was found that, after the properly heated part had been cooled past the "knee" on the "S" curve without transformation (Fig. 8) there was no great hurry in cooling, and parts could be straightened without unduly decreasing their hardenability.

When the austenite transforms and carbon is precipitated to form the structure of maximum hardness known as martensite, the change is accompanied by a definite increase in volume. It is this increase in volume which

[4] See Transactions of the Electrochemical Society, Vol. 79, 1941, pp. 215-241: "Surface Hardening by Induction," by H. B. Osborn, Jr.

[5] See *Welding Journal*, Vol. 20, February, 1941, pp. 105-107: "Preheating – Welding – Normalizing by Induction Hysteresis," by C. J. Holslag.

[6] See "Induction Heating," by E. Blasko. American Society of Metals Preprint No. 48 for meeting held Oct. 21-25, 1940.

[7] See Transactions of the American Institute of Mining and Metallurgical Engineers – Iron and Steel Division, Vol. 90, 1930, pp. 117-144 + (disc.) 144-154: "Transformation of Austenite at Constant Subcritical Temperatures," by E. S. Davenport and E. C. Bain.

■ Fig. 10 – Pitting in gear tooth due to subsurface tension stress – failure progresses to surface and parallel to surface

contributes to most of the stresses set up in quenching. Since cooling proceeds from the surface to the center of a piece, if the cooling is drastic, as in water, there will always be at the time of center transformation a considerable difference in temperature between the outside and center. Thus, with the outside of the part cold and rigid and the center expanding, surface tension stresses will be set up which make any assumption of the designer very uncertain. This, of course, is true primarily of a deep-hardening steel, and any steel which hardens throughout in oil or water will be subject to tension stresses in the outer portion, which may actually cause rupture in the form of quenching cracks.

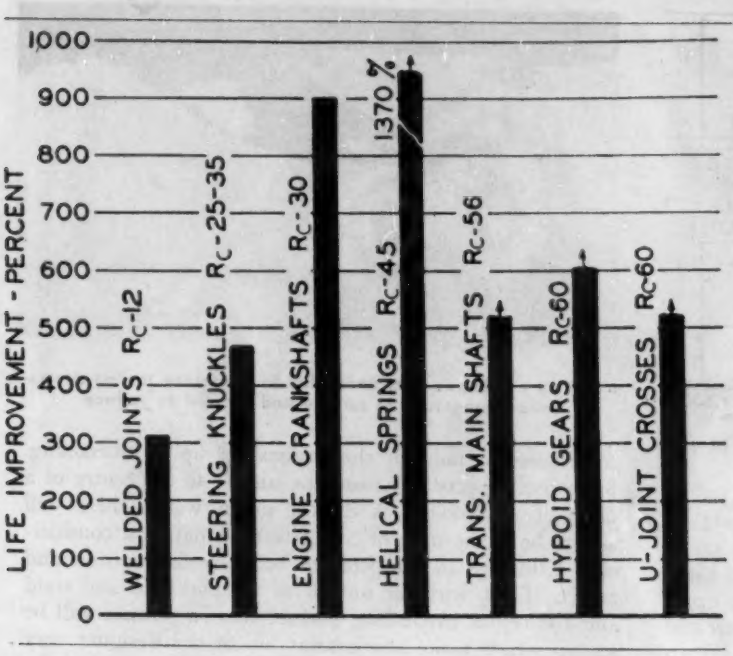
The designer is not much interested in those parts which actually crack in quenching, but he is very much interested in the intensity and distribution of stresses throughout any given part; for no formulas used by the designer cover the variation and amount of stresses incidental to heat-treatment. Since, as Almen has stated, fatigue failures only occur in tension, the presence of unknown tension stresses constitute a direct hazard. Every attempt should ordinarily be made to obtain initial compression stresses if possible. It is necessary to use care in some cases to avoid excessive compression stresses in the surface zone since they must be offset by tension stresses at a greater distance from the surface. In the case of rolling contact, as in gears or roller bearings, the fatigue failure may actually take place by subsurface tension if the surface compression loads are excessive. A thorough study of Almen's work is recommended to all designers and metallurgists. Each part is a special case, and in some cases considerable judgment is required to obtain high compression stresses where desirable and to avoid them where undesirable. See Figs. 9, 10, and 11.

Internal stresses incident to uneven cooling and nonuniform transformation are greatly aggravated by nonuniform sections and by sudden changes in section, so that these should, if possible, be discussed by the designer and metallurgist before a given design is adopted.

Internal stresses may not necessarily be severe enough to cause direct fatigue failure, but they are always the primary factor in distortion, which often governs the stock removed in grinding as well as the performance of the given part.

Allowances are usually made for distortion in designing heat-treated parts by increased size or extra processing. This is especially true in gear tooth design where allowance is made for tooth movement in heat-treating by extra length of tooth face and heavier teeth.

Distortion in heat-treating is becoming of increasing



■ Fig. 11 – Chart showing benefits due to surface peening based on fatigue life (Almen)

importance to the designer because of the unknown it introduces in determining contact areas and because it indicates unknown internal stresses which make accurate designing impossible. The development of heat-treatments which minimize internal stresses and resulting distortion promises to become an important factor in promoting more accurate designing.

An important contribution to our heat-treating practice was made by Shepherd,[8] who determined that molten salts of certain types would cool at about the same rate as a good quenching oil. This permits their use in cooling oil-hardening steels fast enough to ensure their getting past the knee on the "S" curve untransformed. If the bath is at a temperature of 400-500 F, the part can be brought to this temperature uniformly throughout before transformation starts. After a uniform temperature has been obtained, the part can be cooled in air and hardens with the maximum of uniformity and the minimum of internal stresses. This practice is known as martempering.

The underlying factor governing the results obtained by the process is that:

(a) Uniformity in temperature throughout a part is approached before transformation starts.

(b) Transformation takes place at almost the same time throughout a given part, thereby preventing high stresses due to nonuniform volume changes.

The development of such a practice, while requiring perhaps the use of a deeper-hardening steel, will permit the designer to assume for many parts that grinding of highly loaded surfaces can be done before heat-treating or eliminated altogether. He can assume that the uncertain internal stresses of oil- or water-quenched parts can be reduced to a point where he can estimate accurately the useful strength of a given stressed area. Thus, an important step in precision designing can be taken.

[8] See *Iron Age*, Vol. 151, Jan. 28, 1943, pp. 50-52; Feb. 4, 1943, pp. 45-48: "Martempering," by B. F. Shepherd.

[9] See *Iron Age*, Vol. 151, June 24, 1943, pp. 44-47, 130; July 1, 1943, pp. 48-54; July 8, 1943, pp. 74-77; Vol. 152, July 15, 1943, pp. 70-77; July 22, 60-67: "Annealing of Steel," by P. Payson.

[10] See ASM Transactions, Vol. 31, March, 1943, pp. 218-245 + (disc.) 245-256: "Hardening of Tool Steels," by P. Payson and J. L. Klein.

Localized stressing or change in contact area due to the operation of Hooke's law is, apart from distortion, due to internal stressing and should never be overlooked. Reducing by design the adverse effects of deflection because of lack of rigidity is the first step to be taken in obtaining precision in designing.

Martempering in all its phases and possibilities should be thoroughly developed by the metallurgical engineer, and he should demonstrate its advantages and make them familiar to the designing engineer. There is a possibility that with this type of treatment much of the allowance now made for distortion in heat-treatment and, in many cases, corrective grinding can be eliminated. This permits the designer to design more closely to theoretical by decreasing dimensions or to increase loading on designs in operation. Combinations of martempering, induction heating, or induction softening will possibly permit designs now considered impractical. Built-up combinations by welding or brazing can possibly be made with grinding eliminated from surfaces of high hardness, if this is desirable. See Fig. 12.

The isothermal transformation of highly alloyed steels as a means of annealing is being developed, and promises to reduce the aversion of the production departments to this type of steel. There always has been difficulty with the more highly alloyed steels due to reduced machinability, which restricts their use. The isothermal annealing promises to encourage their use because of the substantial time reduction and the greater uniformity of results.[9, 10]

This means that these steels might be used in many intricate parts where they offer better performance with possible reduction in size or increase in loading. If the isothermal annealing of the higher alloyed steels is combined with their better ability to process through the martempering cycle, then the designer should be able to design for higher loadings than ever thought safely possible.

The martempering cycle is especially suitable to the higher alloyed steels and can be combined with short cycle tempering baths to give maximum properties, minimum distortion, and internal stresses, together with the minimum overall heat-treating cycle. These developments promise to affect the design of almost every mechanism in the future. Higher strength, better contacts, and lower unit contact pressures permit smaller parts with closer spacing of supports, smaller housing sizes, and hence decreased deflection, under loading.

The criterion of performance of the newer designs, with increase of loading, promises to be the bearings. This indicates the possibility of heavier demands on the bearing manufacturer for increased performance.

■ Controlled Atmosphere Heating

It is usual for the designer to predicate his design on his experience with parts used in similar applications. Where

heavy fatigue stresses or severe loading is present, the size of parts is usually considerably in excess of the theoretical requirements. Some blame for this condition lies in the prevalence of a soft, low-strength surface on most parts that are operating with applied stresses in the as-forged or as-rolled surface.

This is often the case in such items as springs, steering arms, connecting rods, and bolts. With ordinary practice these parts have in their unmachined surface a zone of almost carbon-free steel, with a relatively low tensile or fatigue strength. This soft surface is due to the removal of carbon by the products of combustion in the heating for the processing operations. Extensive experiments show that in many applications the low-strength decarburized surface is the controlling factor in the performance of the part. It is for this reason that the development of controlled atmosphere furnaces has been greatly accelerated during the past few years. In many cases, the decrease in size resulting from machining off the decarburized surface zone would be accompanied by an improvement in performance which would possibly pay for the machining operation.

It is believed that the studies already made indicate the potential value of the removal of the soft surface layer as a most important means of permitting a lower factor of safety and obtaining increased performance. This one phase of metallurgical activity should be considered carefully by every designer of parts that now give less than the desired performance.

The production of drop forgings free from decarburization has not received the attention it should, primarily because the metallurgist has failed to emphasize the possibilities to the designing engineer.

The steel mill and drop forge furnace offers great room for improvement in respect to decarburization, but in many cases the machining away of the decarburized surface in the final part is cheaper than installing atmosphere control on the high-temperature heating furnaces in the steel mill and forge shop.

Recent developments indicate that the recarburization of low carbon or carbon-free surface zones is quite possible, and seems to offer an interesting field for improvement in performance of many applications. When subject to severe reversals of stress in bending or torsion, the strength of the surface zone is the strength of the piece, and few of us realize how low this strength is, or how much money has been wasted on alloys in heat-treated parts where the criterion of performance lies in the low strength of the decarburized outer layer of the part.

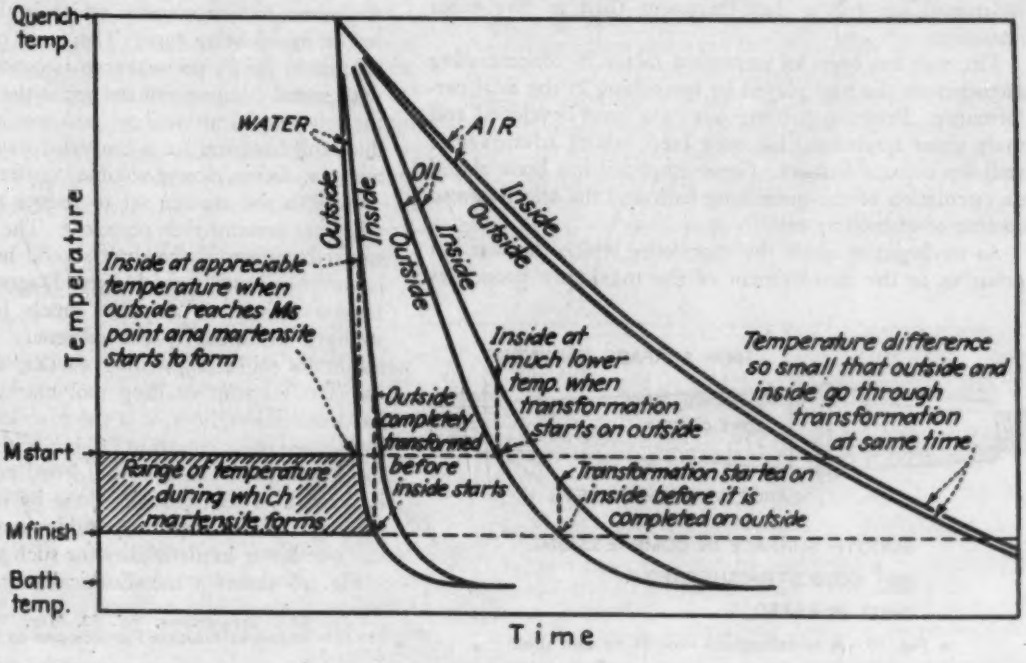
■ Improved Quenching

As noted before, the designing engineer is working continually with what is known as strength of materials, and is interested primarily in the usable or net strength of any material under load. In the case of steel, he has been taught to place great reliance on the results of laboratory tensile tests which presume to indicate the strength of the steel part in service. These tests are hardly more than academic in their value, especially in heat-treated parts. They do not indicate the internal stresses in a surface before the working load is applied. Those internal or initial stresses which are due to cold working, volume changes in heat-treating, machining, straightening, and assembly stresses are often far greater than the applied or working load.

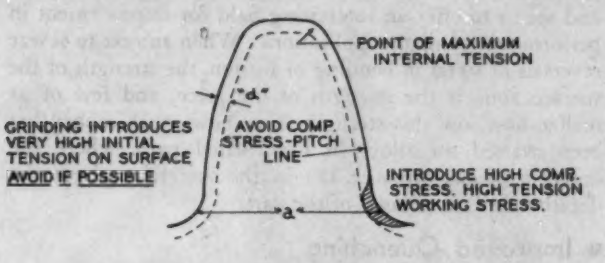
The useful net working strength is the difference between the internal or initial stress and the strength of the steel, as indicated by the tensile test.

In some cases the available or useful strength may be considerably greater than the strength indicated by the tensile test – in most cases it is less. To have the initial strength, in a given part, useful strength, and thereby increase the capacity above that indicated by the tensile test, should be the goal of the designing engineer and the metallurgical engineer. This is especially true in parts subjected to severe and repeatedly applied loads where early fatigue failure threatens.

It is to accomplish this that the engineer is striving to introduce initial compression stresses where highly concen-



■ Fig. 12 – Cooling rate of outside and center of similar rounds quenched in water, oil, and air (Shepherd)


POINT OF MAXIMUM INTERNAL TENSION
"d"
GRINDING INTRODUCES VERY HIGH INITIAL TENSION ON SURFACE
AVOID IF POSSIBLE
AVOID COMP. STRESS-PITCH LINE
INTRODUCE HIGH COMP. STRESS. HIGH TENSION WORKING STRESS.
"a"

"a" SHOULD BE MAXIMUM - TO REDUCE TENSION LOADING AT ROOT OF TOOTH. CASE DEPTH "d" AND CORE HARDNESS GOVERNED BY TYPE OF FAILURE. INITIAL FAILURE SHOULD BE UNIFORM PITTING ALONG PITCH LINE.

■ Fig. 13 – A metallurgical view of a gear tooth

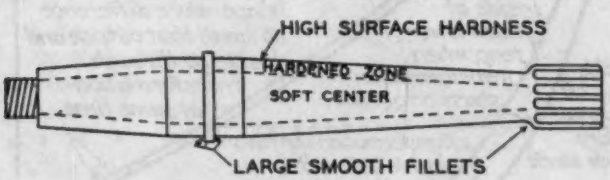
trated tension stresses are developed in service. As so well explained by Almen, this can be done mechanically by cold working the surface or metallurgically by case hardening, nitriding, or the use of shallow-hardening steels.

The use of shallow-hardening steels, wherever possible, is a contribution which the metallurgist can make to the designer. Their primary advantage lies in the reduction of unfavorable internal stresses, and in many cases the development of favorable stresses, especially in members subjected to severe tension or bending loads which tend to cause early fatigue failures because of high tension stresses concentrated at the surface. Such parts would be axle shafts, springs, cranks, connecting rods, or steering knuckles.

The use of shallow hardening to reduce tension stresses in the surface section should not be confused with poor or retarded quenching. In fact, to obtain the advantage of shallow hardening it is especially necessary that the structures in the hardened portion be those which indicate thorough and complete quenching. This is the first requirement for high useful strength in service. In fact, it might be said that, whether he knows it or not, everyone, from the designing engineer to the final user of a critically loaded part, is more interested in the structure obtained in quenching during heat-treatment than in any other operation.

The war has been an important factor in concentrating attention on the part played by quenching in the final performance. Better quenching oils have been developed and more water quenching has been used, taking advantage of well-worked-out fixtures. Great emphasis has been placed on circulation of the quenching bath and the selection and cooling of quenching oils.

As we begin to apply the experience which the war has given us in the development of the maximum properties which come with better quenched structures, the designer can begin to design closer and closer to the theoretical minimum. The maximum improvement in this direction can only come when, as pointed out by Eddy,[11] the hardenability of the steel being quenched is of maximum uniformity.



SMOOTH SURFACE IN COMPRESSION
NOT COLD STRAIGHTENED –
SHOT BLASTED

■ Fig. 14 – A metallurgical view of an axle shaft

■ Difficulties in Application of Technique

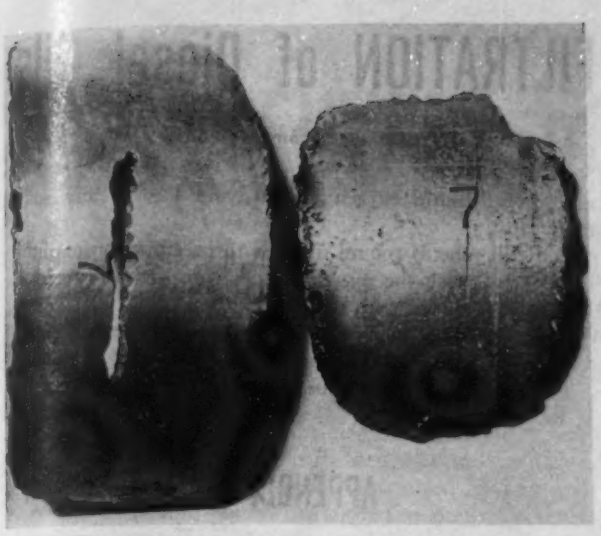
To take full advantage of the improvements in control that have been developed in heat-treating and the new knowledge regarding hardenability and time-temperature transformation characteristics which we have available, it is necessary for the designing engineer, the metallurgical engineer, and the operating department to cooperate in removing some factors of uncertainty which are, at present, all too common.

There is still some difference of opinion, apparently, among metallurgists as well as designers regarding the methods used for arriving at the evaluation of some of these simple factors. Thus, in the case of hardened gears, the question of case depth versus core hardness is often a matter of opinion and tradition. In Fig. 13 are shown some of the factors which the metallurgical engineers might consider of primary importance in starting to design a gear. The metallurgist would rather see a gear fail by uniform pitting at the pitch line than by a tension fatigue failure at the root of the tooth. This means that if the beam section of the tooth is sufficient the metallurgist can work out the resistance to pitting and wear by decreasing the compression load due to deep cases and low strength cores. He might vary the heat-treatment and core hardness, which would give him improved performance in rolling and sliding contact, but which would adversely affect the strength due to high tension at the root of the tooth. This again might be due to gear tooth movement in heat-treating or deflection in operation. The newer techniques in heat-treating give promise of reducing tooth movement. The deflection can only be corrected by design.

In Figs. 14 and 15 an axle shaft is shown with some notes on the factors of concern to the metallurgist. Here we have a torsion member which is also loaded in bending in the semifloating type. The main concern of the metallurgist, as far as performance is concerned, is to start out with initial compression stresses at the surface. This means the selection of an analysis and treatment which will produce full hardness for a limited distance below the surface with a softer, lower-volume center. The compression stresses in the surface act to reduce the working load by whatever amount they represent. The surface of the highly stressed portion of the shaft should be free from ruptures, however minute, which would serve to start an early fatigue failure at a point much below the estimated strength assumed by the designer. One cause of these might be cold-straightening checks; another might be an inspector's stamp or deep tool mark. If the shaft is not hardened throughout, it is the function of the metallurgist to increase the strength in fatigue of the shaft by increasing the hardness of the fully hardened zone if more strength is desired. This is usually done by increasing the carbon content and indicates a possible trend to higher carbon steels of lower hardenability for such applications.

Fig. 16 shows a metallurgical reaction to some funda-

[11] See SAE Transactions, Vol. 52, May, 1944, pp. 169-180 + (disc.) 180-182: "Effect of Wartime Developments on Future Steels," by W. P. Eddy, Jr.



■ Fig. 15 – Concentration of stress by inspector's stamp – oil-well drill pipe fatigue failure

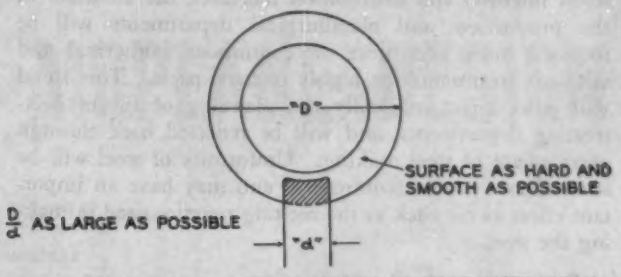
mentals of a worm and wheel. Here the newer techniques of heat-treating should make possible worms of sufficient accuracy to eliminate grinding of the final hardened worm thread surface. The important fundamental from the metallurgical standpoint is to keep the worm diameter a maximum, the worm tooth surface as smooth as possible, and, most important of all, to keep the worm wheel thickness to a minimum. This restriction on wheel width is to take advantage of the improved contact at the center of the worm, the wide tolerance in dimension as compared to a wider wheel, and the easier lubrication problem. The sensitivity of adjustment with worm operation seems to increase as the square of the wheel width. The problem really becomes one for the metallurgist, whose function it is to recommend a material in the wheel with sufficient strength and bearing qualities to operate with an available lubricant. This material is probably present in some of our harder high-strength heat-treated bronzes.

Fig. 17 shows the metallurgist's viewpoint in approaching a connecting-rod design. Much of the same thinking is applicable to steering arms, knuckles, springs, and so on, where as-forged parts are highly stressed. Here, smooth machining or grinding would seem the only way to approach maximum uniformity in properties of the highly stressed surface. This is the surest way to remove low-strength decarburized surfaces and notches incidental to forging.

■ Effect on Production

The effect of wartime developments in the heat-treatment of steel on the production of automotive equipment will be felt directly in the increased use of induction- and flame-heating equipment which will permit line processing. The use of this type of equipment will be of value also in the relief of welding stresses in built-up parts and the localized annealing or softening of parts to facilitate assembly.

The increased use and experience with induction and flame heating will be also reflected in its effect on case-hardened applications. As induction heating progresses there will probably be a trend to replace the long and, in some cases, more expensive case-hardening operating by induction heating.


"D"
SURFACE AS HARD AND SMOOTH AS POSSIBLE
D/d AS LARGE AS POSSIBLE
"d"
"d" = WHEEL WIDTH TO GIVE FULL CENTER CONTACT WHEN FULLY LOADED. BRONZE AS HARD AS POSSIBLE WITH HIGH YIELD.

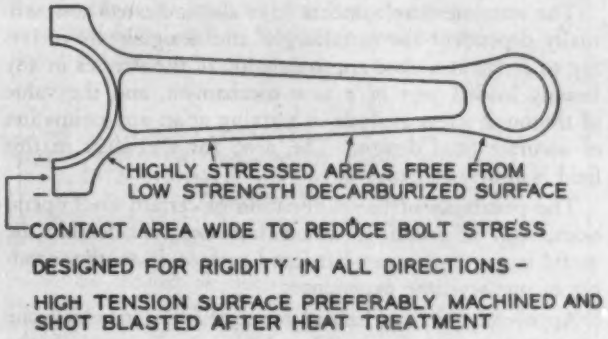
■ Fig. 16 – A metallurgical view of a worm and wheel

There is some difference of opinion as to the relative qualities of case-hardened and induction-hardened parts. The initial compression stresses introduced by case hardening are probably somewhat greater, so the better wear resistance due to higher carbide concentration in the case may be a deciding factor. In any event the trend to increased application of the induction-heating and flame-heating processes promises to continue and will greatly affect the layout of the automotive heat-treating plant of the near future.

The use of salt-bath, isothermal, and martempering treatments will be of value in processing highly stressed parts, such as gears, axle shafts, and, especially, in parts where distortion is a problem. This type of equipment will, in most cases, have to be developed, as it is still very much in the laboratory stage. Its value in reducing over-all time of heat-treatment, plus the possible elimination of grinding hard, highly stressed surfaces, will, in many cases, outweigh its increased cost. The combining of such operations with treatments to eliminate decarburization and to control initial stresses should certainly permit designing closer to the theoretical size. This will, in turn, affect the entire automotive processing because of the possibility of drastic reduction in the size of transmissions, rear-axle gears, and other highly stressed parts. It is believed that the future offers some important changes in this direction.

In the case of truck or tractor parts, the improved load-carrying capacity will also be reflected in smaller parts, though more probably in increased load capacity.

As pressure is exerted to obtain the maximum useful strength in heat-treated parts, and as our knowledge of


HIGHLY STRESSED AREAS FREE FROM LOW STRENGTH DECARBURIZED SURFACE
CONTACT AREA WIDE TO REDUCE BOLT STRESS
DESIGNED FOR RIGIDITY IN ALL DIRECTIONS –
HIGH TENSION SURFACE PREFERABLY MACHINED AND SHOT BLASTED AFTER HEAT TREATMENT

■ Fig. 17 – A metallurgical view of a connecting rod

stress intensity and distribution increases, the attention of the production and metallurgical departments will be focussed more and more on continuous isothermal and salt-bath treatments for highly stressed parts. This trend will, also, affect markedly the planning of future heat-treating departments, and will be reflected back through every phase of steel making. Uniformity of steel will be stressed as a major requirement and may have an important effect as far back as the melting practice used in making the steel.

■ Summary

Recent developments in heat-treating and metallurgical processing indicate that the prospects are good for the production of heat-treated parts with greatly decreased distortion, with improved useful strength at the point of maximum load application, and with the high hardness area localized to ensure simpler fabrication in assembly. This should, with sufficient uniformity and with proper selection of steel analysis, permit a decrease in size or increase in loading. There will, however, be no change in the modulus of elasticity so that, unless this is taken into account, the problem of deflection will remain and little or no actual improvement in performance may be possible. This would suggest the value of decreased size where possible, rather than increased loading, since decrease in size could mean closer spacing of supports with decreased deflection. The decrease in size should also make possible more rigid housings with again decreased deflection.

The possible improvement in load-carrying capacity can be traced directly to the increase in our knowledge of transformation phenomena as developed in the isothermal and martempering treatments.

It is related directly to more knowledge of stress intensity and location, and the control or reduction in internal or initial stresses in the heat-treated part.

It is also directly related to improved structures resulting from better quenching, from better steel selection, and from control of decarburization.

It is due directly to better control of distortion with increased areas of stress application, and hence lower unit stresses because of change in shape in heat-treatment.

The wartime developments in heat-treating have indicated the possibility for improved design of many mechanisms on the basis of aircraft-engine experience, and the necessity for much closer cooperation between the designing engineer, the metallurgical engineer, and the production engineer during the initial development of any important design.

The wartime developments have also indicated how mutually dependent the metallurgist and designer are in trying to arrive at a close approximation of the stresses in any heavily loaded part of a new mechanism, and the value of thorough stress analysis in arriving at an approximation of accurate final design. The need for specialists in this field is clearly shown.

The possibility of the elimination of certain final operations, such as grinding, which decreases considerably the useful strength of a case-hardened surface, is another product of our wartime experience.

Apparently, we are headed for a new era in which our expanded knowledge will result in an increase of the actual useful strength of highly stressed parts and thereby permit an important reduction in the "factor of ignorance."

FILTRATION of Diesel Oils

continued from page 585

amount of liquid to remain to avoid loss of any solid material.

8. Refill tube to 100-ml mark with precipitation naphtha in order to wash the remaining oil from the solid residue. Mix solid material with naphtha by hand.

9. Shake tube for 3 min on mechanical shaker.

10. Whirl tube for 15 min in centrifuge.

11. Rinse outside of tube and dry in 150 F oven for ½ hr.

APPENDIX II

Determination of Dirt Index of Used Lubricating Oil

(A method of obtaining a rapid and approximate indication of the solids content of used lubricating oil)

Material Required

1 General Electric exposure (light) meter, calibrated to read in foot-candles.

1 100-w incandescent lamp bulb.

1 rheostat for varying light intensity of lamp.

Supply of white blotters.

Supply of Whatman's No. 1 filter paper cut into 1¼ x 4 in. strips.

Procedure

1. Mount exposure meter rigidly on solid foundation.

2. Arrange lamp directly over meter window on adjustable mounting and connect in series with rheostat.

3. Adjust lamp by changing either its location, voltage, or both to obtain 70-ft-c reading on meter.

4. Immerse filter paper strip for period of 30 sec in representative sample of oil to be tested. Remove excess oil from strip and blot lightly between two sheets of clean, white blotting paper.

5. Completely cover window of light meter with oil-treated strip and read intensity of light passing through strip.

6. Repeat steps 4 and 5, using new, clean oil of the same type as that under test.

7. Divide reading obtained with test oil by that obtained with new oil and multiply by 100. This value is termed the dirt index.

8. Note: With exceptionally dirty oils which have a dirt index less than 5, the scale can be considerably extended by diluting one part of the dirty oil with four parts of water-white mineral oil.

12. Cool in dessicator to room temperature.

13. Weigh.

14. The net weight increase multiplied by 100 and divided by the weight of the oil used (45 g) is the per cent of solids.

15. Note: The above procedure is not suitable with water present in the sample. To eliminate the water, the wash is made up of 50% naphtha and 50% acetone. This naturally changes the composition of the solid residue.

FUNDAMENTALS of AIRPLANE DESIGN

by A. L. KLEIN
Design Consultant
Douglas Aircraft Co., Inc.

DURING the expansion of the aircraft industry it is very easy for us to forget the early struggles and hard-won lessons of the past. We are now immersed in long and elaborate discussions of the merits of this or that manner of making some piece of accessory equipment and never seem to question the advisability of doing away with the gadget entirely. The writer through his contacts with young engineers and students has noticed that they frequently lose sight of the principal objectives in aircraft design and construction. This has left him with the feeling that these objectives and the hard, underlying principles of sound engineering need a restatement.

The three underlying fundamentals of aircraft engineering that must influence every design decision are: safety, weight, and serviceability. The great importance of these factors has caused the aircraft industry to set up and staff enormous engineering departments to do a detailed engineering job with a thoroughness not required in any other industry. The safety of present aircraft is due to the continuous vigilance and efforts of the government agencies, the airlines, and the manufacturers. The present high level of reliability has only been achieved by these strenuous efforts. The designer must not relax or these past gains will be lost. Many people seem to feel that airplanes crash only when some major catastrophe occurs. Even a cursory study of accident records will disabuse one quickly of this opinion. Airplanes have crashed because the pilot dropped a microphone, or because he happened to kick a small lever in the floor. Several accidents have been due to the malfunctioning of comparatively minor accessory equipment.

The young engineer is likely to feel that the device that he is designing is so fine and wonderful that it deserves the utmost care and consideration by the flight crew, the ground personnel, and even by the enemy. He is terribly shocked to discover that his brain child is kicked around and given the same type of care as are tools supplied with second-hand automobiles. The sound and only safe attitude to take in the development of any gadget for an airplane is to assume that it will be neglected, lubricated with the wrong oil, maladjusted, and misused. If the device will still function under these circumstances, then it may be satisfactory; if it does not, it will be potentially dangerous. As every device and piece of equipment is subject to failure, the results of such failures must be considered. If they can't cause accidents, the device may be satisfactory.

Failures of maintenance personnel to replace parts may be as much an engineering failure as are those due to poor design. In many cases designers have made parts so that it is impossible for inspection personnel to determine if these parts are properly installed. In such cases, failures due to the omission of necessary bushings, bolts, and so on, can properly be blamed upon the carelessness of the designer. The designer must make every effort to be sure that there is only one possible method for assembling the component, and that, the proper one. Assembly and adjusting sequences which involve procedures in many locations in the plane are always dangerous since it is easy for some items to be left unattached or improperly adjusted.

It took the industry 10 years to learn how to make multi-engined airplanes that would be safe after the failure of

[This paper was presented at the SAE National Aeronautic Meeting, New York City, April 6, 1944.]

DESIGNERS and vendors of aircraft parts need both foresight and pessimism, Dr. Klein feels, if they are not to violate the fundamentals of safety, weight, manufacturability, and serviceability in manufacturing aircraft.

Safety requires that gadgets work not only if they are treated properly but even if they are neglected or misused.

Weight is so important that it justifies an incredible amount of work in engineering and the shop to save it.

Airplane manufacturers are harassed by several problems that must be kept in mind by designers. The manufacturer must schedule and keep track of a great number of parts. The parts must fit after they are made. Some operations cause parts to expand, others to shrink, making the problem of assembly a major one—made even more difficult if there are serious deflection troubles.

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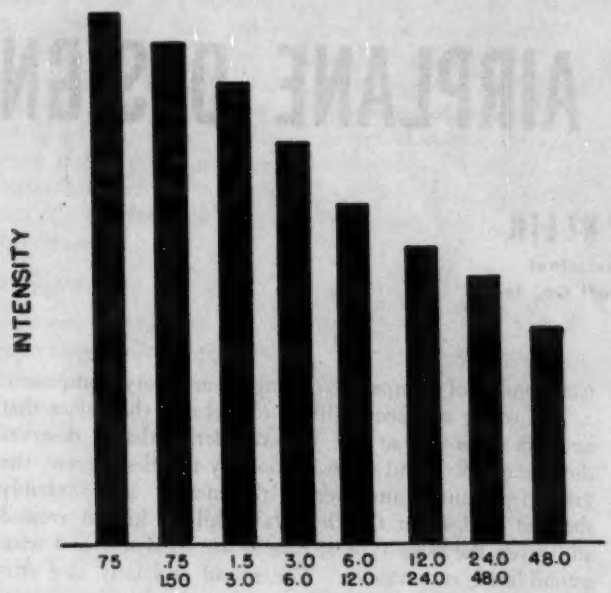


■ Fig. 1 – Distribution of excitation energy (DC-3 cabin)

one engine; this achievement has turned out to be of fundamental commercial and military importance. The merit of American military aircraft lies, in part, in their ability to fly after some of the engines have been shot out.

The airplane industry's wide use of cable controls in place of push-rods and torque tubes is another example of the necessity of avoiding mechanical parts that can fail suddenly and without warning. Some of the services tried to force a wider use of push-pull controls, but the hard facts of military experience forced them to revise their requirements and to insist on cables. In airline transports, where excellent supervision and carefully kept records are available, push-rods have given continual trouble. Even push-rods in the cockpit have caused difficulties. These difficulties might have been foreseen by a brief study of the vibration spectrum of the airplane which would have indicated the impossibility of avoiding resonance and inevitable failures. See Fig. 1. Push-rods, after failure, frequently form ratchets and so violate the principle of "failing safe." Steel cable is an ideal type of aircraft device: It is a multiple structure; it has internal damping; and long before it fails completely, it gives an easily detectable indication of potential danger. Furthermore, cables seldom jam and, therefore, permit the operation of the mechanism by an alternate or auxiliary means.

As every device and part known to man will fail sooner or later, it is imperative that the designer make such failures noncatastrophic. Thus, landing gears need auxiliary lowering means, brakes need stand-by power, and every engine has two magnetos. Cable systems are duplicated on controls that are not themselves duplicates, and all other possible means of securing duplicate or emergency operation of essential devices are taken. All devices essential to the safe operation of the airplane must not only have stand-bys, but also, they should not depend on the satisfactory operation of some auxiliary source of power. The only really reliable source of auxiliary power is the muscles of the pilot. Every other source of auxiliary power makes the airplane less safe by introducing more possibilities of failure. The engine builders follow this principle; they not only do not depend on the ship's electrical system for their ignition but they duplicate the magnetos in order to be sure. In spite of these precautions, the ignition system causes a considerable portion of engine failures. However, this principle of duplication must not allow the designer to relax his effort to increase, by improved design, the reliability index of the primary means of operation.

The high standards of engineering integrity developed by the aircraft manufacturers must be applied to accessory devices offered for aircraft. Practically no commercial devices are tested through the whole range of temperatures, pressures, and vibration common on aircraft. The principles of safety after failure are not adhered to. Practically no aircraft electrical devices have two limit switches, although many of them have been wound up into knots through limit switch failures. The poor practices of many designers in not detailing the assembly of their components is one of the principal causes of the unsatisfactory reliability of their equipment. The writer has yet to see a radio set mounted on a set of shock mounts in such a manner that the mounts do not defeat their own purpose by multiplying the vibration rather than reducing it.

■ Weight

In studying the effects of weight in aircraft it is convenient to use commercial statistics. The writer will confine his remarks to commercial aircraft, although he does not believe and would hesitate to say that weight in military airplanes is less important than it is in a commercial airplane. The only reason for using commercial data is to secure results expressed in dollars rather than in the more important intangibles of fighting aircraft. There is much argument about the proper methods to be used for calculating the value of weight in commercial aircraft. The author has chosen some representative figures which are easy to use and which produce results which are consistent with more refined methods of calculation. The methods used and the results are given in Table 1. It will be noted that the results are expressed in terms of earning power. It is thought that the value of $600 per lb is fairly representative of DC-3 operations; in the post-war period, the rates will probably go down and the cruising speed up, with the two effects tending to counteract each other.

This value for weight is so high that it should justify an incredible amount of work in engineering and in the shop. A simple calculation indicates that a saving of 250 lb per airplane is worth more than the present first cost

Table 1 – The Value of Weight
(The data used are all pre-war)

Assumptions:
Revenue per passenger-mile: $0.05 per 200-lb unit
Scheduled cruising speed: 160 mph
Service life: 25,000 hr
Passenger-load factor: 60%

Calculations:
Distance traveled by airplane during its life: 160 mph × 25,000 hr = 4,000,000 miles

Revenue expected per pound-mile: $\frac{\$0.05}{200} = \$0.00025$ lb-mile $= \frac{1}{40}$ ¢ per lb-mile

Potential revenue on 1 lb during life of the airplane:
$0.00025 per lb-mile × 4,000,000 mile = $1000.00
Actual revenue at 60% load factor will be:
0.60 × $1000 = $600.00 per lb, or $37.50 per oz
Gold is $35.00 per oz

of the ship. It is obvious from the study in the table that the cost is not in the weight itself, but in transporting the weight for 4,000,000 miles. Unfortunately, airline operators will not recognize such values for weight saving. Most airplane factories will spend at least $10.00 to save a pound of weight without question, knowing full well that the value of the weight is much greater than the cost of saving it. Changes other than such direct weight savings are of equal importance; in most cases, quite small changes in cruising speed have a tremendous effect on the earning capacity of aircraft. In one case on a modern transport a small device weighing 40 lb saved 200 lb in fuel per flight by increasing the cruising speed 1½ mph. The instance studied was a long-range case, since at 700 miles the device broke even as regards weight saving; while for shorter ranges, it represented a complete loss, causing a decrease in payload, an increase in first cost, higher maintenance, and less safety. The device was, therefore, suitable only for long-range aircraft and should not be used elsewhere. It is quite easy to show that landing gear doors are not desirable from an economic standpoint on transport aircraft flying less than 100 miles. Similar studies indicate that no airplane should have all of the devices which separately might be required in some airplanes.

The detrimental effects of ill-advised maintenance features are frequently overlooked. The writer once was involved in the design of an airplane that probably had the most highly developed maintenance features of any airplane ever built. This airplane was designed to use a definite size hangar, forcing the use of a triple tail, five rudders, five rudder tabs, and a very complicated control system. The tail surfaces of this airplane were at least 15% heavier than a simple tail would have been. The complicated controls resulted in high friction, high cost, and multitudinous dangerous mechanisms. Another feature of this airplane was facilities for refueling from the bottom of the wing. This feature required an extra set of fuel gages at the refueling point, light to illuminate them at night, filler lines directly into the filler caps, the tank filler caps designed as safety valves to protect the tanks from surges on overfilling, check valves in the refueling to prevent the loss of fuel through them in flight, and especially large vents necessary to permit easy escape of air during the high-speed filling operation. These features weighed 20 lb per tank, and every feature was essential to safety. The cost of the installation in loss of revenue was, therefore, $12,000 per tank or approximately $100,000 per airplane (eight tanks). When this fancy fuel system was finished, it was obvious that no one wanted it. Engineers frequently are confused by talking only to maintenance personnel and may, therefore, get a false idea of the relative values of many maintenance items. It takes about two hours to clean an airplane after it has been on a long trip, and so if all the necessary items can be completed comfortably in that time, it is sufficient. It would have been much better in the preceding case to have spent some money in providing better ground facilities in order to obviate the need for carrying around such a mass of questionable apparatus. Quick interchangeability is desirable, but safety and weight are more important. Interchangeability is required when devices are of poor reliability. If a device becomes extremely trustworthy, then its interchangeability ceases to be important. In any case the interchangeable items will not be stocked except at overhaul and a few other important bases. It is easy to show

Table 2 – Distribution of Weight Empty of a Modern Transport

I. Structural group	
Wings	
Tail	
Fuselage	34%
II. Powerplant	
Engines	
Propellers	
Controls	
Accessories	
Nacelle and cowling	
Fuel and oil system	38
III. Fixed equipment	
Surface controls	
Hydraulics	
Electrical communications	
Instruments	
De-icers	
Furnishings	18
IV. Landing gear and tail skid	12
Total	100%

that a higher utilization of the equipment owned will be achieved by assembling most of the spares into aircraft and flying them. Spares in the stockroom represent only an expense; when they are flying, they are earning. The writer feels that airplanes will become more and more like ships, and that spares will tend to become more in the nature of raw material rather than parts. It is interesting to note that naval vessels and the newer merchant ships have no hatches for the removal of engines or boilers. The tool used for removing them is the cutting torch.

During the development of the art of airplane design, it was found that the weight of any item, however small, was important. There is a tendency on the part of many engineers new to the industry to believe that small items are unimportant and that weight saving is necessary only on the larger ones. In order to indicate the relative unimportance of the parts commonly thought important, a rough weight breakdown of the C-54A is given in Table 2. From this table it is easy to see that "the primary structure is of secondary importance." The design of accessory and equipment items furnishes a ready field for weight savings that collectively might be worth far more than the entire cost of the equipment. A good layout of the equipment on an airplane will save more weight than the use of the latest and fanciest alloys together with the best possible structure. It is very easy to expend a great deal of weight in the injudicious routing of engine controls and other items around structural components. The typical connections between the cockpit and an engine on large multiengined airplanes weigh from 10 to 20 lb per ft of run. It is therefore preferable to take the weight by penalties upon the structure in preference to using an elaborate routing of the equipment. Weight savings of considerable magnitude can be made in this manner in addition to a very great saving in installation and maintenance time. Sometimes the most "efficient" structure in the eyes of the structural engineer is not the lightest. Structural engineers frequently describe structures as efficient which have all of their material working. Frequently, a simpler structure will weigh less than one of the most efficient ones. As an example, the tail surfaces of two airplanes shown in Figs. 2 and 3 are of the structurally efficient type and heavy in comparison to the inefficient but light type.

A good way to make parts lighter is to make them smaller. The dimensions of many items are purely a matter of custom with no rationality behind them. Designers have tendencies to make things such as engine controls larger in large airplanes, even though they are still operated

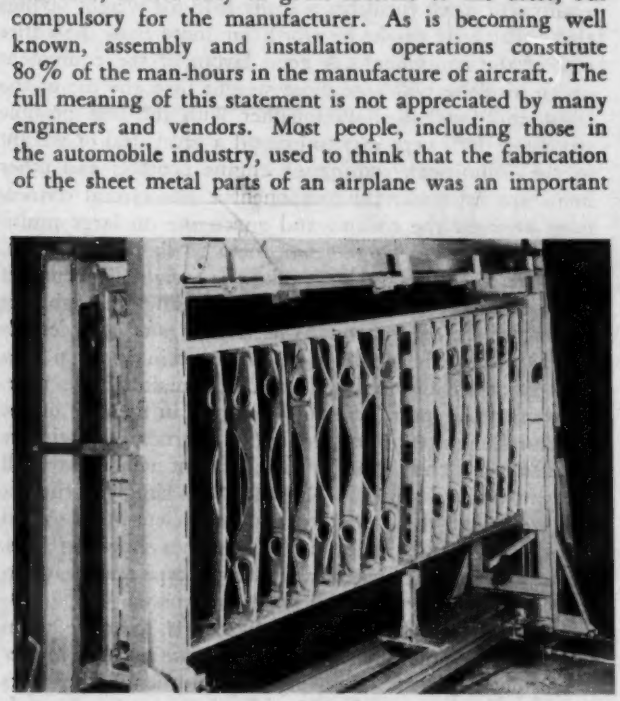
■ Fig. 2 – Structure of C-53 vertical stabilizer

by the same size men. The accessory manufacturers are particularly afflicted with size difficulties.

The increase in electrical generator output from 600 to 7200 w, with only a 40% increase in weight, may be taken as an example of the latent improvements possible in many fields. Almost all other gadgets are capable of similar weight reduction if equal engineering skill and initiative are applied. Another example is the development of the new type of aircraft electrical conductor. This new wire is not only lighter (60 lb saved per plane on the C-54) but is fire resistant and superior to other conductors in high- and low-temperature bending and abrasion resistance.

■ Serviceability and Manufacturability

Airplanes happen to be the most complex objects in quantity manufacture. Interchangeability of components, therefore, is not only of great interest to the users, but compulsory for the manufacturer. As is becoming well known, assembly and installation operations constitute 80% of the man-hours in the manufacture of aircraft. The full meaning of this statement is not appreciated by many engineers and vendors. Most people, including those in the automobile industry, used to think that the fabrication of the sheet metal parts of an airplane was an important part of its manufacture. This is far from the case. The use of the common sheet metal methods and the special techniques developed by the aircraft industry make the fabrication of sheet metal parts a job of great interest but of no particular importance or difficulty.



■ Fig. 3 – Structure of C-54 vertical stabilizer

The principal problems in aircraft manufacture are two in number. The first, and by far the most difficult, is to schedule and keep track of the great number of parts involved. As an analogy to the problem, we might take the job of knowing where each of 200,000 rabbits is at all times in a field 2000 ft square (4,000,000 sq ft). By designing components with fewer parts, the engineer can enormously simplify the problem of factory management and make a considerable saving in the cost of building and operating airplanes.

The second problem is to have parts that fit after they are manufactured. A common error is to believe that because parts are made by good workmen in an accurate jig they will be of the correct dimensions. This is definitely not so. There are three types of error associated with manufacturing operations which cause changes in dimensions during manufacture and thus will cause parts to be in error even though the tools are perfect. Thermal effects produce one of these difficulties. Unfortunately, aluminum alloys have a thermal coefficient of expansion of approximately 0.000012 in. per in. per deg (F), while the coefficient for most ferrous materials is near 0.000006 in. per in. per deg (F). Therefore, as aluminum parts must be made with steel machine tools and in steel jigs, there will be only one temperature at which the aluminum parts will have the correct dimensions. In ordinary shop work this difficulty is not acute, since most of the parts, being ferrous, have the same thermal coefficient as have the tools used. For small aluminum parts, the errors introduced by this cause are still negligible. As an example, take a part 10 in. long manufactured 10 deg hotter than the base temperature. Since the differential coefficient of thermal expansion of aluminum alloys and steel may be taken as 0.000006 in. per in. per deg (F), the part when it is returned to the base temperature will be too short by $10 \times 10 \times 0.000006$ in. per in. per deg, or 0.0006 in. This is not a very serious error, for in airframe manufacture the tolerances generally are at least ± 0.010 in. But if we take the case of a large wing attachment with a 100-in. spacing between bolts, and consider the problem of field interchangeability of a wing made in summer and a center section made in winter, assuming a temperature range of 50 deg (this is easily possible in badly heated Southern plants), we find that the misfit will be $100 \times 50 \times 0.000006$, or 0.030 in. This mismatch assumes that *no other errors exist* and may be in either direction. Thus, in order to be interchangeable, the end holes would have to be made at least 0.015 in. oversize. This problem is a real one and not of a theoretical nature. It occurs continually with parts assembled from different locations, or even with parts made at different times of day in the same plant.

Another source of errors in parts is the changes in dimensions caused by the various manufacturing processes. Heat-treated aluminum alloys, in common with other materials, have internal strains introduced by heat-treatment. When such a part is machined, it will warp in such a manner as to give up some of the potential energy stored by the heat-treatment process. A part 130 in. long shrinks approximately ⅛ in. during machining or approximately 0.001 in.

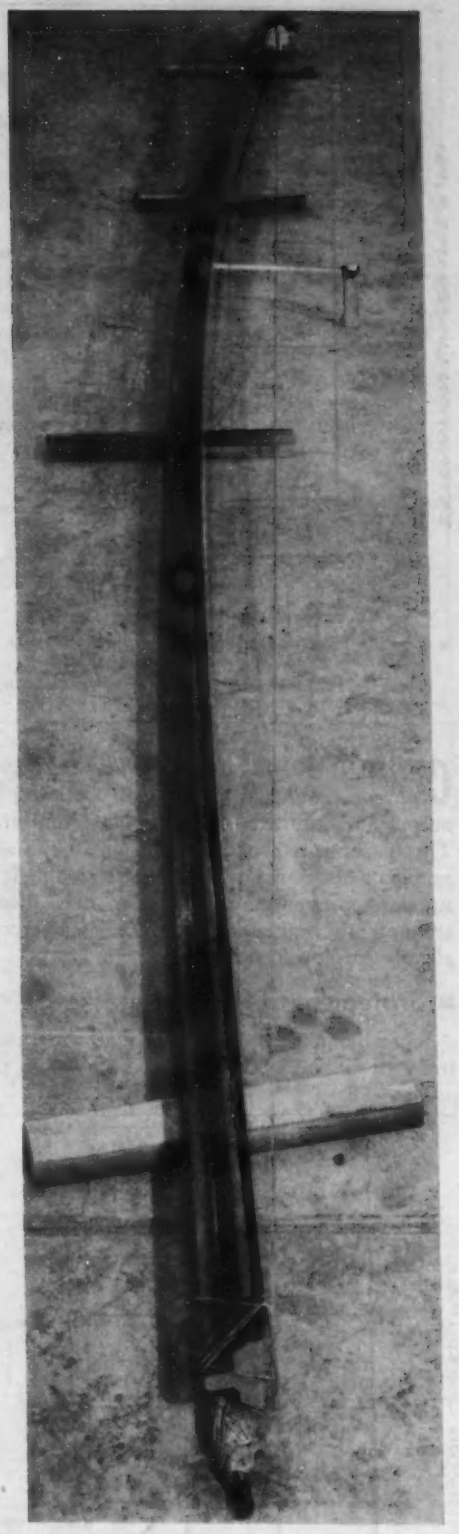
per in. of length. Other parts behave in a similar manner; some warp when the stresses are relieved unsymmetrically. See Fig. 4.

Riveting expands both the riveted parts and the rivets; in fact, rivets are superior to bolts because of this expansion, which fills a drilled and unreamed hole. Bolts, in order to fit tightly, must be placed in reamed holes, which entail additional operations and shop expense. During riveting, aircraft components expand and distort, thus making it impossible to hold dimensions. In many cases an assembly must be machined to accurate dimensions after riveting is completed. The amount of the expansion due to riveting can be controlled with the proper equipment; but variations in driving, hardness of the rivets, oversizing of the holes, and numerous other factors will cause this effect to be of an uncertain magnitude.

In general, welding operations cause components to shrink. Theoretically, spot welding can be made distortionless by recompressing the spots on a modern spot welder, using the forging effect of the recompression cycle to compensate for the shrinkage in the spot weld. Practically, spot-welded assemblies are as variable as riveted assemblies; there is little to choose between them, as the result depends more on workmanship and tooling than on the process.

In addition to the intrinsic errors mentioned above, there is another major difficulty due to the very nature of aircraft. Weight is so critical in all aircraft that extreme efforts are made to conserve it. These efforts result in an airplane being made of thin materials working at high stresses. These conditions result in very large deflections in many airplane structures. In the DC-3, for instance, the wing tips are 20 in. higher in relation to the body when in the air than they are on the ground. More modern airplanes have larger deflections due to the development of structures of greater efficiency designed to smaller load factors. Because of this flexibility, fuselages frequently sag as much as 1½ in. on being removed from their jigs. Other parts deflect noticeable and troublesome amounts on being removed from their supports. The above effects result in no airplane doors being interchangeable. After a number of airplanes have been made, the doors can be made a shape that approximates the shape of the fuselage and then be handworked a small amount to make them fit their individual fuselages. The problem of making truly interchangeable doors will probably never be solved unless large weight sacrifices become possible.

These deflection troubles do not seem to be well enough known to aircraft component manufacturers. One such concern, in spite of many protests, recently made a gun turret which mounted to the fuselage with 16 ft. This design was inoperative in flight, and a tubular ring had to be squeezed in and mounted to the fuselage at three points in order to make the turret operable. Deflections of 0.002 to 0.003 in. per in. of run are rather common in aircraft. Careful designers frequently lay out cable fairheads so that they will be straight in flight and mount tubing so that it has sufficient flexibility to accommodate itself to the large structural deflections of its supports.

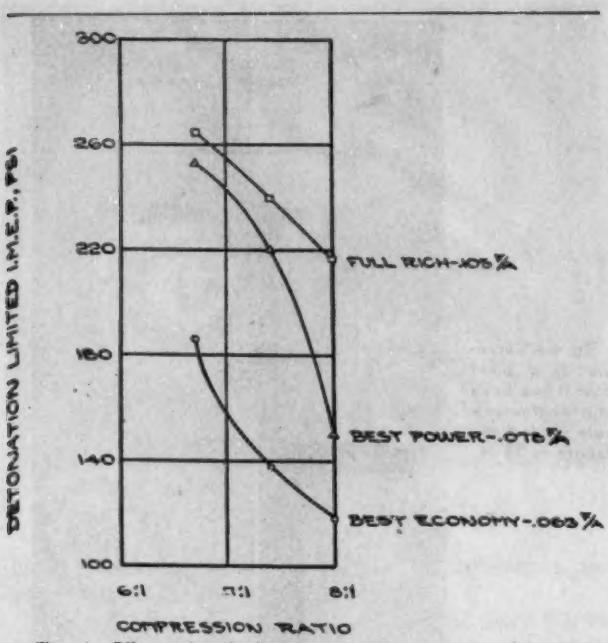


■ Fig. 4 – Curvature of a billet after it has been split lengthwise – scale reads 8 in.; length is 28 ft

■ Conclusion

This brief review of the importance of safety and weight and the difficulties encountered in manufacturing and servicing airplanes has, it is hoped, brought out the need of foresight and pessimism on the part of the designers and vendors of aircraft components. An epigrammatic summary might be:

1. Parts and devices must "fail safe."
2. Airplanes must be built and serviced by simple tools and simple minds.
3. Actuating systems must be autonomous.
4. Simplicity pays off.



■ Fig. 1 – Effect of cylinder compression ratio on detonation-limited output

Engine speed, spark advance, cylinder temperature, and charge-air temperature held constant

The RELATION of COOLING to

by E. A. DROEGEMUELLER and D. S. HERSEY
Pratt & Whitney Aircraft Division and
W. A. KUHRT
United Aircraft Corp., Research Division

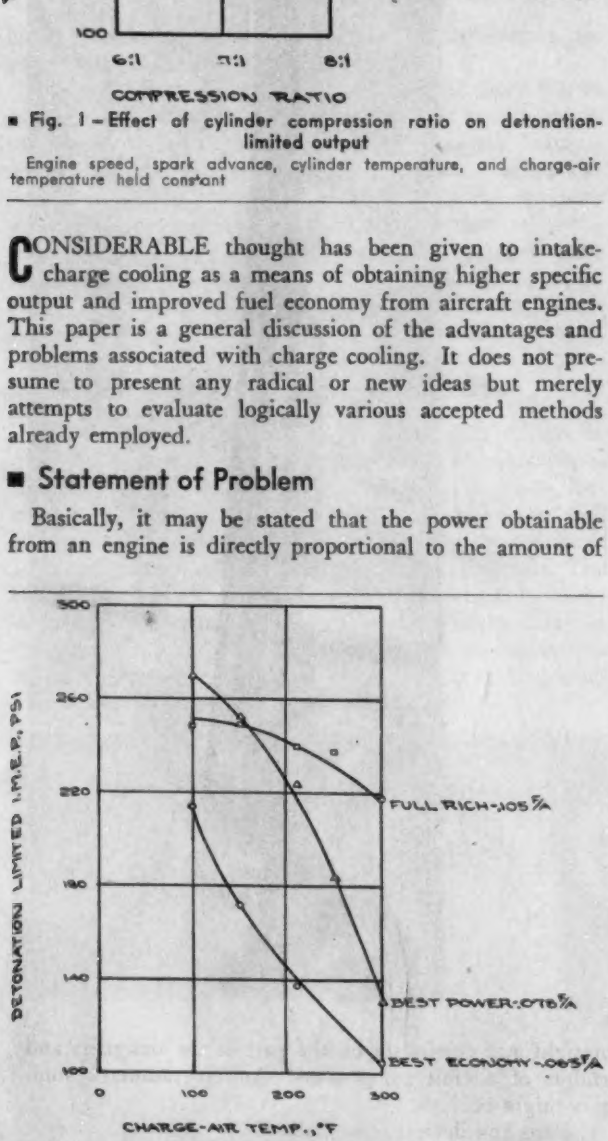
CONSIDERABLE thought has been given to intake-charge cooling as a means of obtaining higher specific output and improved fuel economy from aircraft engines. This paper is a general discussion of the advantages and problems associated with charge cooling. It does not presume to present any radical or new ideas but merely attempts to evaluate logically various accepted methods already employed.

■ Statement of Problem

Basically, it may be stated that the power obtainable from an engine is directly proportional to the amount of air charged into the cylinders. The fuel, fuel-air ratio, spark timing, compression ratio, mechanical efficiency, and valve timing are the variables affecting the proportionality constant. In high-output aircraft engines, attempt is made to derive the greatest amount of mechanical power at the lowest possible fuel consumption consistent with that power – without destroying the engine!

Aside from mechanical considerations, the major output limitation in present-day aircraft engines appears to be detonation, both for maximum- and cruise-power operation.

At maximum output, fuel economy is not too critical, whereas at cruise power, it is all important. With this in mind, the effects of intake-charge cooling on detonation, cylinder charging, and induction-system arrangements will be discussed.



■ Fig. 2 – Effect of charge-air temperature on detonation-limited cylinder output

Compression ratio, engine speed, spark advance, and cylinder temperature held constant

■ Definitions

Before continuing this discussion, various pertinent induction-system temperatures should be defined:

Intake-charge temperature is the temperature of the mixture within the cylinder at the moment the intake valve closes. This is an immeasurable temperature which affects not only the detonation limit but also the charging characteristics of the engine. Among other things, it is dependent upon the type of fuel, intake-port-mixture temperature, and mean cylinder temperature.

Charge-air temperature is a concept which will be used. It is defined as the temperature of the supercharged air which would have existed at the intake port if no fuel cooling had occurred.

Intake-port temperature is the measured mixture temperature at the intake port. This temperature is not a true mixture temperature because of the fact that the fuel may not be entirely vaporized at this point.

None of the above temperatures can be determined with any accuracy because of the inherent shortcomings of thermocouples when employed in high-velocity, pulsating-flow air streams containing free moisture.

[This paper was presented at the SAE National Aeronautic Meeting, New York City, April 6, 1944.]

f INTAKE-CHARGE
o ENGINE
ERFORMANCE

THE need of intake-charge cooling for the suppression of detonation in highly supercharged engines is indicated by the authors of this paper. The relation of detonation-limited output to intake-charge temperature under numerous engine design and operating conditions is illustrated.

Several different methods of reducing the intake-charge temperature are discussed and evaluated, special emphasis being put on the use of intercoolers, aftercoolers, and secondary-liquid injection. It is shown that the choice of any one of these methods depends largely on the type of engine installation and the use of the airplane.

■ ■ ■

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The concept of charge-air temperature as defined above permits correlation of single-cylinder and multicylinder engine tests as based on temperatures measured in the induction system prior to the introduction of fuel.

■ Detonation

An attempt to discuss and define the phenomenon of detonation will not be made, but it is generally accepted as being largely dependent on the character of the fuel and the temperature and pressure within the cylinder *in the last part of the charge to burn,* sometimes called the *end gas.* The end-gas temperature and pressure are related to the type of fuel, intake-port temperature, mean cylinder temperature, amount of boost, spark timing, mixture ratio, cylinder compression ratio, and cylinder design.

The following curves are presented to illustrate a discussion of engine detonation limits. They are typical of modern aircraft engines, and the trends are substantiated by previously published information.

Figs. 1, 2, 3, and 4 are plots of detonation-limited indicated mean effective pressure versus compression ratio, charge-air temperature, cylinder-head reference temperature, and spark advance, respectively.

Fig. 5 shows calculated plots of indicated specific fuel consumption versus cylinder compression ratio at various spark-advance settings.

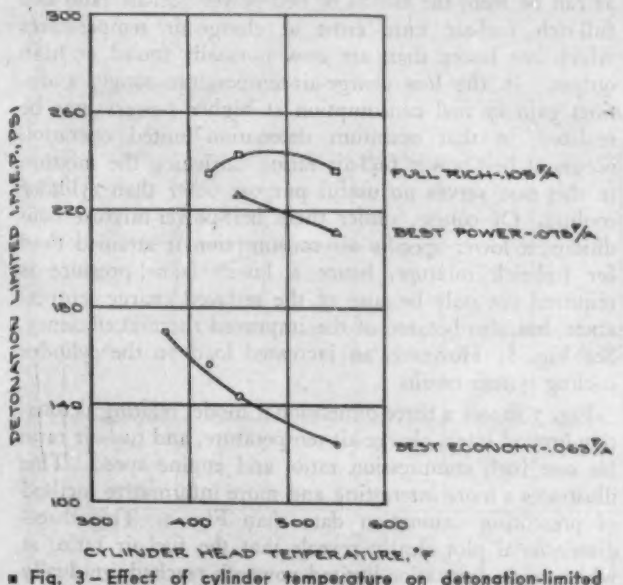
Fig. 6 shows plots of detonation-limited indicated mean effective pressure versus charge-air temperature for three fixed cylinder-head reference temperatures.

Fig. 7 shows two views of a three-dimensional plot of indicated mean effective pressure versus charge-air temperature and fuel-air ratio.

The above curves are based on full-scale, single-cylinder tests in which the fuel quality as well as the other independent variables were maintained constant during each particular investigation.

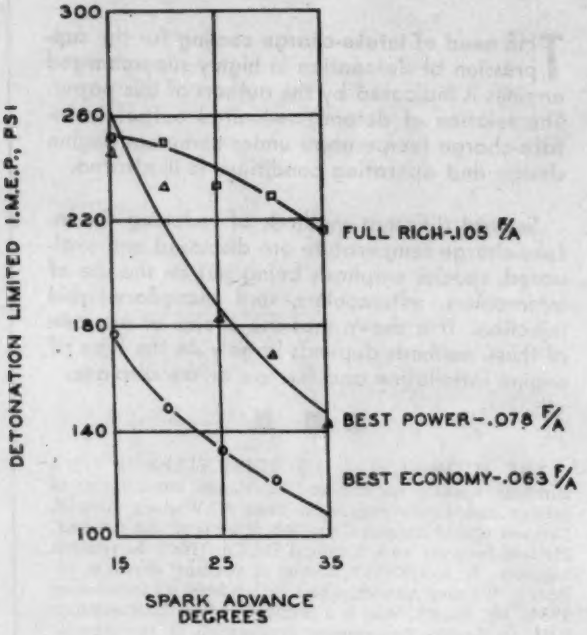
■ Compression Ratio

Compression ratio is one of the many variables in engine design which is the result of a compromise. Examining Figs. 5 and 1, we see that for cruise operation, where fuel consumption is of major importance, a high compression ratio is desirable, but for maximum-power operation where high mean effective pressure is of primary importance, a low compression ratio is preferable. Some current aircraft engines are built with moderate cylinder compression ratio to permit operation at very high boost pressures, thus



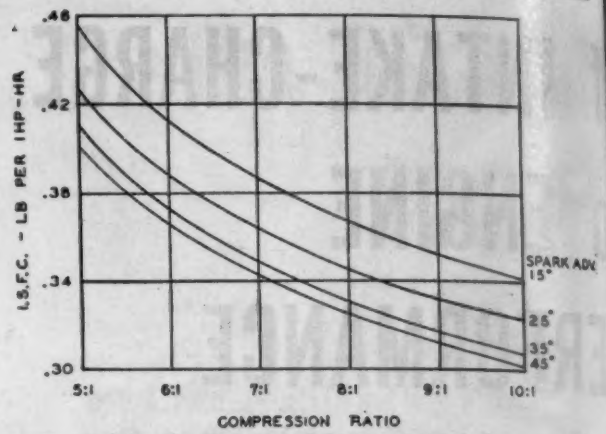
■ Fig. 3 – Effect of cylinder temperature on detonation-limited cylinder output

Compression ratio, engine speed, spark advance, and charge-air temperature held constant



■ Fig. 4 (left) – Effect of spark setting on detonation-limited cylinder output

Compression ratio, engine speed, cylinder temperature, and charge-air temperature held constant



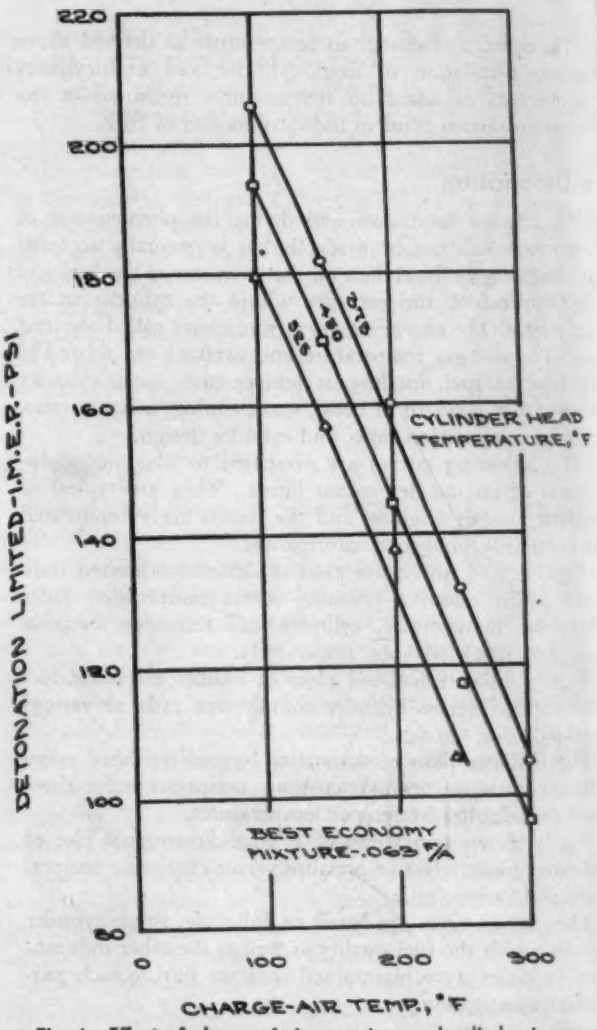
■ Fig. 5 (above) – Variation of best-economy fuel consumption with compression ratio and spark advance

obtaining high specific output by sacrificing cruise economy. Other engines are built with slightly higher compression ratio to improve fuel economy at some expense of high specific output. The current range of compression ratios appears to be between 6:1 and 7:1.

■ Charge-Air Temperature and Fuel-Air Ratio

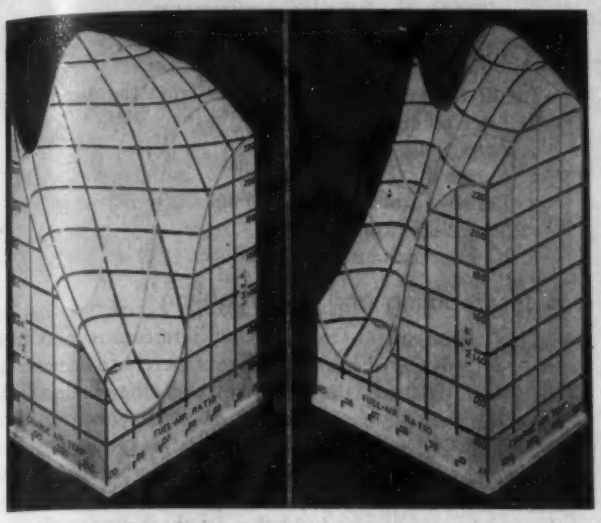
Fig. 2 indicates the wide variations in knock sensitivity not only with fuel-air ratio but also with charge-air temperature. The best-power and best-economy fuel-air ratios (approximately 0.078 and 0.063), as illustrated in Fig. 2, have much steeper temperature-response slopes than the nominal full-rich fuel-air ratio (approximately 0.105) and, as can be seen, the curves of best-power fuel-air ratio and full-rich fuel-air ratio cross at charge-air temperatures which are lower than are now normally found at high output. In the low charge-air-temperature range, a distinct gain in fuel consumption at higher powers may be realized, in that optimum detonation-limited operation occurs at best-power fuel-air ratio. Enriching the mixture in this case serves no useful purpose other than cylinder cooling. Of course, under these best-power-mixture conditions, a lower specific air consumption is attained than for full-rich mixture, hence a lower boost pressure is required not only because of the reduced charge temperature, but also because of the improved thermal efficiency. See Fig. 8. However, an increased load on the cylinder cooling system results.

Fig. 7 shows a three-dimensional model relating detonation-limited imep, charge-air temperature, and fuel-air ratio for one fuel, compression ratio, and engine speed. This illustrates a more interesting and more informative method of presenting detonation data than Fig. 2. This three-dimensional plot clearly reveals that the fuel-air ratio, at which peak detonation-limited power is reached, gradually decreases with decreasing charge-air temperature at a fairly regular and appreciable rate. It also establishes the fact that the influence of charge-air temperature on detonation-



■ Fig. 6 – Effect of charge-air temperature and cylinder temperature on detonation-limited cylinder output at best-economy mixture

Compression ratio, engine speed, spark advance, and mixture ratio held constant.



■ Fig. 7 – Variation of detonation-limited output with charge-air temperature and mixture ratio

limited output has a considerably greater effect at the lean than at the rich mixtures.

The degree of mixture response decreases with decreasing charge-air temperature, indicating that the so-called rich-mixture-appreciation characteristic of a fuel depends on the relative severity of the conditions under which it is being evaluated, and might better be called "lean-mixture depreciation."

■ Cylinder Temperature

The magnitude of the effect of cylinder temperature on detonation-limited engine performance at cruising mixture has not been generally appreciated. Figs. 3 and 6 indicate that this effect can be of considerable importance. On examining Fig. 3, it is found that a 50-deg decrease in cylinder temperature may allow an approximate 14% increase in power at lean mixtures. However, a reduction of cylinder temperature at full-rich mixture permits practically no increase in detonation-limited output.

In spite of the fact that reduced cylinder temperature permits increased detonation-limited power at economy mixtures, some compromise may be desirable because higher thermal efficiency and lower cooling power are realized at high cylinder temperature.

The higher thermal efficiency is due to the fact that, at the higher cylinder temperatures, a lower proportion of the combustion heat is dissipated to the cooling air. Also, the drag cost of obtaining reduced cylinder temperatures may be greater than the improved economy made possible by the increased detonation limits.

■ Spark-Advance Setting

Fig. 4 shows the change in knock limit, at different fuel-air ratios, as spark setting is varied. As in the case of the charge-air temperature curves, the best-power fuel-air ratio curve crosses the full-rich fuel-air ratio curve, and in the range of low spark-advance settings, greater detonation-free power can be realized with the best-power fuel-air ratio. Retarding the spark results in some loss in output and some increase in specific fuel consumption, which may be balanced by a gain in output and fuel consumption obtained by operation at best-power-mixture rather than rich mixture. See Fig. 8. At a given manifold pressure the net result of the combined change may be an increase or decrease in output and cooling requirements, depending on the magnitude of the changes and the particular engine involved.

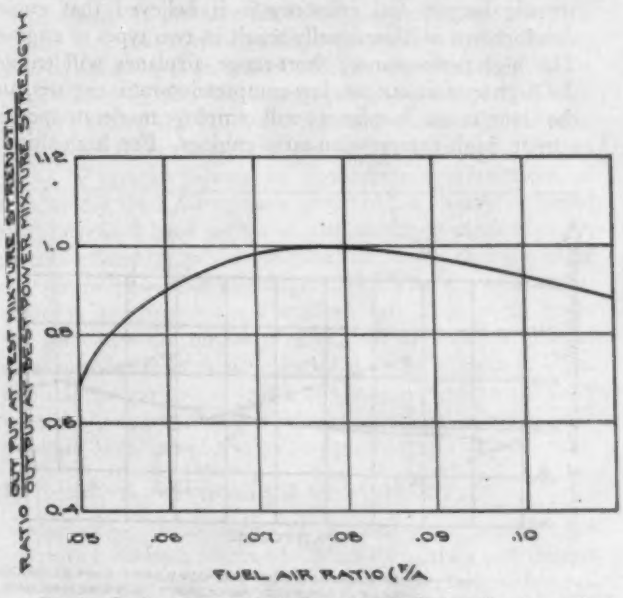
Under cruising conditions, in the nondetonating range, it is not desirable to operate at the lower spark-advance settings because of the resulting poorer economy, as shown in Fig. 5. Therefore, it is found again that a compromise is necessary, unless sufficient reduction in charge temperature is made to permit economical operation at all cruise-power conditions. This compromise may not be too much of a problem, however, since there are means of mechanically controlling the spark-advance setting for different power regimes. In actual practice, a spark advance of approximately 15 to 25 deg is used for maximum-power operation, where the fuel consumption is not important, and a spark advance of approximately 30 to 40 deg may be used for cruise-power operation.

■ Advantages of Intake-Charge Cooling

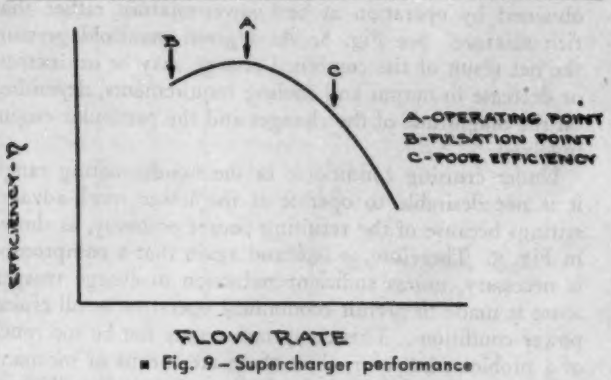
The attainment of a low intake-charge temperature permits a certain freedom of engine design which is not always apparent when charge cooling is first considered. For example, referring to the curves of Figs. 1 and 2, it is obvious that extremely high cylinder mean effective pressures can be obtained merely by a progressive lowering of the cylinder compression ratio, accompanied by an increased boosting by the supercharger, and progressive decreases in the charge-air temperature.

Compression-ratio reductions cannot be carried to extremes as, with the progressive loss in thermal efficiency, the condition would soon be reached when the cylinder output would be employed solely to obtain enough power to operate the supercharger and no net engine output would be available, regardless of the detonation limit.

One noteworthy advantage of high-output operation with reduced compression ratio is that the peak combus-



■ Fig. 8 – Engine output plotted against fuel-air ratio at constant charge airflow



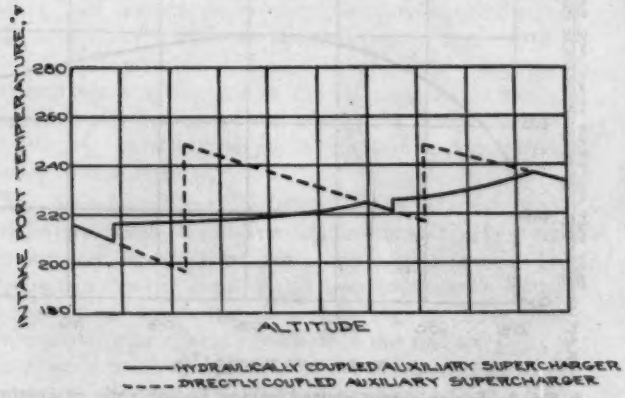
■ Fig. 9 – Supercharger performance

tion pressures within the cylinder are proportionately reduced at a given cylinder output.

Considering the economy point of view, it may be seen that, without charge cooling, the detonation-limited cruising output of an otto-cycle engine is very markedly lowered as the compression ratio is increased. Therefore, for cruising-economy improvements, it becomes very important to provide as high a compression ratio as the fuel will stand with as much mixture cooling and cylinder cooling as can be obtained under a proposed cruising regime. For higher-output cruising operation, a reduction in spark-setting, slight enrichment of the mixture, more charge cooling, higher speed of operation, or even secondary-liquid injection (alcohol or water) can be resorted to to suppress detonation. Reasonable take-off output can be obtained by more drastic applications of these same detonation-suppressing devices. However, it is unlikely that the high specific outputs obtainable with the reduced compression ratios can be approached.

Fig. 2 indicates that charge cooling is relatively more effective in the cruising range of operation than in the high-output range. However, it may be said generally that charge cooling will benefit both output and economy if intelligently applied.

To avoid the undesirable compromise between high specific output and economy it is believed that engine development will eventually result in two types of engines. The high-performance, short-range airplanes will employ the high-specific-output, low-compression-ratio engines, and the long-range airplanes will employ moderate-specific-output, high compression-ratio engines. For high-altitude operation, each will make use of the charge-cooling system best suited to its particular requirements.



■ Fig. 10 – Effect of supercharger drive

■ Charge-Cooling Methods

Intake-charge cooling may result from the use of:

1. Intercooling.
2. Aftercooling.
3. External cylinder cooling.
4. Decreased boost requirements.
5. Secondary liquids.
6. Reduced exhaust back pressure.

■ Intercooling and Aftercooling

First consider the functions of intercoolers and aftercoolers. The use of heat exchangers merely to decrease the charge temperature is obvious, however, the relation of the cooler unit to the engine induction system may not be so apparent. The first notable fact is that a conventional heat exchanger cannot work in an airplane installation until some work, such as supercharging, has been done on the charge air prior to its induction in the cylinder. It must then be remembered that the employment of a heat exchanger must be accomplished with low pressure losses in the induction-air system to ensure any net gain. For example, a normal heat-exchanger installation must accomplish at least 20 deg of charge cooling merely to pay for a 1.5-in. mercury pressure loss at the intake manifold if no full-throttle power reduction is to be expected because of its use. In addition to this, even more cooling is required to compensate for the additional weight and drag involved in its installation.

Under certain critical detonation conditions, the use of a heat exchanger may be justified even though the charge-ducting losses might be considered excessive from the full-throttle-power point of view. For example, a cargo plane cruising at part throttle under certain atmospheric conditions might utilize a relatively small cooler to avoid detonation at the desired cruise power. The duct losses could be regained by increased throttle opening.

On a highly boosted engine, supercharged to high altitudes, it can be demonstrated that a heat exchanger may be installed which will produce much greater temperature reductions than 20 deg for less than 1.5-in. loss in manifold pressure. Its use then becomes not only practical, but in many cases very desirable.

A combination of supercharger and aftercooler becomes in effect a dual supercharging device. The supercharger increases the charge density with an increase in pressure and *temperature*. The heat exchanger further increases the density but with a decrease in temperature and *pressure*. The overall result is a substantial increase in charge density at the intake port. The combined system will always result in some rise in temperature. No heat exchanger can lower the final temperature to the initial temperature; this could be accomplished only by a refrigeration system.

From the above description of the action of the heat exchanger, it can be seen that its function is merely that of a different form of supercharger, and it might be called a "thermal supercharger." The power cost of this supercharging appears in increased aerodynamic drag on the airplane resulting from its installation.

It should be apparent that the greatest effect attainable from an intake-charge cooler will be for the condition in which the cooling is accomplished after all mechanical supercharging is completed and the highest charge temper-

ature is impressed on the cooling surfaces. Practical limitations in engine construction, airplane space, and weight alter this requirement in many cases. So-called aftercoolers, even of the air-liquid-air variety may be too complicated for practical installation on certain types of engines. These practical problems may nullify any gains to be expected from the installation.

In considering the application of an intake-charge heat exchanger to an engine, its effect on the performance of the components of the induction system must be evaluated.

It cannot be expected that the mere addition of a heat exchanger to an existing supercharged engine induction system will always result in increased overall performance. For best performance, the supercharger characteristics must be carefully matched to the other components of the installation in order to meet engine requirements. For example, too high a flow rate through the impeller might be brought about by the addition of an aftercooler in an existing engine-supercharger system. This would cause the supercharger to operate inefficiently (Fig. 9, Point *C*) with corresponding excessive temperature rise and decreased pressure rise. Another installation of heat exchanger might cause too low a flow rate through the impeller (Fig. 9, Point *B*), which might result in low efficiency or even such instability (pulsation) as to prevent operation of the engine.

Regardless of the effect of the heat exchanger on the supercharger performance, the performance of charge ducting and cooling-air ducting arrangements constitutes a very large item in the performance of the installed engine. Poor ducting may nullify the advantages of a heat-exchanger system altogether. For example, an intercooler may be chosen and installed which would, under ideal conditions, perform in accordance with estimates. However, when connected to improper ducting, on either the charge-air or cooling-air sides, the exchanger performance may be ruined, not only because of excessive pressure losses but also because the cooling passages do not run full.

■ Cylinder Cooling

Cylinder cooling has been discussed in great detail in many published articles, and for this reason it will not be discussed at this time. However, the secondary effects of cylinder cooling on intake-charge cooling illustrated in the curves shows the gains which may be realized with lower cylinder temperatures. Improvements in this field may be expected from the aircraft and aircraft-engine manufacturers with the use of more efficient cylinder finning and forced cooling.

■ Decreased Boost Requirements

As a result of improvements in intercoolers and aftercoolers, considerable reduction in supercharging requirements may be realized. As the intake-charge temperature is decreased, the manifold pressure required to produce a given intake-port density will be lowered. In lowering the manifold pressure, the supercharger will not be required to operate at as high tip speeds, and, therefore, the supercharger may operate in a more favorable range of efficiency. Also, with the reduced tip speed, lower temperature rises will be experienced in the supercharger.

Still further reductions in the supercharger temperature-rise performance will be had by using some type of infinitely variable-speed drive. With this type of drive, all operation is at full throttle, and temperature rise is at a minimum, as it is a function of the impeller tip speed.

In present two-stage supercharged engines, where the auxiliary stage does not have an infinitely variable-speed drive, excessive intake-charge temperatures are encountered at the shifting points. These points occur where it is necessary to increase the impeller tip speed, in order to maintain the desired output, as the altitude is increased. Inasmuch as this is a step-by-step process in a directly coupled supercharger, excessive impeller tip speeds are experienced in certain altitude ranges, below the critical altitude, and throttling is necessary to prevent excessive boosting. As indicated in Fig. 10, the critical temperatures are entirely eliminated by the use of an infinitely variable-speed, hydraulically coupled drive to the auxiliary supercharger. This effectively is the sort of control obtained in the exhaust-driven turbosupercharger.

■ Secondary Liquids

Recently it has been announced that water injection is in use on the Pratt & Whitney R-2800 engine. As the result of laboratory experiments, the use of water as a secondary liquid has been suggested many times in past years, but not until recently has this innovation been used on actual combat aircraft. Water and other secondary liquids, such as alcohols, reduce the intake-charge temperatures considerably because of their latent heat of vaporization. This reduces the detonation tendency to such an extent that considerably greater powers may be taken from any given engine, or the secondary liquid may be used to obtain lower primary fuel consumption. The latter of these two advantages is important only when fuel is at a premium. The application of water injection to the R-2800 engines resulted in performance improvements of these engines in accord with previously published work on this subject.

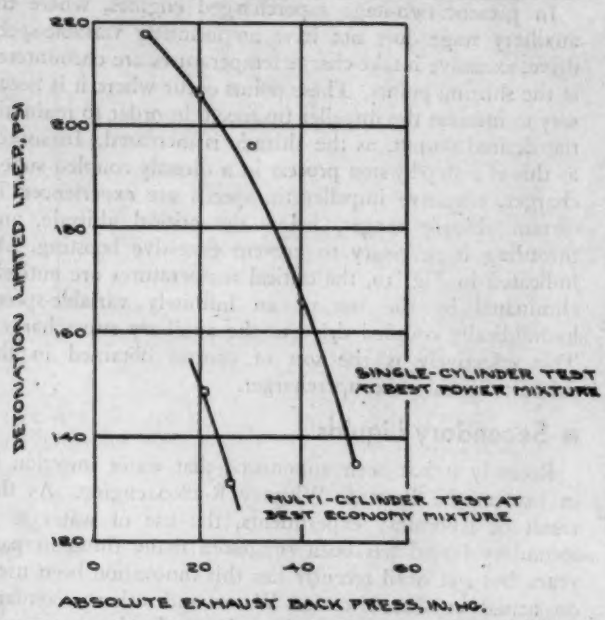
This method of intake-charge cooling is most effective, if properly employed, and can be used on almost any type of engine, provided good distribution of the secondary liquid is obtained.

■ Reduced Back Pressure

One important factor frequently overlooked is the effect of exhaust back pressure on the intake-charge temperature. As the back pressure on an engine is increased, the amount of residual gas left in the cylinder at the time the exhaust valve closes is also increased; and, because of the high temperature of the residuals, the subsequent mixture temperature of fresh charge and residuals is increased. Fig. 11 shows a typical change in detonation characteristics as affected by the exhaust back pressure. It should be noted that increased back pressure also results in reduced volumetric efficiency, which further decreases the allowable maximum power of the engine by increasing the supercharging requirements. Therefore, any attempt to lower the back pressure on the engine pays for itself in greater maximum powers and reduced fuel consumption. The two-stage engine employing a mechanically driven auxiliary supercharger has the advantage of lower exhaust back pressure over that of the turbosupercharged engine.

■ Relative Advantages of Methods

Various potential means of decreasing the charge temperature have been discussed. What then is the best charge-cooling arrangement? The answer to this question depends largely on the type of airplane on which the charge cooling is to be accomplished, the characteristics of the engine, the



■ Fig. 11 – Effect of exhaust back pressure on detonation-limited cylinder output

tactical use of the airplane, and the weight and space limitations.

■ Arrangement of Cooling Units

Modifications of the component parts of the supercharging system so as to increase their efficiencies, or modifications in the arrangement of the component parts may, in a large degree, affect engine and airplane performance. The function of intercoolers and aftercoolers is obvious, but the arrangement and the choice of the use of either may not be quite so apparent. For example, in a two-stage engine, it would seem obvious to use aftercoolers, rather than intercoolers, to take advantage of the greatest temperature difference available within the induction system. However, the use of effective aftercoolers in radial aircooled engines is difficult without a careful redesign of the accessory or manifold sections of these engines. Aftercoolers may be fitted with little difficulty into the present two-bank type of in-line liquid-cooled engines in which only one supercharger outlet is connected to the intake-manifold system.

The optimum arrangement for multistage supercharged engines, from the engine-performance standpoint, would consist of some type of cooler located between each stage plus a cooler located after the last stage of supercharging. The practical use of this type of induction system is almost prohibitive in most aircraft installations, since an excessive amount of ducting would be required for both the intake charge and the cooling air.

The most convenient arrangement for a two-stage radial engine is an auxiliary stage, an intercooler, and a main stage in sequence. With this arrangement maximum practical efficiency is obtained by concentrating the supercharging in the auxiliary stage and reducing the main-stage supercharging to the minimum point where satisfactory engine operation is obtained.

Considering the practical aspects of intercoolers and aftercoolers, it is felt that the specifications of aftercoolers will be harder to meet than those of intercoolers. Un installed on fuel injection engines, aftercoolers would h to withstand backfire pressures which, if they are suffic to cause failure in the core, will present a serious hazard and prevent operation of the engine altoget This fire hazard is especially present in combat operati Failure of an intercooler under such backfire condit will not present a fire hazard and will allow engine o ation with at least one stage of supercharging. For satis tory engine operation, aftercoolers will have to be desig by the engine manufacturer, and probably cannot be alte in size to allow for special operation of different type aircraft. The size of intercoolers, on the other hand, be controlled by the airplane manufacturer to take care his particular requirements. Of course, liquid-cooled af coolers for an aircooled engine are anomalous since th present all the complications and weight difficulties o liquid-cooled-engine cooling system.

■ Expedient Modifications

As a modification to existing airplanes additional char cooling can probably be most easily accomplished by mea of air-liquid-air coolers or by water injection. The a liquid-air cooler system has weight, vulnerability, a complication penalties which are the cost of its appreciab space-saving and aerodynamic advantages. This syste does have the advantage that the cooling is available ov extended periods of time.

The water-injection system is probably simpler, involve very little drag penalty and probably no more weigh increase. It also is just as effective at sea-level as at hig altitudes. This is not true of other systems. Its majo disadvantage is that the economical period of operation somewhat limited both as to time and as to range o engine-output operation. It is unlikely that water injectio will solve any but very special cases of charge cooling fo cruising operations.

It is expected that for installations which are in th design stage it will be found to be more efficient to provid for direct air-to-air intercooling or aftercooling, unless th engine installation is such that duct losses nullify th obvious weight and efficiency advantages of these coolers

Liquid injection will undoubtedly be continued as a charge-cooling means in cases where an increased rang between maximum and cruising output is desirable. This may be found to be true even if charge-cooling radiators are employed. The radiator then takes care of increased cruising-output requirements and the water injection covers the short periods of extreme output needed for take-off or special military high-output conditions. The advantage of liquid injection on high boost engines operating at low altitude is very definite, as the high supercharger-discharge temperatures accompanying the high boost must otherwise be reduced to safe values by very large volume aircoolers.

It is well realized that no airplane manufacturer desires to include more charge-cooler weight or volume than is absolutely necessary. Sometimes it is possible, by engine-design or operating regime changes, to minimize the cooler-size requirements. It is essential, therefore, that close touch be maintained between the engine manufacturer, the airplane manufacturer, the petroleum industry, and the customer to permit the engine manufacturer to make the best choice of engine constants to fulfill the specified performance with minimum charge cooling.